Jesse M. Smith.

PRESIDENT 1909

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS

TRANSACTIONS

VOLUME 31

WASHINGTON MEETING
NEW YORK MEETING
1909

NEW YORK
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29 WEST 39TH STREET
1910

OFFICERS

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

1909

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Terms expire at Annual Meeting of 1909

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Terms expire at Annual Meeting of 1910

PAST PRESIDENTS

Members of the Council for 1909

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Terms expire at Annual Meeting of 1909

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Terms expire at Annual Meeting of 1910

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Terms expire at Annual Meeting of 1911

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PAST PRESIDENTS

THURSTON, R. H.	1880–1882	Died Oct. 25, 1903
LEAVITT, E. D.	1883	Cambridge, Mass.
SWEET, JOHN E.	1884	Syracuse, N. Y.
HOLLOWAY, J. F.	1885	Died Sept. 1, 1896
SELLERS, COLEMAN	1886	Died Dec. 28, 1907
BABCOCK, GEORGE H.	1887	Died Dec. 16, 1893
SEE, HORACE	1888	Died Dec. 14, 1909
TOWNE, HENRY R.	1889	New York.
SMITH, OBERLIN	1890	Bridgeton, N. J.
Hunt, Robert W.	1891	Chicago, Ill.
LORING, CHARLES H.	1892	Died Feb. 5, 1907
COXE, ECKLEY B.	1892–1894	Died May 13, 1895
DAVIS, E. F. C.	1894	Died Aug. 6, 1895
BILLINGS, CHARLES E.	1895	Hartford, Conn.
FRITZ, JOHN	1896	Bethlehem, Pa.
WARNER, Worcester R.	1897	Cleveland, O.
HUNT, CHARLES WALLACE	1898	New York.
MELVILLE, GEORGE W.	1899	Philadelphia, Pa.
MORGAN, CHARLES H.	1900	Worcester, Mass.
WELLMAN, S. T.	1901	Cleveland, O.
REYNOLDS, EDWIN	1902	Died Feb. 19, 1909
DODGE, JAMES M.	1903	Philadelphia, Pa.

PAST-PRESIDENTS AND HONORARY COUNCILORS

1909

SWASEY, AMBROSE	1904	Cleveland, O
FREEMAN, JOHN R.	1905	Providence, R. I.
TAYLOR, FRED. W.	1906	Philadelphia, Pa.
HUTTON, F. R.	1907	New York.
HOLMAN, M. L.	1908	St. Louis, Mo.

According to the Constitution, Article C 27, the five Past-Presidents who last held the office shall be members of the Council, with all the rights, privileges and duties of the other members of the Council.

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Alex. C. Humphreys
F. M. Whyte
F. R. Hutton
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1909

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D. S. Jacobus (2)
Geo. I. Rockwood (5)
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H. W. Spangler (4)

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Chas. B. Dudley (4)
R. C. Carpenter (2)
R. H. Rice (3)

Note.—Numbers in parentheses indicate length of term in years that the member has yet to serve.

SPECIAL COMMITTEES

1909

On a Standard Tonnage Basis for Refrigeration

D. S. Jacobus
A. P. Trautwein
E. F. Miller
G. T. Voorhees
Philip DeC. Ball

On Society History

John E. Sweet
Chas. Wallace Hunt
H. H. Suplee

On Constitution and By-Laws

Chas. Wallace Hunt, *Chairman*
G. M. Basford
Jesse M. Smith
F. R. Hutton
D. S. Jacobus

On Conservation of Natural Resources

Geo. F. Swain, *Chairman*
Charles Whiting Baker
Calvin W. Rice
L. D. Burlingame
M. L. Holman

On International Standard for Pipe Threads

E. M. Herr, *Chairman*
William J. Baldwin
Geo. M. Bond
Stanley G. Flagg, Jr.

On Thurston Memorial

Alex. C. Humphreys, *Chairman*
R. C. Carpenter
Fred J. Miller
Chas. Wallace Hunt
J. W. Lieb, Jr.

On Standards for Involute Gears

Wilfred Lewis, *Chairman*
Hugo Bilgram
Gaetano Lanza
E. R. Fellows
C. R. Gabriel

On Power Tests

D. S. Jacobus, *Chairman*
Edward T. Adams
George H. Barrus
L. P. Breckenridge
William Kent
Charles E. Lucke
Edward F. Miller
Arthur West
Albert C. Wood

On Land and Building Fund

Fred J. Miller, *Chairman*
R. C. McKinney
James M. Dodge

On Student Branches

F. R. Hutton, Honorary Secretary

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1909

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H. V. O. Coes
A. E. Johnson
F. S. King
D. B. Rushmore
A. F. Stillman
G. M. S. Tait
George W. Whyte
S. S. Wyer

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C. W. Obert

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L. V. Goebbels
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A. J. Wood
L. S. Marks
T. M. Phetteplace
G. J. Rathbun
W. Rautenstrauch
S. A. Reeve
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G. L. Knight
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J. P. Sparrow
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Louis C. Doelling
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James D. Andrew
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OFFICERS OF STUDENT BRANCHES

STUDENT BRANCH	AUTHORIZED BY COUNCIL	HONORARY CHAIRMAN	PRESIDENT	SECRETARY
	1908			
Stevens Inst. of Tech., Hoboken, N. J.	December 4	Alex. C. Humphreys	H. H. Haynes	R. H. Upson
Cornell University, Ithaca, N. Y.	December 4	R. C. Carpenter		C. F. Hirshfeld
	1909			
Armour Inst. of Tech., Chicago, Ill.	March 9	C. F. Gebhardt	N. J. Boughton	M. C. Shedd
Leland Stanford, Jr. Universiry, Palo Alto, Cal.	March 9	W. F. Durand	P. H. Van Etten	H. L. Hess
Polytechnic Institute, Brooklyn, N. Y.	March 9	W. D. Ennis	J. S. Kerins	Percy Gianella
State Agri. College of Oregon, Corvallis, Ore.	March 9	Thos. M. Gardner	C. L. Knopf	S. H. Graf
Purdue University, Lafayette, Ind.	March 9	L. V. Ludy	E. A. Kirk	J. R. Jackson
Univ. of Kansas, Lawrence, Kan.	March 9	P. F. Walker	H. S. Coleman	John Garver
New York Univ., New York	November 9	C. E. Houghton	Harry Anderson	Andrew Hamilton
Univ. of Illinois, Urbana, Ill.	November 9	W. F. M. Goss	W. F. Colman	S. G. Wood
Penna. State College, State College, Pa.	November 9			
Columbia University, New York.	November 9			
Mass. Inst. of Tech., Boston, Mass.	November 9		Fredk. A. Dewey	
Univ. of Cincinnati, Cincinnati, O.	November 9			
Univ. of Wisconsin, Madison, Wis.	November 9			

SUMMARY OF MEMBERSHIP

December 31, 1909

United States

Alabama	19	Nebraska	3
Alaska	1	Nevada	5
Arizona	5	New Hampshire	15
Arkansas	2	New Jersey	202
California	74	New Mexico	2
Colorado	30	New York	1062
Connecticut	144	North Carolina	13
Delaware	18	North Dakota	1
District of Columbia	32	Ohio	277
Georgia	19	Oklahoma	1
Hawaii	3	Oregon	11
Idaho	2	Pennsylvania	459
Illinois	241	Philippine Islands	1
Indiana	59	Porto Rico	3
Iowa	9	Rhode Island	69
Kansas	11	South Carolina	3
Kentucky	6	Tennessee	13
Louisiana	30	Texas	15
Maine	15	Utah	9
Maryland	33	Vermont	11
Massachusetts	339	Virginia	28
Michigan	110	Washington	15
Minnesota	22	West Virginia	8
Mississippi	1	Wisconsin	93
Missouri	64	Wyoming	1
Montana	10		
		Total	3619
Address unknown			6

Foreign Countries

Africa	14	India	4
Australia	8	Italy	1
Belgium	5	Japan	9
Canada	45	Mexico	14
Central America	6	New Zealand	1
China	3	Norway	1
Cuba	4	Russia	3
England	45	Scotland	3
Finland	1	South America	11
France	10	Sweden	5
Germany	8	Switzerland	3
Holland	1		
Hungary	2	Total	207

SUMMARY OF MEMBERSHIP

By Residence

December 31, 1909

Membership in United States	3619
Foreign membership	207
Address unknown	6
Total	3832

By Grade

December 31, 1909

Honorary members	15
Members	2565
Associates	398
Juniors	854
Total membership (including life members)	3832

MEMBERSHIP OF GAS POWER SECTION

United States

Alabama	4	Missouri	8
California	7	Nebraska	2
Connecticut	5	New Jersey	14
District of Columbia	5	New York	141
Delaware	1	Ohio	29
Georgia	1	Pennsylvania	25
Illinois	22	Rhode Island	7
Indiana	9	Vermont	1
Kansas	2	Virginia	1
Maryland	1	Washington	2
Massachusetts	25	Wisconsin	17
Michigan	13		
Minnesota	5	Total	347

Foreign Countries

Belgium	1	Mexico	1
Canada	2	Switzerland	1
Germany	1		
		Total	6

GAS POWER SECTION

By Residence

Membership in United States	347
Membership in foreign countries	6
Total membership	353

By Grades

Members of the Society	222
Affiliates	131
Total	353

STUDENT BRANCHES

Armour Institute of Technology	16
Brooklyn Polytechnic Institute	22
Columbia University	4
Cornell University	123
Leland Stanford, Jr., University	13
Massachusetts Institute of Technology	1
Pennsylvania State College	35
Purdue University	3
State Agricultural College of Oregon	10
Stevens Institute of Technology	55
University of Cincinnati	23
University of Illinois	44
University of Kansas	7
University of Wisconsin	26
Total	382

ATTENDANCE AT MEETINGS, 1909

The following figures show the attendance at the several meetings of the Society during 1909:

January 12	NEW YORK		168
February 23	NEW YORK		133
March 9	NEW YORK		251
March 24	NEW YORK		625
April 13	NEW YORK		307
May 4–7	WASHINGTON, D. C. SPRING MEETING	Members...276	
		Guests......333	609
October 12	NEW YORK		192
November 9	NEW YORK		161
December 7–10	NEW YORK. ANNUAL MEETING	Members...628	
		Guests.....435	1063

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WASHINGTON, NEW YORK AND MONTHLY MEETINGS, 1909

CONTENTS

TRANSACTIONS

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOLUME 31-1909

THIS volume contains the papers and proceedings of The American Society of Mechanical Engineers for the year 1909, covering the thirtieth year of the Society's history.

The newly-elected President, Jesse M. Smith of New York, was introduced, as is customary, at the Annual Meeting of the Society in December 1908, an account of which appeared in Volume 30 of Transactions, covering the proceedings of 1908.

The annual report of the Council, presented at the annual meeting of 1909, gives a record of the work of the year, and follows an account of that meeting.

JESSE MERRICK SMITH

JESSE M. SMITH, President of the Society for 1909, was born in Newark, Ohio, in 1848. He moved to Detroit, Michigan, with his father's family in 1862. In 1865 he entered Rensselaer Polytechnic Institute, Troy, New York, remaining there three years. The following year he spent traveling in Europe, and entered L'École Centrale des Arts et Manufactures, Paris, France, receiving after three years of study the degree of M. E. in 1872. During his vacations, he traveled among the manufacturing plants of France, Germany and Belgium, and attended lectures in the Polytechnic Institute in Berlin. After graduation in Paris he traveled three months among the iron and machine works of England.

He began the practice of engineering in 1873, designing and superintending the erection of blast furnaces for smelting iron from native ores with raw bituminous coal in the Hocking Valley, Ohio. He made surveys of coal mines, opened mines and built coal handling machinery for them. He surveyed and constructed railroads from mines to furnaces.

Upon the death of his father in 1880, Mr. Smith returned to Detroit and opened an office as Consulting Engineer. He designed and constructed a high-speed center-crank steam engine with shaft governor, containing the feature of the modern inertia weight governor, and put it in operation driving a Brush dynamo producing 40 arc lights in 1883. In 1890 he presented a paper before the Society on this governor.

He represented the United States Electric Lighting Company in Ohio and Michigan from 1884 to 1886, during which time he erected a number of the early incandescent electric light plants, including one of 1000 lights in the Stillman Hotel, Cleveland, Ohio, which was the first hotel lighted exclusively and continuously by electricity from its own plant. He returned to the work of consulting engineer in 1886 and continued in it until 1898. During this time he designed and erected several power plants and several plants for electric lighting and electric railways; also apparatus for steam heating with exhaust steam in several large manufacturing plants.

He began in 1883 to be called as an expert witness in the U. S. courts in patent litigation. This practice gradually grew and displaced the work of consulting engineer until 1898, when he moved to New York to continue his practice as expert in patent cases exclusively.

Among the notable cases in which he has acted as expert are: steam injectors under the Hancock Inspirator patents; cylinder lubricators for locomotives; roller mills and middlings purifiers for flour manufacture; cyclone dust collectors; quick action air brakes under Westinghouse patents; pneumatic tires for automobiles; automobiles under the Selden patent; induction electric motors under Tesla patents; pressure filters; incandescent electric lamps; steam heating apparatus; typewriters; armored concrete construction; the calculagraph, etc.

He became a member of the Society in 1883 and was a member of the Council as Manager, 1891 to 1894, and Vice-President 1894 to 1896 and 1899 to 1901.

He is a charter member of the American Institute of Electrical Engineers; La Société des Ingenieurs Civils de France; l'Association des Anciens Élèves de l'École Centrale des Arts et Manufactures; The Detroit Engineering Society; the Society for the Advancement of Science; the National Geographical Society; the Engineers Club; the Machinery Club and the Ohio Society of New York.

No. 1229

MEETINGS JANUARY–JUNE

NEW YORK MEETING, JANUARY 12

The first meeting of the Society for the year 1909 was held in the Engineering Societies Building, on the evening of January 12, when a paper on The Transmission of Power by Leather Belting was given by Carl G. Barth.

This paper, with the discussion, constitutes one of the most comprehensive presentations of the subject of belting that has been given before the Society. Mr. Barth has deduced a theory of belting based on the well-known experiments of Lewis and Bancroft and other engineers who have investigated different factors of the belting problem. Moreover, in systematizing manufacturing plants and especially machine shops, where the scientific operation of machine tools is involved, a careful study was made of the whole belting problem, resulting in additional material for his paper.

Following the discussion pertaining strictly to the subject-matter of the paper, there was a general discussion upon the transmission of power by electricity and by rope, and by the modern types of chains used for power transmission. Written discussions were submitted by the following: A. F. Nagle, Prof. Wm. W. Bird, Prof. C. H. Benjamin, H. K. Hathaway, Prof. L. P. Breckenridge, and Prof. W. S. Aldrich. Oral discussions were given by Henry R. Towne, Wilfred Lewis, W. D. Hamerstadt, Fred. W. Taylor, Charles Robbins, Geo. N. Van Derhoef, Walter C. Allen, Dwight V. Merrick, Fred. A. Waldron, S. B. Flint and A. A. Cary.

NEW YORK MEETING, FEBRUARY 23

A meeting was held in the Engineering Societies Building on Tuesday evening, February 23, the subject for discussion being Safety Valves. The meeting was opened by Frederic M. Whyte, general mechanical engineer of the New York Central Lines, with a paper upon Safety Valves, giving special attention to locomotive practice. He was followed by Philip G. Darling, mechanical engineer with Manning, Maxwell & Moore, with a paper on Safety-Valve Capacity.

The papers and discussion, which covered locomotive, marine and stationary practice, as well as the use of safety valves on heating boilers, brought together such late data as were available and emphasized the need of further information for the purpose of establishing a more rational and uniform practice.

The papers were discussed by: L. D. Lovekin, A. C. Ashton, A. B. Carhart, E. A. May, H. O. Pond, F. J. Cole, Dr. Chas. E. Lucke, Jesse M. Smith, G. P. Robinson, Wm. H. Boehm, H. C. McCarty, M. W. Sewall, Geo. I. Rockwood, A. A. Cary, Dr. A. D. Risteen, F. L. DuBosque, N. B. Payne, Frank Creelman. The discussion was adjourned to the Spring meeting.

NEW YORK MEETING, MARCH 9

A particularly interesting occasion was the lecture on Modern Physics, given by Dr. William Hallock, Professor of Physics, Columbia University, on Tuesday evening, March 9.

The lecture included a review of discoveries introductory to the X-ray, radio-activity and allied phenomena; experimental demonstration of different forms of radiation, including heat; development of the essential identity of radiant light, heat and Hertz waves, together with the evidence of the electro-magnetic nature of light radiations; differentiations between these forms of radiation and those of so-called radio-active material, followed by the bearing of the facts developed by radio-activity upon the possible genesis of the chemical elements; the kinetic theory of gases and its relation to the modern theory of solutions; the moving ion as the determining factor in electrical conduction; the distinction between the chemical and the physical ion; the atom and the relation of its structure to the phenomena of radiation and absorption; the principle of relativity and its relation to the structure of the atom and the electron; the universal application of the *force, mass, time* theory to molecular and cosmic phenomena.

JOINT MEETING ON CONSERVATION

A meeting of the national engineering societies on the conservation of our natural resources was held in the Engineering Societies Building on the evening of March 24. Onward Bates, President

Am. Soc. C. E., who was expected to preside, was unable to attend, and Dr. James Douglas, Past-President Am. Inst. M. E., acted as chairman. At the opening of the meeting, the Chairman announced a congratulatory telegram from President Taft, which was read by John Hays Hammond, President Am. Inst. M. E.

In his opening remarks Dr. Douglas said that in a great movement of this kind there could be no dividing line between engineers in different branches of the profession. The great inventions like that of the Bessemer process had required a combination of the skill of engineers who had specialized in different fields. He said that in looking back we must be struck with the advance made in the reduction of waste in the use of natural supplies, especially in saving coal, both in mining it and in using it in metallurgical work.

The first address was upon The Conservation of Water, by John R. Freeman, Past-President Am. Soc. M. E., Mem. Am. Soc. C. E., consulting engineer of the Department of Additional Water Supply for the City of New York. He spoke of the relation of stream flow to lumbering, emphasizing the importance of accurate stream measurements in order to obtain precise knowledge of the effect of forests and of the value of water powers. Interesting figures were given, comparing the efficiency of turbines of the old days with those of the present time. Other phases of the conservation of water, such as the purity of the water courses, navigation, irrigation, etc., were considered. He recommended the collecting of facts by the different States, regarding the notable opportunities for power development within their borders, and the making of careful surveys, thus placing reliable information at the disposal of those inclined to take advantage of such natural opportunities for power.

The address of Dr. R. W. Raymond, Secretary Am. Inst. M. E. was upon Conservation by Legislation. He defined true conservation as the diminution not of use but of waste. The best method for the prevention of waste is by the progressive education of the people, rather than by legislation. He urged that government information pertaining to natural resources and their conservation should be collected with care and not hurried, and stated without bias or argument in favor of any measure or policy. Hasty and ill-considered legislation, especially if advocated by selfish interests, is a peril. He dealt with specific examples of such legislation and urged that the work of the departments of the Federal government should be carefully planned in advance instead of expanding without a definitely arranged plan.

Charles Whiting Baker, Mem. Am. Soc. M. E., Editor of Engineering News, spoke on The Waste of our Natural Resources by Fire. He gave new statistics upon the fire laws in the United States, with the striking illustration that we are burning every year in this country a street of buildings a thousand miles long that would reach from New York to Chicago. That this destruction is not necessary is proved by the experience of European countries where the per capita fire loss is in most cases only a few cents annually, while in this country it is $2.50. Referring to the destruction by forest fires, he said that effective laws for the protection of forests must be enacted before capital will be invested in the development or preservation of timber lands.

The last address was by Lewis B. Stillwell, Mem. A. I. E. E., consulting electrical engineer, upon Electricity and the Conservation of Energy. He illustrated by interesting figures the function of electricity in the conservation of power resources, showing results accomplished in three typical cases, namely, the plants of the Niagara Falls Power Co., the Northeast Coast Power System at Newcastle-upon-Tyne, and the plants of the Interborough Rapid Transit Co., New York. The Niagara plant showed the possibility in water power development and the Northeast Coast plant the economy resulting from the substitution of large steam-driven units for small steam plants, widely distributed. In the case of the Interborough Company, comparisons were made of the cost under the present system of electrical distribution and that which would have obtained if locomotives had been used instead.

ST. LOUIS MEETING, APRIL 10

A meeting of members of the Society residing in St. Louis and vicinity was called by Wm. H. Bryan, member of the Meetings Committee, and held on Saturday evening, April 10. Prof. E. H. Ohle acted as secretary and about twenty engineers were present. This was the first monthly meeting of the Society to be held outside of New York City. All present expressed themselves in favor of local meetings and it was voted that a committee of three, composed of the chairman and two others appointed by him, should be formed to lay out a plan of organization and to report in sixty days.

The following topics were discussed: *a* local organization with occasional professional and social meetings; *b* increase in membership; *c* contributions to The Journal; *d* making up a party to attend the

Spring Meeting at Washington, May 4-7; *e* extending an invitation to the Society to meet in St. Louis at some future time; *f* other means of promoting the Society's welfare, not only locally, but generally.

JOHN FRITZ MEDAL AWARD

The John Fritz Medal, the only medal which the four National Engineering Societies confer, was presented to Charles T. Porter, Hon. Mem. Am. Soc. M. E., on Tuesday evening, April 13. The presentation took place in the auditorium of the Engineering Societies Building, before distinguished invited guests and an audience representing the entire engineering profession. The medal was conferred upon Mr. Porter for his work in advancing the knowledge of steam engineering and for improvements in engine construction. Addresses were made by Dean W. F. M. Goss of the University of Illinois, upon The Debt of Modern Civilization to the Steam Engine as a Source of Power; by Prof. F. R. Hutton of Columbia University, Honorary Secretary Am. Soc. M. E., on The Debt of the Modern Steam Engine to Charles T. Porter; by Robert W. Hunt of Chicago, on The Debt of the Era of Steel to the High Speed Steam Engine; by Frank J. Sprague of New York, on The Debt of the Era of Electricity to the High-Speed Steam Engine.

Henry R. Towne, Past-President Am. Soc. M. E., and Chairman of the Board of Award of the John Fritz Medal for 1909–1910, presided at the meeting, and in his opening remarks spoke briefly of the origin and history of the medal, introducing Dean W. F. M. Goss of the University of Illinois.

At the close of Professor Goss's address, Mr. Towne in a short introductory speech recalled that Mr. Porter was the third person and the first American to whom was accorded the distinction of Honorary Membership in The American Society of Mechanical Engineers. On account of this relation, Mr. Porter was introduced by Jesse M. Smith, President of the Society, who said, by way of introduction:

The John Fritz Medal, established in 1902 by the American engineering profession as a meed of recognition for 'notable scientific or industrial achievement,' was awarded in the year 1908 by a board representing the four National Engineering Societies, to a distinguished mechanical engineer for 'his work in advancing the knowledge of steam-engineering and for improvements in engine construction.' I present to you, and to this company, the engineer to whom this high distinction has been granted.

He is honored because he saw the possibilities of the high-speed steam engine; because his mechanical genius in design made those possibilities real; and because he recognized the necessity for, and then applied, the very best mechanical construction to the realization of his ideals.

He then introduced into the development of the power plant an idea and an influence so revolutionary as to make an epoch in the history of the art of engine building; and which has been as world-wide in its effects as the use of the reciprocating engine.

Many of the present generation of engineers have inherited, without effort and often without knowledge of their origin, the results which cost him many years of painstaking study and experiment to establish.

That he may receive the John Fritz Medal awarded to him, I now have the honor to present Charles Talbot Porter.

Mr. E. Gybbon Spilsbury, Chairman of the Board of 1908, by which the award was made, said in presenting the medal to Mr. Porter:

Under instructions from the Board of Award of the John Fritz Medal, it is my privilege and pleasure to inform you that for your work in advancing the knowledge of steam engineering and for improvements in engine construction, you have been chosen as the worthy recipient of the medal for the year 1908–1909.

This medal was instituted in 1902 to commemorate the 80th anniversary of the successful and honored career of our beloved colleague John Fritz, and its award by a committee selected from the membership of the four great engineering societies of the United States is the highest honor which the engineering profession can confer on any of its members.

Charles Talbot Porter, in the presence of this distinguished company, I now present you this medal, together with an engraved certificate of the award, and confer upon you all the rights and honors and the distinction which attach to this emblem. May you live long and happily to enjoy the appreciation which is your due at the hands of those you have so benefited by your work.

After the presentation, Mr. Towne read the following telegram from John Fritz:

With all my heart regret my inability to be with my dear friends and associates this evening. I cannot be with you in person, but I will be with you in spirit. Please convey to my dear friend Porter my sincere congratulations and best wishes.

Congratulatory cablegrams were also read from Wm. H. Maw, Editor of *London Engineering*, from the sons and grandsons of Wm. A. Hoyle, with whom Mr. Porter was associated in his early work, the Iron and Steel Institute of Great Britain, the Institution of Mechanical Engineers of Great Britain, E. D. Leavitt, Past-President Am.Soc.M.E., an early associate of Mr. Porter, and many others.

The addresses of the evening followed.

ADDRESS OF DEAN W. F. M. GOSS

Dean W. F. M. Goss spoke of the debt of modern civilization to the steam engine. Dreams of the possibilities of steam belong to the days of Addison, Steele, Swift and Defoe; days when there were brilliant men of letters, triumphs in architecture, achievements on the battlefield, but when there were no means for performing industrial work. There were no large factories in England because there was no way by which their machinery could be driven. Mines were abandoned because they were flooded with water; women and girls were toiling in the mines amid suffering and degradation. The movement of merchandise by land was laborious and traveling by sea slow and dangerous.

Into the midst of such conditions came the steam engine. It freed the mines of England from water, revived dormant industries, introduced new systems of manufacture, supplied power, water and effective means of sanitation to cities, and was later supplemented in all these respects by electricity for lighting, power and transportation. Steam usurped the place of wind in the propulsion of ships, and through the agency of the locomotive has carried civilization to the farthest ends of the earth. These achievements are direct contributions to the upbuilding of civilization, the key-note of which is service. The dwellers on the earth are beginning to see that if one nation suffers severely, all are likely to suffer in some degree, and they are learning sympathy for their fellow-men.

ADDRESS OF PROF. F. R. HUTTON

It was assigned to Prof. F. R. Hutton to speak in detail of the debt of the reciprocating steam engine to the pioneer work of Mr. Porter. It owes to him the first vision of the advantages that come from making the crank shaft turn at a high rate of revolution, whereby the weight of the motor per horsepower is reduced. From this seed-thought has sprung the modern design of the motor for the self-propelled vehicle and for the aëroplane. The high speed involved the solution of difficult problems, owing to the necessity for starting and stopping heavy parts of the mechanism in each revolution. To Mr. Porter we owe the recognition of these problems.

Perhaps the most important debt of all is the requirement that the standard of mechanical construction in the high-speed engine must be of the highest type. We owe to Mr. Porter many manufacturing details which now are commonplaces of modern practice.

Mr. Porter created a form of steam-engine condenser to be attached directly to the engine and operated at a much higher rate of speed than that at which the ordinary pump could be used; and finally, invented a sensitive steam engine governor in two forms.

The address closed with a tribute to Prof. Chas. B. Richards, associated with Mr. Porter's early work of designing, and John F. Allen, who had conceived many details of the first high-speed engine which Mr. Porter combined into a harmonious whole.

ADDRESS OF ROBERT W. HUNT

Robert W. Hunt said it was scarcely conceivable that one could have witnessed in a single lifetime the remarkable development in the steel industry which he had observed since the birth of the Bessemer processes. These accomplishments were made practically possible by the discovery of a more rapid power. The early processes were deliberate because man was habituated to slow movements. The first power came from the slow-turning water wheel; later from the slow-speed steam engine. Faster movements were obtained through gears and belts. Among the first engineers to attach the rolling mill engine direct to its train of rolls were John and George Fritz, but the speed of the stroke of their engine was limited. Charles T. Porter was the first to give to the rolling-mill a controllable direct-connected economical high-speed engine.

Mr. Hunt referred to two engines in a rolling-mill plant in Troy, N. Y., in 1876. One set of rolls was driven by a walking-beam low-pressure engine, taken from the steamboat Swallow, a Hudson River boat, and the other set was driven by Porter-Allen engines. The contrast between the steamboat engine with a slow speed of 35 or 40 r.p.m., and Mr. Porter's little engines, humming away at high speed, and accomplishing much greater results, was an instructive sight.

ADDRESS OF FRANK J. SPRAGUE

Frank J. Sprague recalled that in 1867, at the French exhibition, Charles T. Porter installed two Porter-Allen engines, the only high-speed engines exhibited, to drive generators for supplying current for lighthouse apparatus. While these engines were not directly coupled, it is a curious fact that the piston speeds and revolutions were what is common today in isolated direct-coupled plants. In the dozen years following, Mr. Porter built many engines with certain common characteristics, high piston speed and revolutions, solid engine bed and babbitted bearings, but there was no direct coupliug to dynamos until 1880, when Mr. Porter installed a high-speed engine for Mr. Edison in his laboratory at Menlo Park. Shortly after this Mr. Porter was invited to construct for the Edison Station at Pearl Street, New York, the first of a series of engines for so-called steam dynamos, each independently driven by a direct-coupled engine.

Mr. Sprague likened the relations of electricity and the high-speed engine, not to debtor and creditor, but rather to a close partnership, an industrial marriage, one of the most important in the engineering world, that of the prime mover and the electric generator. Here were two machines, destined to be joined together, economizing space, increasing economy, augmenting capacity, reducing investments, increasing dividends. Primarily and largely due to Mr. Porter, the high speed possibilities of the engine were commercially demonstrated.

BOSTON MEETING, APRIL 16

A meeting was held in Boston in the auditorium of the Edison Building, on April 16, 1909, to discuss the advisability of holding meetings of the Society in that city. Irving E. Moultrop, Manager of the Society, was elected temporary chairman and Ralph E. Curtis temporary secretary. About 160 members and guests were present, including the President and the Secretary of the Society. There was a general discussion in which the following participated: Irving E. Moultrop, Henry Bartlett, Henry F. Bryant, Vice-President of the Boston Society of Engineers, James D. Andrew, Fred R. Low, Paul Winsor, Prof. W. W. Bird, Prof. Geo. F. Swain, Prof. L. S. Marks, Prof. D. C. Jackson, Prof. Gardner C. Anthony, Francis W. Dean, E. G. Bailey, Prof. C. G. Lanza.

During the discussion there were brief addresses by the President and the Secretary. President Smith said that it is the desire of the officers of the Society that these meetings be as free and open as is consistent with the traditions and the high professional standards which the Society has maintained during its thirty years of experience. He emphasized the fact that such meetings are meetings of the Society as a body rather than of local sections or branches. Papers presented would be published in The Journal when accepted, making it possible to discuss them in all the cities where meetings are held. He emphasized the Society's friendly spirit of coöperation with other engineering societies and the particular esteem in which engineers and members held the Boston Society of Civil Engineers.

Secretary Rice said it had become evident that two conventions a year are not sufficient for a national society and that the holding of meetings more frequently in one place does not create a national spirit. He said the question before the meeting was, how the engineering profession of Boston and vicinity can best get together for the common and individual good; and stated that The American Society of Mechanical Engineers desires to do what will best serve the profession. He expressed the hope that that would be accomplished by bringing together the various organizations in a common headquarters rather than by the formation of a new organization. Coöperation and coördination, he declared, should be the motto of the profession.

ST. LOUIS MEETING, MAY 15

A meeting was held at the Missouri Athletic Club, St. Louis, on May 15, to discuss further the question of holding meetings of the Society in St. Louis. William H. Bryan, member of the Meetings Committee, presided, and Prof. E. L. Ohle acted as Secretary.

The report of the committee on organization, recommending that the Society coöperate with the Engineers Club of St. Louis in the matter of meetings and publication, was presented and discussed by the following: M. L. Holman, Past-President Am.Soc. M.E., R. H. Tait, Wm. H. Bixby, Thomas Appleton, Professor Westcott, F. L. Jefferies, W. M. Armstrong, Prof. H. Wade Hibbard, J. A. Laird, Victor Hugo, E. A. Fessenden. In opening the discussion Mr. Holman said that the question of enlarging the sphere of usefulness of national engineering societies without interfering with local organizations is one which has been given much thought by the engineers of the country. It has taken years to bring the St. Louis Engineers Club, an earlier organization than The American Society of Mechanical Engineers, up to its present standing and it could not afford to take steps which would interfere with its usefulness or impede its growth and importance. This movement, however, was not intended to antagonize local clubs but was for the purpose of bringing more engineers into the societies, both local and national, thus benefiting both organizations.

Following the discussion, the report was unanimously adopted and the meeting concluded with an interesting running account, given by Professor F. H. Vose, of the papers and discussion presented at the Washington meeting.

BOSTON MEETING, JUNE 11

In accordance with the plans of the preliminary meeting of May 15, a professional meeting was held on June 11, and the paper on Small Steam Turbines given by George A. Orrok at the Washington meeting was presented for further discussion.

Prof. Ira N. Hollis, who presided, first outlined the work proposed for the meetings of the Society in Boston, saying that the committee was planning a number of meetings to be held during the fall and winter. The subjects of several unusually timely papers promised for these meetings were announced. As indicating the large number of engineers in the vicinity who might attend, it was stated that notices had been sent to 950, of which number 340 were members of the Society.

The meeting was addressed briefly by Secretary Rice, after which Mr. Orrok's paper was read by Prof. E. F. Miller and the following joined in the discussion: Dr. L. C. Loewenstein, J. A. London, Chas. B. Rearick, F. B. Dowst, Chas. B. Edwards, V. F. Holmes, J. S. Schumaker, Prof. C. A. Read, Prof. I. N. Hollis, Prof. E. F. Miller, John T. Hawkins, R. H. Rice, Chas. H. Manning, C. P. Crissey, Chas. B. Burleigh.

SPRING MEETING, WASHINGTON, D. C.

The 58th meeting of the Society was held in Washington, D. C., a the New Willard Hotel, May 4-7. The total registration was 609, o which 276 were members. Fewer professional sessions than usual wer arranged by the Meetings Committee in order that visiting member and their guests might avail themselves of opportunities to see place of interest at the national capital.

PROGRAM

OPENING SESSION

Tuesday Evening, May 4, at 8.15 o'clock

Reception of the members by the Washington Society of Engi neers and the local members at the New Willard Hotel. Music b the Marine Band.

Address of welcome by Hon. Henry B. F. Macfarland, Presiden of the Board of District Commissioners.

Response by Jesse M. Smith, President of the Society.

SECOND SESSION

Wednesday Morning, May 5

Business Meeting: Reports of committees, tellers of election; new business.

A UNIQUE BELT CONVEYOR, Ellis C. Soper.

Discussed by T. A. Bennett, Harrington Emerson, Fred J Miller.

AUTOMATIC FEEDERS FOR HANDLING MATERIAL IN BULK, C. Kembl Baldwin.

Discussed by T. A. Bennett.

A NEW TRANSMISSION DYNAMOMETER, Prof. Wm. H. Kenerson.

Discussed by A. F. Masury.

POLISHING METALS FOR EXAMINATION WITH THE MICROSCOPE Albert Kingsbury.

Wednesday Afternoon

Special exhibition drill by troops at Fort Myer.

Wednesday Evening

Illustrated lecture by Arthur P. Davis, Chief Engineer of the U. S. Reclamation Service, on HOME-MAKING IN THE ARID REGIONS.

THIRD SESSION

Thursday Morning, May 6

GAS POWER SESSION

Report of the Standardization Committee.
MARINE PRODUCER GAS POWER, C. L. Straub.
Discussed by Geo. Dinkel, Henry Penton, I. E. Moultrop, H. M. Wilson, E. T. Adams.
OPERATION OF A SMALL PRODUCER GAS POWER PLANT, C.W. Obert
Discussed by J. A. Holmes, J. H. Norris, W. A. Bole.
A METHOD OF IMPROVING THE EFFICIENCY OF GAS ENGINES, Thos. E. Butterfield.
Discussed by A. M. Greene, Jr., W. O. Barnes.
OFFSETTING CYLINDERS IN SINGLE-ACTING ENGINES, Prof. T. M. Phetteplace.
Discussed by W. H. Herschel, J. H. Norris.

Thursday Afternoon

Reception of members by William H. Taft, President of the United States.

Thursday Evening

Address, THE ENGINEER IN THE NAVY, by Rear-Admiral George W. Melville, Ret., Past-President Am. Soc. M. E.

Address, REAR-ADMIRAL MELVILLE'S SERVICE TO THE ENGINEERING PROFESSION AND TO THE NATION, and presentation to the National Museum of a portrait of Rear-Admiral Melville; by Walter M. McFarland of Pittsburg, Pa. Acceptance of the portrait by Dr. C. D. Walcott, Secretary of the Smithsonian Institution, representing the Nation.

FOURTH SESSION

Friday Morning, May 7

PROFESSIONAL SESSION

SMALL STEAM TURBINES, George A. Orrok.

Discussed by W. D. Forbes, R. H. Rice, Prof. R. C. Carpenter, H. Y. Haden, F. D. Herbert, W. E. Snyder, W. T. Donnelly, F. H. Ball, C. A. Howard. The discussion was continued at the Boston meeting, June 11.

OIL WELL TESTS, Edmund M. Ivens.

Discussed by F. A. Halsey, S. A. Moss, J. E. Callan.

SAFETY-VALVE discussion, continued from the ebruary meeting in New York: F. L. Pryor, E. F. Miller, G. H. Musgrave, A. B. Carhart, S. B. Paine, M. W. Sewall, A. C. Ashton, A. F. Nagle, J. J. Aull, A. J. Hewlings.

SPECIFIC VOLUME OF SATURATED STEAM, Prof. C. H. Peabody.

Discussed by Prof. W. D. Ennis.

SOME PROPERTIES OF STEAM, Prof. R. C. H. Heck.

Discussed by S. A. Moss, G. A. Goodenough.

A NEW DEPARTURE IN FLEXIBLE STAYBOLTS, H. V. Wille.

Discussed by Wm. Elmer, W. E. Hall, Alfred Lovell, F. J. Cole.

Friday Afternoon

Trip by boat to Mt. Vernon.

LOCAL COMMITTEE

WALTER A. MCFARLAND, *Chairman*

GUSTAV AYRES
ALBERT H. BUCKLER
CHARLES ELI BURGOON
HOWARD A. COOMBS
JAMES B. DILLARD
WILLIAM A. E. DOYING
CHARLES E. FOSTER
H. A. GILLIS
JAMES HAMILTON
FREDERICK E. HEALY
HERMAN HOLLERITH
J. A. HOLMES
ARTHUR E. JOHNSON
FRANK B. KING
HERVEY S. KNIGHT
WALTER R. METZ
GEORGE L. MORTON
HAROLD P. NORTON
WILLARD L. POLLARD
JOHN E. POWELL
ALFRED H. RAYNAL
WILLIAM B. RIDGELY
W. E. SCHOENBORN
GEORGE R. SIMPSON
CHARLES F. SPONSLER
LUCIEN N. SULLIVAN
WILLIAM B. UPTON
CHARLES V. C. WHEELER
EARL WHEELER

COMMITTEE OF THE WASHINGTON SOCIETY OF ENGINEERS

W. A. McFARLAND, Mem. Am. Soc. M. E., *Chairman*
A. E. JOHNSON, Mem. Am. Soc. M. E.
A. H. RAYNAL, Mem. Am. Soc. M. E.
W. E. SCHOENBORN, Mem. Am. Soc. M. E.
W. B. UPTON, Mem. Am. Soc. M. E.
H. W. FULLER, Mem. Am. Inst. E. E.
JOHN C. HOYT, Mem. Am. Soc. C. E., Secretary Washington Soc. Engrs.
D. S. CARLL, Mem. Am. Soc. C. E., President Washington Soc. Engrs.

CHAIRMAN OF THE LADIES' COMMITTEE, MRS. JAMES LORING LUSK

ACCOUNT OF THE MEETING

The Convention opened on Tuesday evening with a reception in the large assembly hall of the New Willard, followed by dancing, with music by the Marine Band. The reception was largely attended and the occasion was a brilliant one. As the guests arrived they were received by the President and Mrs. Smith, Mrs. W. L. Marshall, Mrs. Charles D. Walcott, and Mrs. F. H. Newell.

D. S. Carll, President of the Washington Society of Engineers, called the assembly to order at 9 o'clock, and extended a hearty welcome to the Society on behalf of its local members and of the Washington Society of Engineers. He then introduced Hon. Henry B. F. Macfarland, President of the Board of Commissioners of the District of Columbia.

Speaking on behalf of these same bodies and of the District of Columbia, Mr. Macfarland referred especially to the work of engineers in the city of Washington, and said in part: There is a particularly warm welcome for the Society in the national capital, since engineers more than the men of any other profession have made it what it is. George Washington, in the year of the birth of the Constitution, conceived the idea of a magnificent capital, then ridiculously out of proportion to the youth, weakness and poverty of the new nation. L'Enfant and Ellicott in the beginning, and a long line of able and brilliant engineers since then, chiefly of the United States Army, have rendered important service in carrying out his plans. The past nine years, the great municipal building period of the city, have been occupied with such engineering feats as the installation of the filtration plant, the sewage disposal system, the new pumping system, the District government railway terminal work, the District government building on Pennsylvania Avenue and its approaches, the Connecticut Avenue bridge, and others of a similar character. Washington appreciates engineers.

President Smith in responding for the Society extended the thanks of the members for this cordial welcome and their appreciation of the interesting program prepared for their pleasure and entertainment by the committees of the Washington Society of Engineers and of the local members.

Business Meeting Wednesday Morning, May 5

The report of the tellers of election was received and there being no objection the President declared the names presented duly elected to membership in the Society. The list follows this report.

Mr. Smith in behalf of the Membership Committee presented the following proposed amendments:

C 10 An Associate shall be 30 years of age or over. He must have been so connected with some branch of engineering, or science, or the arts, or industries, that the Council will consider him qualified to coöperate with engineers in the advancement of professional knowledge. He need not be an engineer.

The committee recommends the following to be added at the end of C 11 of the constitution.

A person who is over 30 years of age cannot enter the Society as a Junior.

The report of the Membership Committee published in Transactions, Vol. 30, p. 550, gave in full the reasons for desiring the change. The proposed amendments were discussed, and in accordance with the rules governing the amendments to the constitution, were referred to the annual meeting for final action.

Prof. Ira H. Woolson, who was a member of the Membership Committee for five years, heartily commended the proposed change and hoped it would become a part of the constitution.

Prof. F. R. Hutton proposed an amendment to C 45, adding "Public Relations Committee" after "House Committee."

In view of the fact that it has been brought to the notice of the Society that a movement is under consideration to increase and improve the facilities for the work of the United States Patent Office, Prof. F. R. Hutton introduced the following resolution:

Resolved, That this Society in convention assembled requests the Council of the Society to consider the desirability of taking some action in furtherance of the movement to increase the Patent Office facilities, and, if deemed advisable, that they request the individual members to take steps to urge their influence to this end upon their Senators and Representatives.

The resolution was voted by the meeting.

Professional Session, Wednesday Morning

Four papers were presented at this morning session, two of which related to the conveying of materials. The first was upon A Unique Belt Conveyor, by Ellis C. Soper, of Detroit, Mich., and described an installation consisting of a conveyor one-quarter mile long, so located on an incline that less power is required to operate it empty than when loaded. Data upon performance were given. The second was upon Automatic Feeders for Handling Material in Bulk by C. Kemble Baldwin, of Chicago, Ill. This contained outline drawings and descriptive matter upon different designs of feeders, to enable the engineer to select the type best suited to his needs.

The third paper was upon A New Transmission Dynamometer, by Prof. Wm. H. Kenerson of Providence, R. I. This is made in the form of a shaft coupling. The apparatus contains an oil chamber, one side of which is a diaphragm, and is so arranged that pressure is brought against this diaphragm directly proportional to the amount of power transmitted. A gage or other registering apparatus is connected with the oil chamber by a small tube which indicates the pressure and the water power transmitted.

The last paper was upon Polishing Metals for Examination with the Microscope by Albert Kingsbury, Pittsburg, Pa., in which he described the use of a polishing machine carrying discs faced with common paraffin and charged with wet abrasives. This produces excellent surfaces on all the harder metals and alloys, but has not proved serviceable upon the soft metals, such as lead.

Wednesday Evening Lecture

On Wednesday evening Frederick H. Newell, director of the U. S. Reclamation Service, was expected to lecture on Home Making in the Arid Regions. As he could not be present a lecture on this subject was given instead by Arthur P. Davis, Chief Engineer of the Reclamation Service.

The United States Reclamation Service in its seven years of existence has undertaken 26 projects situated in 16 different states and territories of the West. It has invested in construction about $40,-000,000. Nineteen projects have been brought to a point where some land is now under irrigation. Water is ready for delivery to about half a million acres. An average of about 10,000 laborers are employed on this work, and over 55,000,000 cu. yd. of rock and earth have been excavated. Over 2000 miles of canals have been built and 56 tunnels have been bored, which have a total length of over 13 miles.

Twelve large earthern dams and one high masonry dam have been completed, and two other masonry dams which will rank among the highest dams in the world are in an advanced stage of construction. Many of these projects are in remote localities into which roads had to be built, some of which were carved in precipitous rock, or tunneled through mountains. In the aggregate 342 miles of roads and 793 bridges have been constructed.

In some localities, especially on the Pacific slope, the mild climate, and the nearly perpetual sunshine, produce remarkable results in the growth of fruits, which for color, flavor and physical perfection cannot be equaled in a more humid climate. The chemical force in sunshine and a perfectly regulated water supply are also evident in the yields of vegetables and forage crops.

The lecture was illustrated by many beautiful slides.

Gas Power Section

At this session, F. R. Low, Chairman of the Gas Power Section, presided, and Geo. A. Orrok acted as Secretary. Previous to the reading of the professional papers reports were received from the committees.

Membership Committee: The report showed a total membership of 302, of which 177 were members of The American Society of Mechanical Engineers and 125 were affiliates. The Membership Committee is thoroughly organized with representatives in different cities.

Literature Committee: Prof. C. H. Benjamin gave a verbal report of this committee stating that the committee is organized for work and had laid out a tentative program. It was hoped to index the books on the subject of gas power and articles in periodicals dealing with gas power and allied subjects; also to present reviews of new books and abstracts of important articles. There would be two fields for work: one, a permanent one, and the other in the line of current work relating to popular reviews and abstracts for the benefit of members.

PLANT OPERATIONS COMMITTEE: A verbal report offered by Irving E. Moultrop reported progress and stated that standard forms for obtaining operating data on gas power plants were in preparation. The committee has a large membership and is widely scattered so that it had been impossible to arrange a meeting, but the work had been advanced as far as possible by correspondence.

Mr. C. L. Straub presented a report on gas-producer development abroad, an abstract of which appears as part of his paper on Marine Producer Gas Power, included in this volume.

Mr. Orrok stated with reference to the work of committees that it is conducted with the idea that as the Gas Power Section has been formed while the art is young it will be possible to place a record of its development on file at the headquarters of the Society. Such data in connection with the large library will place at the disposal of anyone interested in the industry the available information upon the subject of gas power.

Following the presentation of the reports came the professional papers, the first of which was on Marine Producer Gas Power, by C. L. Straub of New York. This paper explained the conditions opposing the earlier adoption of producer gas power for marine service and gave a summary of marine gas power plants in operation at present. It compared the updraft and downdraft of producer gas apparatus and contained comparative drawings of the steam equipment and two types of producer gas equipment for a 306-ft. boat.

The next paper was upon The Operation of a Small Producer Gas Power Plant, by C. W. Obert of New York. It presented a general description of a producer gas power plant in the Westchester market building of Swift & Company, Bronx Borough, New York. The author outlined the operating and maintenance system developed for keeping producers and engines in proper condition for continuous operation.

A paper was presented upon A Method of Improving the Efficiency of Gas Engines, by Thomas E. Butterfield of Philadelphia, Pa. It related to the securing of higher efficiency by reducing the clearance and increasing the compression and referred especially to a method of diluting with an inert gas the charge drawn in during the suction stroke of an Otto cycle engine. By this means premature ignition and other troubles incident to high compression are avoided.

The last paper of the session was upon Offsetting Cylinders in Single-Acting Engines by Prof. T. M. Phetteplace of Providence,

R. I. It gave the results of an investigation of this subject in which the various factors entering into the problem were taken into account.

Presentation of Portrait of Rear-Admiral Melville

On Thursday evening, a portrait of Rear-Admiral Geo. W. Melville was presented to the National Gallery at a ceremony held in the auditorium of the New Willard. President Smith presided over the large audience which assembled and an address was made by Rear-Admiral Melville on The Engineer in the Navy. Mr. Walter M. McFarland of Pittsburg, Pa., gave an appreciation of Melville as a man and of his work for the nation and profession. The portrait was accepted for the Nation by Dr. Chas. D. Walcott, Secretary of the Smithsonian Institution.

At the conclusion of the ceremony, President Smith asked that Mr. Sigismond de Ivanowski, the Russian artist who had produced so admirable a likeness of Melville, be escorted to the platform. Mr. de Ivanowski told briefly and simply of his attempt to portray the strong characteristics of his subject and displayed evident pleasure that his efforts were so warmly appreciated.

Abstracts of the addresses are published with the professional papers in this volume.

Professional Session, Friday Morning

Five papers, and a continuation of the Safety Valve discussion given at the February meeting in New York, were scheduled for this session. The first paper was upon Small Steam Turbines by Geo. A. Orrok of New York. The various types of turbines now on the market were illustrated and described and a number of steam consumption curves were given to demonstrate the economy that might be expected from machines of this type.

A paper on Compressed Air Pumping Systems of Oil Wells by Edmund M. Ivens of New Orleans, La., was read, in which a description was given of compressed air plants at Evangeline, La., oil fields, and the results of tests upon these plants with different types of apparatus.

This was followed by the Safety Valve discussion, continued from the February meeting in New York.

Two papers were next presented upon the properties of steam: one by Prof. C. H. Peabody, of Boston, Mass., on Specific Volume of

Saturated Steam, and the other upon Some Properties of Steam by Prof. R. C. H. Heck of New Brunswick, N. J. The former reviewed the results of experiments which might form the basis of a computation of specific volumes at various temperatures and compared the computed results with experimental determinations of the same quantities. The latter paper summarized the important work of Holborn and Henning and compared the results of other investigators. These two papers constituted another step ahead in the work that is now being accomplished toward securing accurate information upon the properties of both saturated and superheated steam.

The last paper was by H. V. Wille, Philadelphia, Pa., on A New Departure in Flexible Staybolts. This paper proposed the employment of tempered spring steel in the manufacture of the stems of staybolts, the ends being of soft steel so as to permit riveting over in the boiler.

This session closed with a unanimous resolution extending the thanks of the Society to those who had afforded such abundant entertainment to their visitors.

Entertainment

During the convention an information bureau was conducted at the Society headquarters by Chairman Walter A. McFarland of the Local Committee, where the various excursions were organized. These included trips not only to the public buildings, but to government institutions and other points of technical interest, among which were the Bureau of Standards, the station of the Potomac Electric Power Co., the Union Railway Terminal, the Naval Gun Factory, the District pumping stations, etc.

At the ladies' headquarters in the New Willard, tea was served each day from four to six o'clock and the visiting ladies as well as many members of the Society, accepted the hospitality extended by the ladies at this time. Sight-seeing automobile trips for the ladies were also arranged on each day, which were largely patronized and greatly enjoyed.

On Wednesday afternoon a large number took the trip to Fort Myer to see the exhibition drill, and the evolutions performed by the several troops, the unusual skill of both riders and drivers and the thoroughly trained horses, called forth round after round of applause. Two battalions of artillery with guns and caissons went through evolutions of great complexity, and two troops of cavalry through various formations, apparently equally difficult, while a troop of bareback riders captivated the audience by their horsemanship.

On Thursday afternoon the reception of members and guests by President Taft in the East room of the White House was very generally attended.

On Friday afternoon following the professional session, many went by boat to Mt. Vernon to visit the estate and home of Washington, and a wreath from the Society was placed on his grave.

A trip to Fort Myer was also made and the dirigible balloon of the Signal Corps of the United States Army inspected.

ELECTIONS TO MEMBERSHIP

The following were declared elected to membership in the Society upon the ballot of May 5, 1909, and their election reported at the Washington Meeting:

MEMBERS

Ahlquist, H., Syracuse, N. Y.
Babbitt, Edward F., Columbus, O.
Behrend, Bernard A., Milwaukee, Wis.
Billings, A. W. K., Havana, Cuba.
Blaisdell, Benjamin H., Manila, P. I.
Bloemeke, R. B., New York.
Borde, G. U., New Orleans, La.
Bruckner, R. E., New York.
Burt, Clayton R., Rockford, Ill.
Bushnell, Douglas Stewart, New York.
Crockard, Frank H., Birmingham, Ala.
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Garvin, George K., New York.
Hazelton, W. S., Detroit, Mich.
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Hogue, Oliver Driscoll, Boston, Mass.
Hunter, John A., Pittsburgh, Pa.
Jewett, A. C., Orono, Me.
Johnstone, F. W., Mexico City, Mexico.
Jones, Walter J., New York.
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Kranz, William George, Sharon, Pa.
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McGlone, R. G., Galveston, Texas.
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Stanton, F. A. O'C., Hoboken, N. J.
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Thompson, Edward C., Boston, Mass.
Thurston, Edward D., Jr., New York.
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Whitcomb, L. A., Brooklyn, N. Y.
Whiting, R. A., New York.
Wiley, J. M., New York.
Woolley, Harold O., Dansville, N. Y.
Wyman, A. H., Milwaukee, Wis.

By direction of the Council announcement is also made of the election of H. K. Hathaway of Philadelphia, Pa., elected on the ballot of July 25, 1907, as an Associate Member, but announcement of whose election has not previously been made.

No. 1230

THE TRANSMISSION OF POWER BY LEATHER BELTING

CONCLUSIONS BASED PRINCIPALLY ON THE EXPERIMENTS OF LEWIS AND BANCROFT

By Carl G. Barth, Philadelphia, Pa.
Member of the Society

In his paper, Slide Rules in the Machine Shop as a Part of the Taylor System of Management, read December 1903, the writer referred to an improved theory and new formulæ developed by him for the pulling power of belting, which had been applied in connection with the slide rules described. He also stated his expectation of presenting his theory and conclusions to the Society at some future time.

2 These conclusions have since been successfully applied in practice by the extensive daily use of these slide rules in task-setting for machine operations, and the present paper was prepared with the general view of submitting this theory to the Society for the criticism and consideration of members who are interested in this subject, and with the special view of supplementing Mr. Taylor's paper, On the Art of Cutting Metals. All the experimental and mathematical data for the slide rules were presented in his paper, excepting the data upon the pulling power of belting, an important element in these slide rules when applied to belt-driven machines.

3 The theory to be presented is only an additional attempt, and it is hoped a fairly successful one, to do something along lines repeatedly touched by various other investigators among the members of the Society, and the writer is glad to acknowledge his indebtedness to nearly all of these, as his work has principally consisted in taking advantage either of carefully conducted experiments recorded by them, or of suggestions of various kinds that have stimulated his interest and set him thinking.

Presented at the New York monthly meeting (January 1909) of The American Society of Mechanical Engineers.

4 Most notable is the paper presented by Mr. Wilfred Lewis at the Chicago meeting in 1886, in which he recorded a series of experiments conducted by himself in the spring of 1885, under the direction of Mr. J. Sellers Bancroft and at the works of William Sellers & Company, Philadelphia. In his paper was shown for the first time the fallacy of the then universal and still common assumption, that the sum of the two tensions in a belt is constant for all loads. It was also shown that the coefficient of friction between a belt and its pulley is considerably higher than was commonly assumed for ordinary working conditions, and that this coefficient varies greatly with the velocity of slip, a fact that has also been pointed out by other investigators.

5 Mr. Lewis did not, however, even attempt to develop either empirical or rational mathematical formulæ to represent the facts that he established, though his experiments, as will subsequently appear, contained all that was necessary for a complete mathematical exposition of the subject.

6 An attempt at an empirical mathematical exposition of the relations between the two tensions in a belt in accordance with the facts established in Mr. Lewis' paper, was later made by Prof. Wm. S. Aldrich in a paper read at the New York meeting in 1898. Mr. Aldrich made an original layout of a great number of Mr. Lewis' experiments in a manner that seemed to the writer to indicate an excellent way to investigate the subject, and which resulted in a discussion of Mr. Aldrich's paper, the substance of which has formed the basis for all of the writer's subsequent work on the subject.

7 But while the writer's complete theory could have been worked out without it, its practical application to the running of belt-driven machine tools could never have been made in the present satisfactory manner without the facts made known by Mr. Taylor in his paper, Notes on Belting, read at the New York meeting in 1893.

8 In this paper Mr. Taylor showed the economy of running belts under much lower tensions than those commonly used, and that the ultimate strength of a belt, or rather of the joint in a belt, does not form a proper basis for the working tension of a belt, since a belt will not long retain a tension that is even a small fraction only of its ultimate strength (see Fig. 4). However, Mr. Taylor's facts and figures were derived from comparatively slow-running belts, and he gave nothing that could be directly applied to the higher and more economical belt speeds. The modification and extension of Mr. Taylor's ideas to include high speed belts have therefore been part of the writer's personal work also.

9 A summary of the writer's work on this subject follows:

a To establish a mathematical formula for the relation between the tension in a belt and the stretch due to this tension, based on experiments made at different times by Mr. Wilfred Lewis, Prof. W. W. Bird and himself. See Fig. 1, 2 and 3 and Par. 1–18 of the Appendix.

b By means of the knowledge of the elastic properties of leather belting expressed by this formula to develop a formula for the relations between the tensions in the two strands of a belt transmitting power, which formula takes account of the influence of the sag in a horizontal belt, and agrees substantially with the results of the experiments made by Mr. Lewis, when plotted in the manner first done by Professor Aldrich. See Fig. 6 and Par. 19–38 of the Appendix.

c To establish a formula to express the relation between the coefficient of friction between a belt and a cast iron pulley, and the velocity with which the belt slips or slides over the pulley, as revealed by plotting the results likewise obtained by Mr. Lewis. See Fig. 7 and Par. 48 of the Appendix.

d The construction of a diagram embodying the formula expressing the relation between the two tensions in a belt, the well known formula for the loss in effective tension due to centrifugal force and the likewise well known formula for the ratio between the effective parts of the two tensions, as determined by the coefficient of friction and the arc of contact of the belt on its pulleys. These formulæ are so correlated on the diagram that problems dealing with the contained variables may be solved graphically, while a direct algebraic solution is possible only for a vertical belt, or what is the same thing, by neglecting the effect of the sag of a horizontal belt. See Plate 1 and Par. 11–24.

e Also, by means of the better knowledge gained of the elastic properties of leather belting, to develop a formula for the creep of a belt on its pulleys due to the difference in the tensions in the two strands, along the lines outlined by Professor Bird in his paper on Belt Creep, read at the Scranton meeting in 1905. See Par. 41–44 of the Appendix.

f The construction of diagrams showing the pulling-power and other relations of the two tensions of a belt of 1 sq. in. cross section and 180 deg. arc of contact at different speeds, under certain conditions and assumptions recommended by the writer. See Fig. 1, 2 and 3, and Par. 38–52. Also a modification of these diagrams for extended practical use, on which may be read off: (1) The pulling power of a belt of any width and thickness and any arc of contact between 140 and 180 deg.; (2) The initial tensions below which the belt must not be allowed to fall in order to confine the slip and the consequent loss of efficiency of transmission within certain limits; (3) The initial tension to which it is recommended that the belt be re-tightened after falling to this minimum limit. See Plate 2 and Par. 53–66.

g Finally, the construction of a slide rule serving the same purpose as the diagram just mentioned, but which is much handier than the diagram. See Fig. 5.

10 With these statements the explanation of the diagram Plate 1 will now be taken up.

DESCRIPTION AND USE OF THE DIAGRAM PLATE 1

11 Taking the extreme left-hand bottom corner point as the origin distances along the bottom line represent the variable tension in the tight strand or side of a belt in terms of the initial tension, while vertical distances measured to any of the bottom group of curves in the middle field of the diagram represent the corresponding tension in the slack side of the belt, also in terms of the initial tension.

12 The particular curve against which to read off a certain tension depends on the center distance of the pulleys of the belt in connection with its initial tension per square inch, and is found by consulting the small diagram directly to the right of this group of curves, in the following manner:

13 Read off the center distance c al6ng the extreme right side of this diagram, then follow along the diagonal to the left from this reading of c until it intersects the vertical line that extends up from the reading of the initial tension t_0 on the base of the diagram.

14 From this point of intersection of c and t_0 go horizontally to the left to the reading of the corresponding value of the ratio

$\frac{c^3}{t_0^{2.5}}$, which leads directly to the proper curve in the bottom group of curves in the middle section of the diagram.

15 Against this curve there can now be read off any simultaneous tensions in the two strands of the belt corresponding to these particular values of c and t_0 of the belt under consideration.

16 Having noted this curve, and turning to the extreme right hand section of the diagram, the ratio of the effective tensions in the two sides of the belt corresponding to the particular coefficient of friction ϕ, and the particular arc of contact α which we wish to count on, may be determined.

17 To this end we read off the arc of contact in degrees on the extreme right-hand side of this section of the diagram, follow this reading horizontally to the left until it intersects that line radiating from the bottom left corner of this section which is marked with the value assumed for ϕ at its termination in the extreme top line of the section, and then from this point of intersection go vertically up or down as the case may be, until we meet the single curve drawn in this section of the diagram. From this point in the curve we now go horizontally to the extreme left side of this section of the diagram and there read off the value of the ratio of the effective tensions, which is

$$\frac{t_2}{t_1} = \frac{1}{\epsilon^{\phi\alpha}}$$

for a belt running so slowly that the centrifugal force has no perceptible influence, and equal to

$$\frac{t_2 - t_c}{t_1 - t_c} = \frac{1}{\epsilon^{\phi\alpha}}$$

when the centrifugal force reduces the total tensions to the effective tensions $t_2 - t_c$ and $t_1 - t_c$ respectively.

18 From the point representing $\frac{1}{\epsilon^{\phi\alpha}}$ we now draw a line to the extreme left bottom corner point of the whole diagram.

19 Any two simultaneous coördinates to this slant line counted from its extremity in the base line of the diagram, will then also be in the ratio $\frac{1}{\epsilon^{\phi\alpha}}$.

20 Passing now to the extreme slanting left side of the diagram, we read off the velocity V of the belt, follow this reading horizontally to the right until we intersect that radiating line from the extreme left bottom corner point of the diagram which is marked with the initial tension t_0 of the belt per square inch, where it terminates, either in the extreme top line of this section of the diagram, or against its extreme right side.

21 From this point of intersection we now go down vertically until we reach the long 45-deg. diagonal of the diagram, on which we then read off the ratio $\frac{t_c}{t_0}$, or the loss in effective tension in terms of the initial tension, due to the centrifugal force in the belt.

22 From this point on the long 45-deg. diagonal of the diagram we now finally draw a line parallel to the line previously drawn to represent the ratio $\frac{1}{e^{\phi \alpha}}$ and extend it to intersect the curve first of all determined to represent all possible relations between the two tensions in the belt.

23 The coördinates of this point then give the two tensions in the belt in terms of its initial tension. By extending the ordinate up to intersect that curve in the middle group of curves which is marked with the same value of $\frac{c^2}{t_0^{2.5}}$ at its terminal in the right side of this middle section of the diagram as the curve dealt with in the bottom group of curves, we read off the difference of the two tensions; that is, the effective pull of the belt, in terms of the initial tension.

24 By likewise extending the ordinate all the way up to meet that curve in the top group of curves which is marked the same as the other two curves, we may also read off the sum of the two tensions, in terms of the initial tension.

25 *Example.* A belt of the pulley center-distance $c = 200$ in. and of $2\frac{1}{2}$ sq. in. cross section, has an initial tension $T_0 = 175$ lb. and runs at a velocity $V = 2000$ ft. per minute. The arc of contact of the belt on each pulley may be taken as 180 deg., and the coefficient of friction ϕ as 0.5. What will be the centrifugal tension $T_c = 2.5\ t_c$ in the belt, what the tension $T_1 = 2.5\ t_1$ in the tight side, and what the tension $T_2 = 2.5\ t_2$ in the slack side? Also, what will be the effective pull $P = T_1 - T_2$ and what the sum $T_1 + T_2$ of the two tensions?

26 *Solution.* The steps have been indicated on the diagram by little circles around the points on the several scales of variables

FOLDER No. 1

that correspond to the values of these variables in the example. Thus, in the small diagram at the bottom of the chart, between the other two diagrams, circles have been drawn around the points indicating the unit initial tension

$$t_0 = \frac{175}{2.5} = 70$$

and the center distance 200, giving the resulting value

$$\frac{c^2}{t_0^{2.5}} = \frac{200^2}{70^{2.5}} = 1, \text{ approximately.}$$

which determines the particular curve in each of the three groups of curves in the middle section of the diagram which apply to the belt under consideration.

27 Also, in the right-hand section of the diagram circles have been drawn indicating the coefficient of friction $\phi = 0.5$, and the arc of contact 180 deg., giving the resulting value of the ratio between the effective tensions $\frac{1}{e^{\phi\alpha}} = 0.208$.

28 From the point $\frac{1}{e^{\phi\alpha}} = 0.208$ has also been drawn a line to the extreme left bottom corner of the whole diagram, the ratio of any two coördinates to this line thus being 0.208 also.

29 Again, in the left-hand section of the diagram circles are drawn about points indicating a velocity $V = 2000$ ft. and the initial tension $t_0 = 70$, leading to a resultant value of

$$\frac{t_c}{t_0} = \frac{T_c}{T_0} = 0.202$$

This means that $0.202 \times 175 = 35.35$ of the total 175 lb. of initial tension in the belt, is made ineffective by the centrifugal force due to $V = 2000$ ft.

30 From the point $\frac{t_c}{t_0} = 0.202$ has also been drawn a line parallel to the line expressing the ratio

$$\frac{1}{e^{\phi\alpha}} = 0.208$$

so that the inclination of this line expresses the same ratio between the effective belt tensions; or

$$\frac{T_2 - T_c}{T_1 - T_c} = \frac{t_2 - t_c}{t_1 - t_c} = 0.208$$

31 The point of intersection of this line with the curve previously found to express all possible relations between the working tensions in the two strands, has also been encircled, and a vertical line has been drawn through this point upwards until it intersects that curve in the top group of curves which is marked $\frac{c^2}{t_0^{2.5}} = 1$, and which accordingly expresses all possible values of the sum of the two tensions in the belt under consideration.

32 The intersection of this vertical line with that curve in the middle group which is likewise marked $\frac{c^2}{t_0^{2.5}} = 1$, and which accordingly expresses all possible values of the difference of the two tensions, has also been encircled.

33 Taking the readings of the point encircled in the bottom curve, we find

$$\delta_1 = \frac{T_1}{T_0} = 1.81, \text{ and } \delta_2 = \frac{T_2}{T_0} = 0.535$$

We therefore get

$$T_1 = 1.81 \times 175 = 316.75 \text{ lb. and}$$
$$T_2 = 0.535 \times 175 = 93.63 \text{ lb.}$$

34 From this we again get

$$P = T_1 - T_2 = 316.75 - 93.63 = 223.12 \text{ lb.}$$

as the effective pull of the belt. Also, $T_1 + T_2 = 316.75 + 93.63 = 410.38$ lb. as the sum of the tensions, as against $175 \times 2 = 350$ lb., the initial sum.

35 But usually we would not be interested in the separate values of the tensions, and then we would read off directly by the encircled point in the middle group of curves,

$$\frac{p}{t_0} = \frac{P}{T_0} = \frac{T_1 - T_2}{T_0} = 1.275$$

which gives $P = 1.275 \times 175 = 223.12$ lb., the same answer as above.

36 If also interested in the sum of the tensions, we would read this off directly by the point encircled in the top group of curves,

$$\frac{t_1 + t_2}{t_0} = \frac{T_1 + T_2}{T_0} = 2.345$$

which gives $T_1 + T_2 = 2.345 \times 175 = 410.38$ lb., the same answer as above.

37 The solution of problems involving long horizontal belts is thus readily enough effected by means of this diagram, but a little consideration will also make it evident that the difference in results obtained by taking the length of a belt into account and by neglecting the same is but slight, except for greater lengths and lower initial tensions than are usually employed in practice. Ordinarily, therefore, we would use merely the very bottom curves in the bottom and top groups, and the very top curve in the middle group of curves in the middle section of the diagram, which curves are all marked

$$\frac{c^2}{t_0^{2.5}} = 0$$

and thus make no difference between horizontal and vertical belts except for exceedingly long belts.

DESCRIPTION OF DIAGRAMS, FIG. 1, 2, and 3

38 We will now take up the consideration of diagrams Fig. 1, 2, and 3, which form the basis of the large diagram Plate 2. These diagrams are worked out theoretically for vertical belts only, but may be applied without hesitation to horizontal belts of the lengths usually met with in practice.

39 In his paper Notes on Belting, Mr. Taylor showed, as already mentioned, the economy of running belts under much lower tensions than those commonly figured on in proportioning a belt to do a certain amount of work.

40 He also divided the belts with which he dealt into two classes: those whose dimensions he could not well increase over what he found in use, such, for instance, as the cone belts on lathes and other machines provided with a cone pulley in a limited space; and those he could

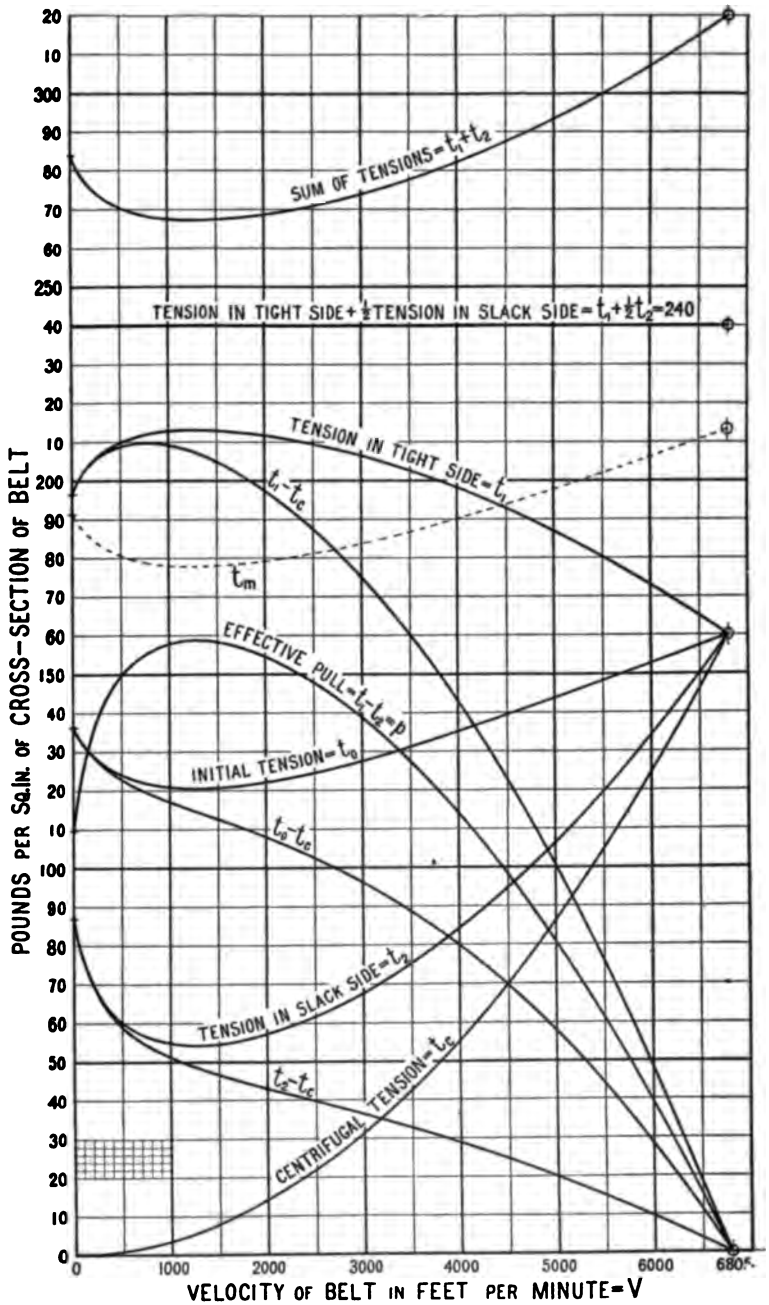

FIG. 1 DIAGRAM SHOWING THE RELATIONS OF PULLING POWER TO TENSIONS AT ALL SPEEDS, FOR A MACHINE BELT OF 1 SQ. IN. SECTION. (SEE OPPOSITE PAGE AND PAR. 9*f*).

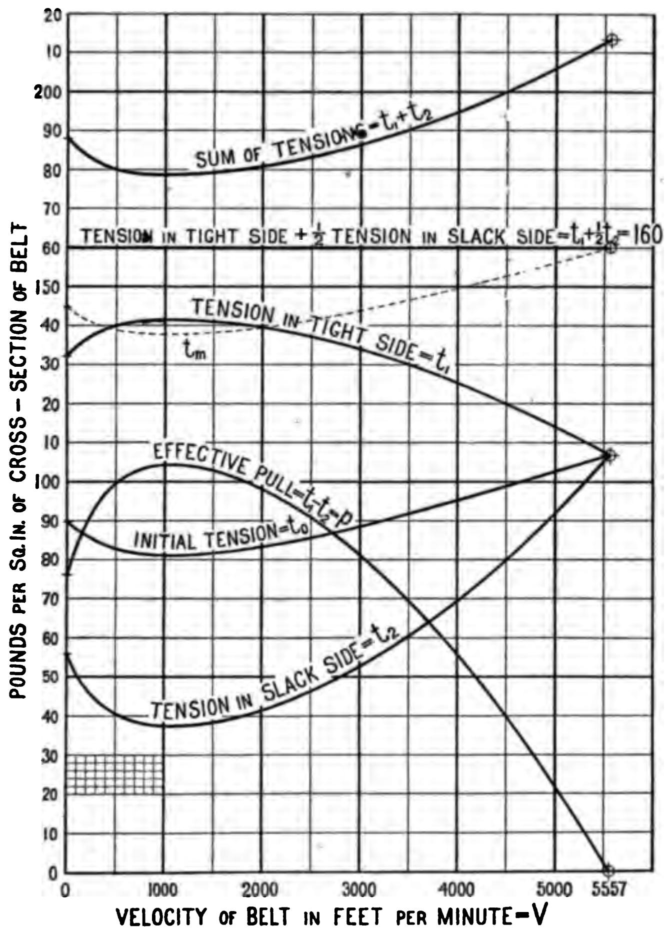

FIG. 2 DIAGRAM SHOWING THE RELATIONS OF PULLING POWER TO TENSIONS AT ALL SPEEDS FOR A COUNTERSHAFT BELT OF 1 SQ. IN. SECTION

FIG. 1 is plotted for: Arc of contact $\alpha = 180$ deg., coefficient of friction $\phi = 0.54 - \frac{140}{500 + V}$ (See Par. 51); and the sum of the tension in the tight side of the belt and one-half the tension in the slack side, $A = 240$ lb. for all velocities. (See Par. 44.) The dotted curve marked t_m gives the initial tension which for p, the same as that figured for $A = 240$ lb., corresponds to $A = 320$ lb. This t_m, the average value of which is about 185 lb. between practical limits of velocities, is the maximum initial tension, to which a machine belt is retightened whenever the tension falls to the minimum initial tension t_0.

FIG. 2 is plotted for the same data as Fig. 1, except that here $A = 160$ lb. for the full-drawn curves. (See Par. 44.) The dotted curve marked t_m gives the initial tension which, for p the same as that figured for $A = 160$ lb., corresponds to $A = 240$ lb. This t_m, the average value of which is about 142 lb. between practical limits of velocities, is the maximum initial tension to which a countershaft belt is retightened whenever the tension falls to the minimum initial tension t_0.

readily increase by providing larger pulleys, such as the belts leading from line-shafts to the counter-shafts of machines.

41 For belts in the first class he adopted higher tensions than for those in the second class. He also devised a set of belt-clamps provided with spring balances, by means of which he could make sure that a belt was put up under a specified initial tension, and could also ascertain its fall in tension at any time desired.

42 All this seems to be the first effort made by an engineer to pay any systematic attention to the belting in a shop, which even today is usually left entirely to the rule-of-thumb method of the machinist or millwright.

43 The reason why Mr. Taylor had adopted, and accordingly recommended, lower belt-tensions than usually counted on in proportioning a belt to do a certain amount of work, was that a belt quickly loses its tension if it exceeds a certain amount, and thus in order to maintain such a tension, approximately, requires frequent retightening, which is a source of too much expense and leads to a rapid destruction of the belt. See Fig. 4 and Par. 8.

44 Taking Mr. Taylor's data as a starting point, the writer has finally adopted the rule, as a basis for his use of belts on belt-driven machines, that for the driving belt of a machine the *minimum initial tension* must be such that when the belt is doing the maximum amount of work intended, the *sum of the tension on the tight side of the belt and one-half the tension on the slack side will equal* 240 *lb. per square inch of cross-section for all belt speeds*; and that for a belt driving a countershaft, or any other belt inconvenient to get at for retightening or more readily made of liberal dimensions, this sum will equal *160 lb.*

45 Further, the maximum initial tension, that is, the initial tension under which a belt is to be put up in the first place, and to which it is to be retightened as often as it drops to the minimum, must be such that the sum defined above is *320 lb.* for a machine belt, and *240 lb.* for a counter-shaft belt or a belt similarly circumstanced.

46 The reason for adopting a uniform sum of the tension in the tight side and one-half the tension in the slack side, as mentioned above, instead of either a uniform initial tension, or a uniform maximum tension alone, is that the aim has been to get as uniform periods as possible for the retightening of belts at all speeds.

47 But evidently, while the maximum tension in a belt must be the greatest factor in determining the rapidity with which the belt will lose its tension as a whole, the accompanying tension in the slack

strand or side must also have some influence, though not proportionally to the same extent; and hence, the idea occurred to the writer of taking it into account in the manner and to the extent stated.

48 On the diagram Fig. 1, various formulae have been plotted for 240 lb. as the constant sum at all speeds of the tight tension and one-half the slack tension per square inch cross section of belt; for a coefficient of friction that varies with the velocity according to Formula 13 in the Appendix; and an arc of contact of 180 deg. The relations of the various tensions in the belt for all speeds may there be studied to great advantage. It will thus be seen that the centrifugal tension completely balances the initial tension at a belt speed of 6805 ft. per minute.

49 On the diagram in Fig. 2 some of these formulae have likewise been plotted, with the lesser value of 160 lb. as the constant sum of the tight tension and one-half the slack tension, but for the same values of the coefficient of friction and the arc of contact. Here the centrifugal tension balances the initial tension at the speed of 5557 ft. per minute.

50 The diagram, Fig. 1, represents the writer's practice in connection with machine belts; that in Fig. 2 his practice in connection with counter-shaft belts (see Par. 44). Both diagrams were used as the basis for the construction of Plate 2, and for the slide rule illustrated in Fig. 5.

51 In the diagram Fig. 3 are given the horse power outputs per square inch of section of belts running under the conditions imposed in the diagrams Fig. 1 and 2.

52 It will be seen that the maximum output is 13.8 h.p. per square inch of a belt under the conditions imposed in Fig. 1, and that this is for a speed of about 4000 ft. (more exactly 4032 ft.); and that for the conditions imposed in Fig. 2, the maximum horse power is 7.46 per square inch, and that this is for a speed of about 3250 ft. (more exactly 3247 ft.).

DESCRIPTION OF DIAGRAM PLATE 2

53 In the diagram Plate 2, the data given on the diagrams Fig. 1, 2, and 3, for a belt of one square inch of section, and an arc of contact of 180 deg., have been so modified that almost any problem relating to belting of any size and any arc of contact can be solved.

54 This will best be illustrated by the following two examples:

55 *Example 1.* The maximum cone step on the counter-shaft of

a lathe is 22 in. in diameter and wide enough to carry a 3 in. double belt. The speed of the shaft is to be 300 r.p.m. Assuming the thickness of a 3 in. double belt to be $\frac{5}{16}$ in., and the arc of contact of the belt to be 170 deg.: (*a*) What pull can the belt be counted on to exert, and what horse power will it transmit with this pull? (*b*) Under what initial tension will the belt first be put up, and retightened

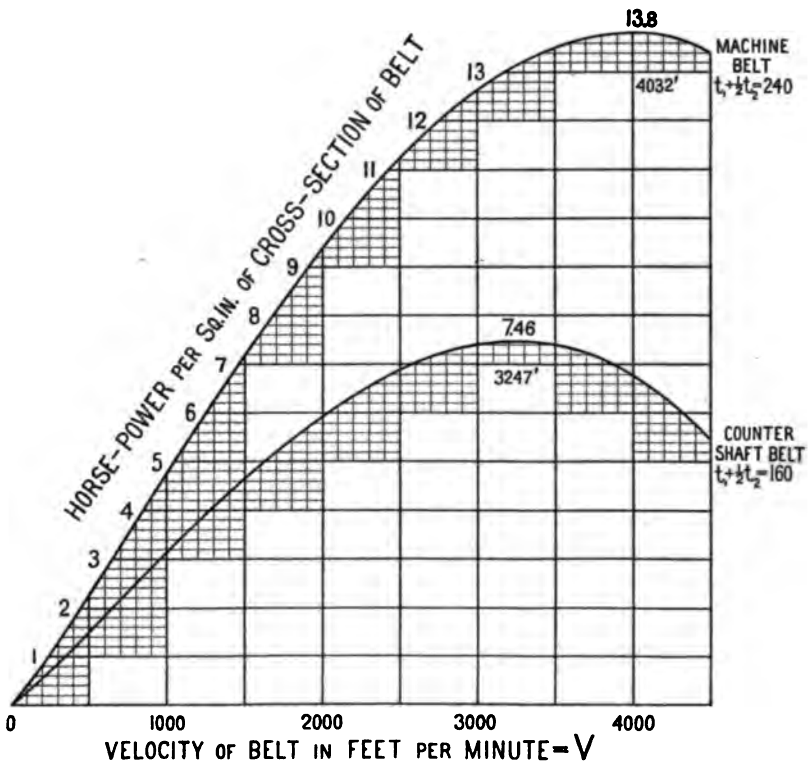

FIG. 3 HORSEPOWER OUTPUT CORRESPONDING TO BELT PULLS IN FIG. 1 AND 2

from time to time? (*c*) And what minimum initial tension must it not be allowed to fall below to insure the above-determined pull without undue slip?

56 *Solution.* To get the answer to question (*a*), we first turn to the small bottom portion of the diagram Plate 2, and on its right hand side note the point reading 300 r.p.m. From this we pass horizontally to the left until we intersect the vertical line from the point

reading 22 in. on the scale of pulley diameters at the bottom line of the diagram. From the point of intersection we follow the diagonal

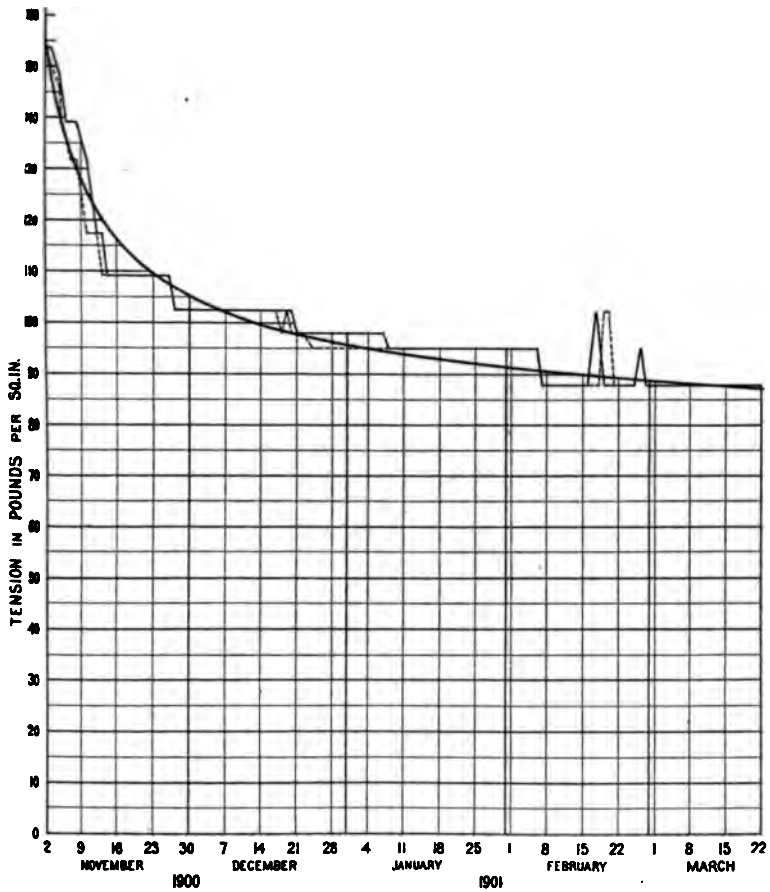

FIG. 4 EXPERIMENTS MADE ON THE FALL IN TENSION IN TWO BELTS $5\frac{3}{4}$ IN. WIDE BY $\frac{15}{32}$ IN. THICK DRIVING A LARGE ROTARY PLANER AT THE WORKS OF THE BETHLEHEM STEEL COMPANY (SEE PAR. 8 AND PAR. 43)

The peculiarly high tensions measured on four days, during the latter part of February 1901, were probably due to something sticking in the belt scales used.

line upwards to the bottom line of the main portion of the diagram, and there read the velocity of the belt to be about 1700 ft. per min.

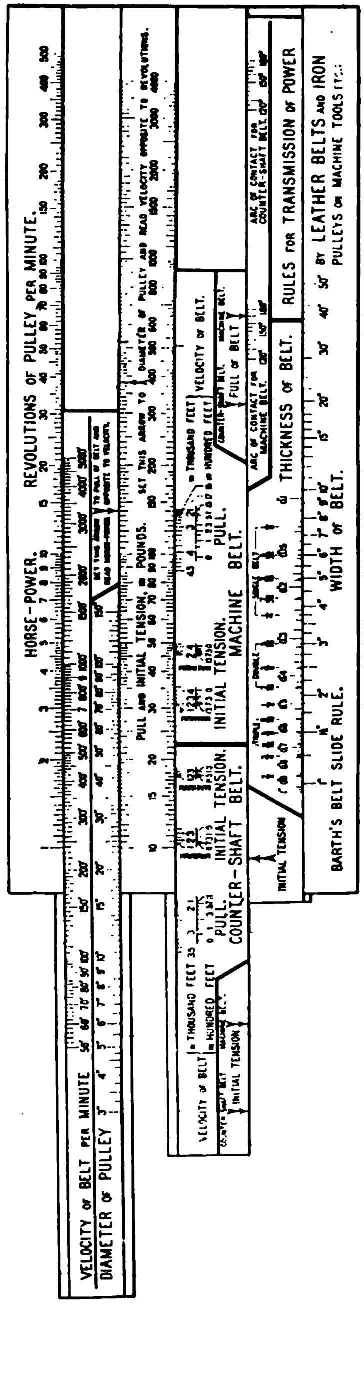

Fig. 5 Slide Rule for the More Convenient Solution of Belt Problems Solved by the Diagram Plate 2, Based on Fig. 1 and 2

FOLDER No. 2

57 We now note the point that corresponds to this belt speed of 1700 ft. per min. in that scale on this same bottom line of the main diagram which is marked "Velocity for Pull of Machine Belt" and interpolate a vertical line extending upwards from this point. Then leaving this for the time being, we turn to the extreme left hand portion of the diagram and there note the point corresponding to the belt thickness $\frac{5}{16}$ in. on the vertical scale to the extreme left, and also the point marked 3 in. on the scale of belt widths at the bottom of the diagram. Following these two points, respectively, horizontally to the right and vertically upwards, until they intersect, we now follow the diagonal from this point of intersection until it terminates against the vertical line marked 170 deg. at the top of the diagram, in the field marked "Arc of Contact," and then continue horizontally until we intersect the interpolated vertical line for the belt speed 1700 ft. already noted.

58 From the point of intersection we follow the diagonal until we meet the vertical scale of pounds, on which we now read the belt pull to be 140 lb.; and continuing horizontally until we meet the vertical line extending upwards from the point corresponding to the belt speed originally found on the scale of belt speeds in this section of the diagram, and from this line diagonally to the vertical scale of horse power, we read off the horse power transmitted to be 7.2. All of these movements are indicated on the diagram by little circles around the various points of intersection.

59 To get the answer to question (*b*), we proceed exactly as before, with the width and thickness of the belt, except that we follow the diagonal across the portion of the diagram headed "Arc of Contact" until we meet the border line for 180 deg. From here on we proceed horizontally until we meet the vertical line that corresponds to the belt speed 1700 ft. in the field marked "Velocity for Maximum Initial Tension for Machine Belt." From the point of intersection on this vertical line we then pass diagonally to the scale of pounds, and there read the maximum initial tension to be 168 lb. Those movements for this solution on the diagram that differ from those for the answer to question (*a*), are indicated by little filled-in circles around the various points of intersection noted.

60 For the answer to question (*c*), we proceed in every respect as we did for question (*b*), except that we of course start from the point corresponding to the belt speed 1700 ft. in that field of the scale on the top line of the diagram which is marked "Velocity for Minimum Initial Tension for Machine Belt." The answer read off on

the vertical scale of pounds is 113 lb. The movements for this solution on the diagram that differ from those for the answer to question (*b*), are indicated by little dotted circles around the points of intersection.

61 *Example 2.* The counter-shaft in Example 1 is to be driven by a belt to run at a speed of 3000 ft. per min. (*a*) What diameter of pulley is required to give this belt speed? (*b*) What pull must the belt transmit? (*c*) What width of double belt must be used? (*d*) And what will be the initial tension under which the belt must be put up, and to which it must be again retightened after falling to the minimum? (*e*) What will be its minimum tension?

62 *Solution.* (*a*) As the counter-shaft is to make 300 r.p.m. and the belt is to run at 3000 ft. per min., we turn to the small diagram at the right hand bottom corner of the main diagram, proceed horizontally to the left from the point marked 300 on the scale of revolutions on the right, until we meet the diagonal line from the point marked 3000 on the horizontal scale of velocities. From the point of intersection we then go vertically down to the scale of pulley diameters, and there read off 38 in. as the nearest even diameter.

63 (*b*) To get the pull of the belt we remember that the cone belt was found in Example 1 to transmit 7.2 h.p. We therefore note that point on the vertical scale of horse powers at the extreme right of the main diagram which corresponds to 7.2, and then follow the diagonal from this point towards the left, until we meet the vertical line extending up from the point marked 3000 on the scale of velocities on the bottom line of this portion of the diagram. From this point of intersection we continue horizontally to the left to the vertical scale of pounds, on which we then read off the pull 80 lb.

64 (*c*) From the point corresponding to this 80 lb. we now continue diagonally to the left until we meet the vertical line extending up from the point corresponding to the belt speed 3000 on the scale marked "Velocity for Pull of Counter-Shaft Belt" at the bottom of this central portion of the main diagram. From this point we continue horizontally to the vertical line corresponding to the arc of contact, which, not being given, we will assume as 160 deg., and then again diagonally in the extreme left hand section of the diagram. Any simultaneous readings of width and thickness from points in the diagonal along which we are now moving, will then give a proper belt, and assuming as in Example 1 a thickness of $\frac{5}{16}$ in., we find the width to be $3\frac{1}{2}$ in.

65 (*d*) To find maximum initial tension for this belt, we proceed

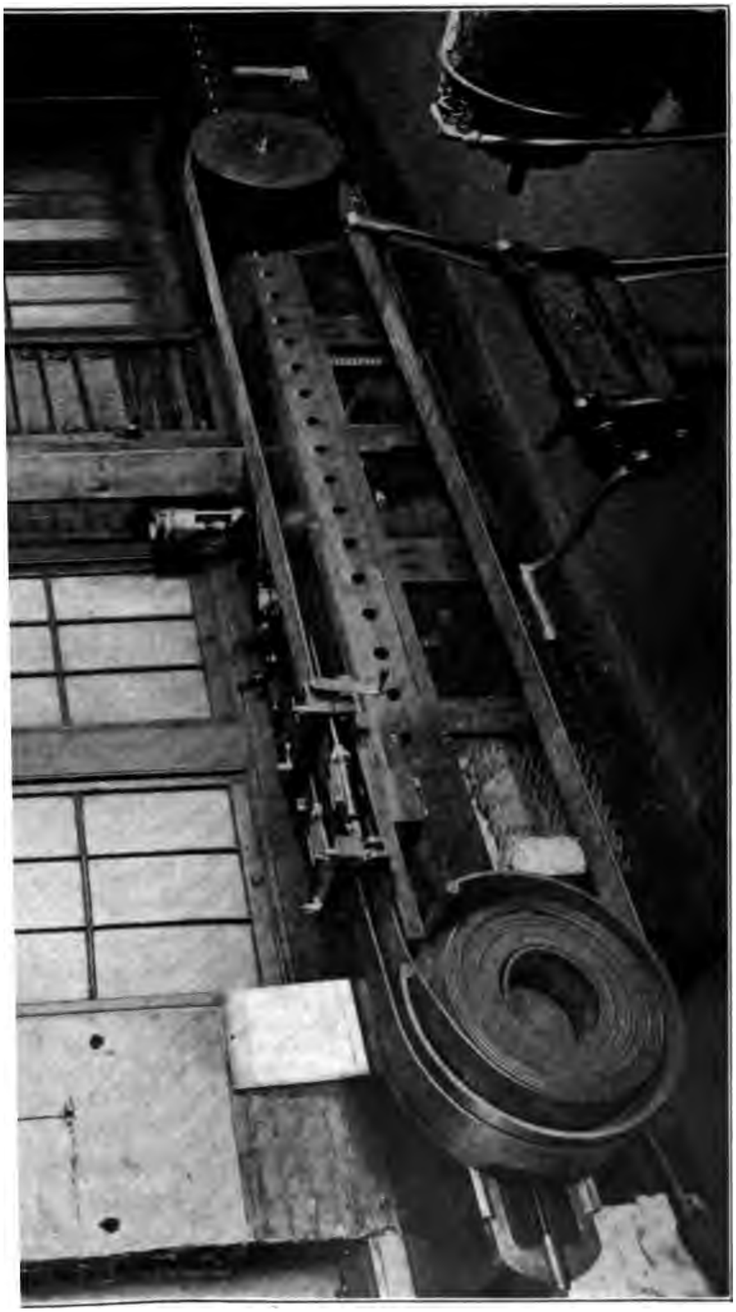

FIG. 6 BELT BENCH IN THE SHOPS OF THE LINK BELT COMPANY, NICETOWN, PHILADELPHIA, PA. THE USE OF THE BELT SCALES IS SHOWN IN DETERMINING THE LENGTH OF A NEW BELT

exactly as in Example 1, except that we use the scale marked "Veloc ity for Maximum Initial Tension for Counter-Shaft Belt" at the to of the middle section of the diagram, and then read this off on th scale of pounds as 157 lb.

66 (*e*) Similarly, we find the minimum initial tension to be 97.5 lt

MEANS OF SECURING AND MAINTAINING DEFINITE TENSIONS IN BELT

67 In his paper, Notes on Belting, Mr. Taylor referred to belt clamps provided with spring balances for weighing the tension i a belt. In the case of endless belts these scales are put directly on belt in its final position over its pulleys, while in the case of a bel with wire lacing, this is cut to length under the required tension o the specially designed belt bench illustrated in Fig. 6. As will be seen this bench is provided with a pair of pulleys which can be so adjuste that a tape-line will measure the same around these pulleys as ove the pulleys on which the belt is to run. A belt cut and laced to giv a certain tension when the bench pulleys have been properly adjustec will then be of a length to assume the same tension over its ow pulleys.

68 This indirect way of securing a desired tension in a belt wa first suggested by our fellow member, Mr. Gullow Gulowsen, who als made the drawings from which the first bench and the first improve belt scale were made by the Bethlehem Steel Company in the yea 1900.

APPENDIX

ELASTIC PROPERTIES OF BELTING

The only experiment recorded by Mr. Lewis to establish the elastic properties of leather belting is the following:

2 "A piece of leather belting 1 sq. in. in section and 92 in. long, was found by experiment to elongate $\frac{1}{4}$ in. when the load was increased from 100 to 150 lb., and only $\frac{1}{8}$ in. when the load was increased from 450 to 500 lb. The total elongation from 50 to 500 lb. was $1\frac{11}{16}$ in., but this would vary with the time of suspension, and the measurements here given were taken as soon as possible after applying the loads."

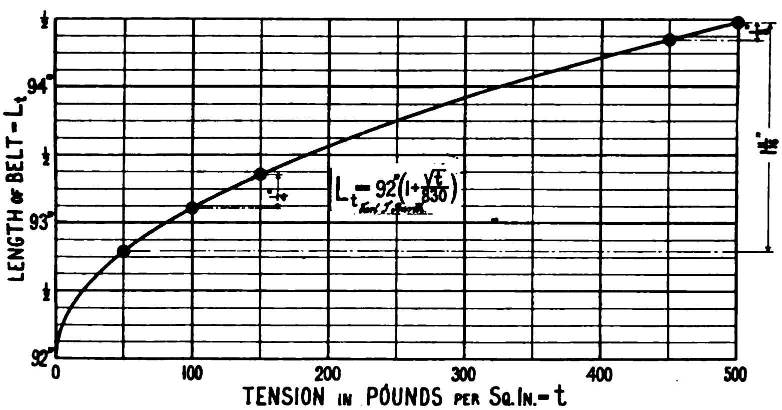

FIG. 1 PLOT OF EXPERIMENTS ON A PIECE OF BELTING 1 SQ. IN. IN SECTION AND 92 IN. LONG. TEST MADE BY WILFRED LEWIS AT THE WORKS OF WM. SELLERS & CO., PHILADELPHIA, IN 1885

3 These data have been plotted in Fig. 1, in which they are remarkably well represented by the formula

$$L_t = 92\left(1 + \frac{\sqrt{t}}{830}\right)$$

in which L_t is the elongated length of the belt, and t the load or tension per square inch.

The full development of the mathematical formulae of this paper, with some other related matter, is given in an unpublished supplement to the Appendix, which is on file in the Library of the Society for the use of any who wish to verify the mathematical work.

4 On the strength of this formula the writer originally established the theorem that *the sum of the square roots of the tensions in a belt is constant for all loads*, when no attention is paid to the weight of the belt.

5 However, he soon realized that it would not be safe to build a theory on a single experiment of this nature; and hence, in July 1901, while in the employ of the Bethlehem Steel Company, he undertook a series of similar experiments, and obtained permission of William Sellers & Company to use their emery testing machine for that purpose with the assistance of their shop engineer, Mr. Leonard Backstrom.

6 Nine pieces of belting were tested in all. The results upon one of those pieces are shown in Fig. 2, which is typical of all of them. Similar diagrams representing the other tests are filed with the unpublished supplement to the Appendix. In all cases the tests were made as rapidly as the loads could be adjusted and the extensometer readings taken.

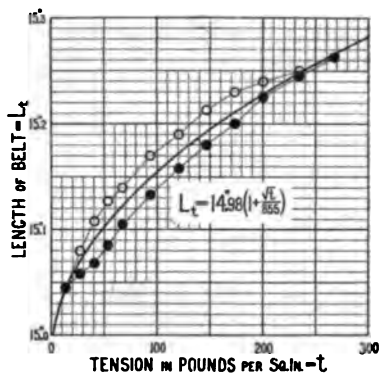

FIG. 2 PLOT OF EXPERIMENTS ON A PIECE OF SLIGHTLY USED DOUBLE BELTING $3\frac{15}{16}$ IN. WIDE BY $\frac{3}{8}$ IN. THICK

Test made by the writer at the works of Wm. Sellers & Co., Philadelphia, in 1901
The Supplement contains eight similar plots.

7 Each piece was several times subjected to a complete cycle of loads between two extremes. During the first few cycles the belts invariably showed different results, but always gave, eventually, practically the same readings for a number of cycles in succession, and these are the readings plotted in the figures.

8 The small filled-in circles represent the readings for an increasing load and the small open circles those obtained for a decreasing load. It is rather astonishing how much lag is shown by every belt. Unquestionably this has some influence on the law of change of tension in a belt, from its minimum to its maximum along its contact with a pulley. This matter has been given some consideration in the Supplement.

9 On account of this lag, apparently it would have been desirable to subject some of these belts to a series of smaller cycles, each between adjacent limits of the load. The best the writer could do with the results obtained was to average

the loop of each cycle by means of a parabolic curve, and thus obtain a value for the constant E for each belt on the supposition that the formula

$$L_t = L\left(1 + \frac{\sqrt{t}}{E}\right) \qquad [1]$$

is approximately correct. In the various formulae given, however, L is not the original 15 in. of length to which the extensometer was originally adjusted for each belt, but an ideal length only, for the estimation of the relations between the tension and the stretch for values never approaching close to zero.

10 But the best experiments for ascertaining the relations between the tension and the stretch in belts are unquestionably those by Prof. W. W. Bird, published in his paper on Belt Creep, read at the Scranton Meeting in 1905.

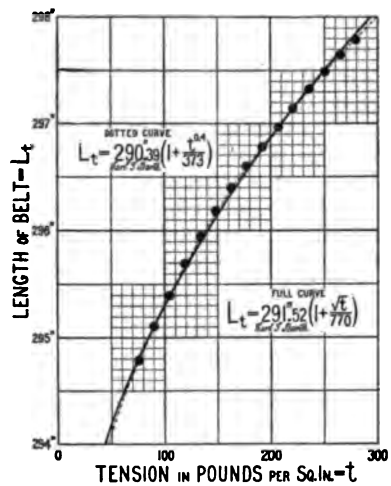

FIG. 3 PLOT OF EXPERIMENTS ON A SINGLE 4-IN., ENDLESS, RUNNING BELT

Test made by Prof. Wm. W. Bird at the Worcester Polytechnic Institute. The Supplement contains a similar plot on a single 6 in. laced belt. See Professor Bird's paper on Belt Creep in Volume 26 of the Transactions.

11 These have been replotted by the writer after making some slight corrections in the lengths given by Professor Bird, allowing for the influence of the sag in the belts and in the tensions given, by the addition of the estimated centrifugal tension, which was not measured by Professor Bird. The centrifugal tension was estimated after obtaining from Professor Bird the information that the belts were run at a speed of about 1000 ft. per minute. One of the diagrams is shown in Fig. 3 and another has been filed with the Supplement.

12 The plots made by the writer differ further from those in the original paper in that he laid off the tension in pounds per square inch of section of the belts.

13 It will be noted that the results have been approximated both by a dotted line representing a special form of the broadly general formula

$$L_t = L\left(1 + \frac{t^n}{E}\right) \qquad [2]$$

and by a full line representing this same formula with the special value ½ for n, or in other words, Formula 1. For regularity of results, these experiments are remarkable.

14 While the dotted curves with their more complicated formulae represent the experiments more closely, the full curves with their simpler formulae also cover the results so well that they may be looked upon as an excellent justification for the assumption previously made by the writer on the strength of the experiments made by Lewis and himself, namely, that *within the limits of ordinary working tensions of a belt, the difference between the lengths of a belt at different tensions is proportional to the difference between the square roots of those tensions.*

15 This proportion is implied in the general Formula 1, when by L we imply, not necessarily the slack length of a belt, but an ideal slack length on the basis of which the formula gives reliable results between ordinary working limits of t.

16 Taking the average of the values of E in all twelve sets of experiments we get 895. Leaving out two experiments, one with a value of E exceeding 1000 and another for which E was less than 800, and taking the ten remaining experiments with values of E between 800 and 1000, we get 890; while if we take the average of only the two experiments by Professor Bird we get only 861.

17 As will be seen hereafter, the writer has adopted 864 as an average working value, because this figure, combined with certain other constants, results in the simple final constant coefficient 0.04 in the right member of Equation 5. For an average practical working formula on which to build an improved theory for the transmission of power by leather belting, we thus have

$$L_t = L\left(1 + \frac{\sqrt{t}}{864}\right) \qquad [3]$$

in which L_t equals the length of a belt under the unit tension t when its slack length is L.

18 However, it will appear later on, that the value 864 adopted for E has significance only in the formulae developed for long horizontal belts, as E disappears in these formulae when the weight of the belt is neglected.

LAW OF VARIATION IN THE TWO TENSIONS OF A LONG HORIZONTAL BELT

19 In developing an expression to represent the law of variation in the two tensions of a long horizontal belt, the free strands of the belt only are considered, and then, for the sake of argument, are assumed to be attached to the ends of two double levers fulcrumed in the middle, as shown in Fig. 4 and 5.

20 That the parts of the belt in contact with the two pulleys remain at practically constant length independent of any variation in the tensions of the two strands, and thus have no material influence on this variation, will be shown in the Supplement.

21 In Fig. 4 the levers are parallel to each other, and the two strands of belting whose normal slack lengths l are supposed to be equal, must form equal catenaries under equal unit tensions t_0, corresponding to the equal initial tensions in the strands of a belt continuous over its two pulleys.

22 In Fig. 5 the levers have been moved through equal angles in opposite directions, thus tightening the bottom strand to the unit tension t_1, and slackening the top strand to the unit tension t_2, with corresponding changes in the respective catenaries.

23 Under this arrangement the sum of the chords of the catenaries remains constant under all variations of the tensions, a condition that corresponds to that of an actual working belt over its two pulleys, which remain at a constant distance apart under all conditions of tension in the belt.

24 In considering the problem, the customary approximations in dealing with catenaries were made, in connection with Formula 1 for the elastic properties of leather belting, and then the following general formula developed for the relations between the unit working tensions t_1 and t_2 in a belt, as dependent on the initial unit tension t_0, the weight W per square inch of cross section of each free strand, and the elasticity constant E in Formula 1.

$$\sqrt{t_1} + \sqrt{t_2} = 2\sqrt{t_0} + \frac{W^2E}{24}\left(\frac{1}{t_1^2} + \frac{1}{t_2^2} - \frac{2}{t_0^2}\right) \qquad [4]$$

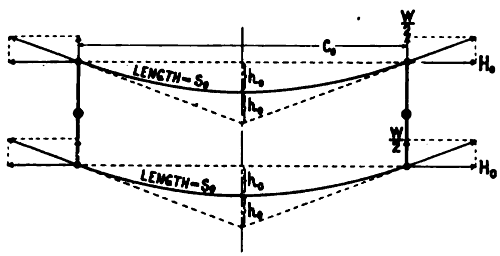

FIG. 4

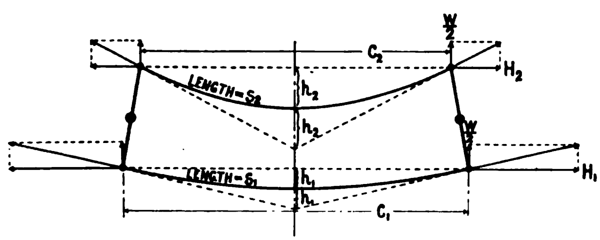

FIG. 5

FIG. 4 AND 5 REPRESENTATION OF CHANGE IN TENSIONS AND SAGS IN STRANDS OF A HORIZONTAL BELT

25 The average weight of a cubic inch of leather belting being about $\frac{1}{30}$ lb. we may write $W = \frac{l}{30}$, in which l is the center distance in inches between the two pulleys of a horizontal drive; and by also assuming 864 as the average value of E, as already done in Formula 3, we may write more specifically

$$\sqrt{t_1} + \sqrt{t_2} = 2\sqrt{t_0} + 0.04\,l^2\left(\frac{1}{t_1^2} + \frac{1}{t_2^2} - \frac{2}{t_0^2}\right) \qquad [5]$$

26 For very short belts the last terms in Equations 4 and 5, which are only tentatively solvable equations, become very small as compared with the term $2\sqrt{t_0}$, and by neglecting it entirely we write for vertical belts as well as for short horizontal belts, or even approximately for all belts:

$$\sqrt{t_1} + \sqrt{t_2} = 2\sqrt{t_0} \qquad [6]$$

This formula may be enunciated as a new theorem of the relations of the tension in a belt, thus: *Under any variation of the effective pull of a belt, the sum of the square roots of the tensions in the two strands remain constant*, as against the old fallacious supposition that the sum of these tensions remains constant.

27 However, without entirely neglecting the last term in Formulae 4 and 5 above, these may be made solvable with respect to t_1, by first approximating the value of this term by the substitution in it alone of an average relation of the tensions t_1 and t_2. As will appear from a study of the diagram Plate 1 in the body of the paper, such an average relation of the tensions is

$$t_1 = \frac{t_0^2}{t_2}$$

28 Substituting this in the last term of Equation 5 and then solving this with respect to t_1 we get

$$t_1 = \left[2\sqrt{t_0} - \sqrt{t_2} + 0.04\,l^2\left(\frac{t_2^2}{t_0^4} + \frac{1}{t_2^2} - \frac{2}{t_0^2}\right)\right]^2, \qquad [7]$$

which gives practically identical results with the original Equation 5 for such values of t_0 and l as fall within ordinary practice.

29 If we express the tensions t_1 and t_2 in terms of the initial tension t_0 by writing

$\delta_1 = \frac{t_1}{t_0}$ and $\delta_2 = \frac{t_2}{t_0}$, Formula 7 reduces to

$$\delta_1 = \left[2 - \sqrt{\delta_2} + 0.04\,\frac{l^2}{t_0^{2.5}}\left(\delta_2^2 + \frac{1}{\delta_2^2} - 2\right)\right]^2 \qquad [8]$$

in which form it is under certain circumstances more readily applied.

TESTING FORMULA 8 BY THE RESULTS OF LEWIS' EXPERIMENTS ON HORIZONTAL BELTS

30 In Fig. 6 the two tensions simultaneously obtained by Mr. Lewis in one series of his experiments have been plotted in terms of the initial tensions of the belt, the tensions in the tight side of the belt being laid off horizontally and the tensions in the slack side vertically, in the same manner as is done on the diagram Plate 1, in the body of the paper. Similar diagrams, representing five additional series of experiments made by Lewis, are filed with the Supplement to the Appendix.

31 However, as the apparatus used by Mr. Lewis did not measure the centrifugal tension in his belts, and as he had no occasion to calculate values of this

quantity, these have been calculated for the present purpose, and added to the effective tensions tabulated by Mr. Lewis.

32 Each experiment is represented by one of the small filled-in circles, and is numbered the same as in the tables from which the experiments were taken from Mr. Lewis' paper.

33 Unfortunately, but very naturally, the initial tension did not remain constant throughout a set of experiments, and in plotting the tensions it was therefore necessary to estimate for each individual experiment where the initial tension was between the values measured at the beginning and at the end of each set of experiments. This was done by assuming that the initial tension measured at the beginning of a set of experiments held good for the first experiment, and that the initial tension measured at the end of a set of experiments held good for the last experiment, and that there was an equal drop for each experiment.

34 That the initial tension was not constant during each set of experiments is the reason why the actual tensions obtained were not plotted, but instead their ratios δ_1 and δ_2 to their respective initial tensions.

35 In each figure, Equation 8 is also given with the center distance of the pulleys for each particular belt introduced as an approximate value of l.

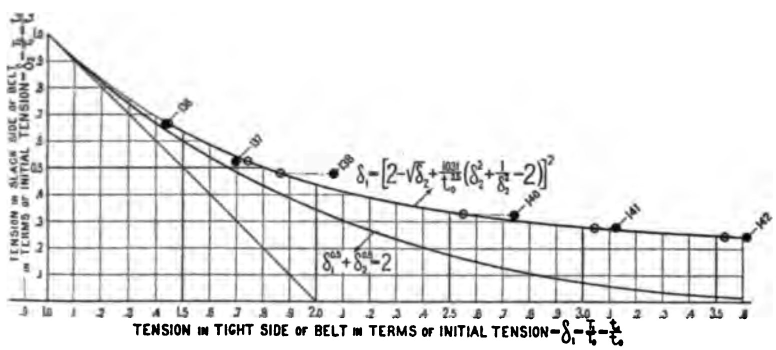

FIG. 6 PLOT OF EXPERIMENTS BY WILFRED LEWIS

Horizontal Double Belt $2\frac{1}{4}$ in. wide, $\frac{5}{16}$ in. thick and 32 ft. long. 20 in. Pulleys. Average Value of Initial Tension $t_0 = 70$ lb. per sq. in. (The Supplement contains five similar plots.)

36 By the introduction of the value obtained for δ_2 in each experiment plotted, a corresponding value was calculated for δ_1 by the formula mentioned, and this value also plotted, and then a curve was drawn to cover the points thus calculated. The points themselves are indicated by the little circles drawn around them.

37 The close coincidence between the curves representing Formula 8 and the experimental results, is certainly all that can be desired in the way of an experimental verification of the soundness of Formulae 7 and 8.

38 The other, lower curve drawn on each diagram represents the relation between the tensions in a belt when the influence of its weight is neglected, as given by Equation 6, which is also given on each diagram, in the form

$$\delta_1^{0.5} + \delta_2^{0.5} = 2 \qquad [9]$$

BELT CREEP AND ITS INFLUENCE ON THE COEFFICIENT OF FRICTION BETWEEN A BELT AND ITS PULLEY

39 In the paper on his experiments, Mr. Lewis drew the conclusion that the friction between a belt and its pulley varies greatly with its velocity of slip, so that the greater the slip the greater the friction. But as he did not make a substantial study of the elastic properties of leather, upon which the phenomenon of *belt creep* depends, he had no means of distinguishing in his experiments between the necessary slip due to the creep of the belt, and the amount that was slip pure and simple of the belt as a whole.

40 In most of his experiments the belt did an amount of work that called for much greater friction between the belt and its pulley than that corresponding to the creep of the belt alone, and this resulted in additional or true slip that produced the friction needed to make the belt exert the pull called for.

41 By means of Formula 3 it was possible also to derive a formula that gives a good idea of the actual creep of a belt in terms of the tensions in its two strands, which was the object of Professor Bird's paper on Belt Creep, from which the experiments plotted in Fig. 3 of his paper were taken.

42 This formula, the mathematical development of which is given in the Supplement, is

$$v = \tfrac{1}{2}(V_1 - V_2) = \frac{V_1}{2}\,\frac{\sqrt{t_1} - \sqrt{t_2}}{864 + \sqrt{t}} \qquad [10]$$

in which

v = actual average velocity of the creep of the belt on each of its two pulleys.

V_1 = velocity of the tight strand of the belt, which is the same as the circumferential velocity of the driving pulley.

V_2 = velocity of the slack strand of the belt, which is the same as the circumferential velocity of the driven pulley.

43 The total creep of the belt on both pulleys together expressed in per cent of V_1 is then

$$x = \frac{100\,(\sqrt{t_1} - \sqrt{t_2})}{864 + \sqrt{t_1}} \qquad [11]$$

44 It must be borne in mind, however, that Formulae 10 and 11 take account of creep only, and have nothing to do with any additional slip due to a sliding of the belt as a whole over its pulleys, though the expression

$$v = \tfrac{1}{2}(V_1 - V_2)$$

taken by itself always represents the total sum of the average creep of the belt and the additional sliding of the belt as a whole, over each of its pulleys, when such additional sliding does take place.

45 Considering the matter in this light Mr. Lewis calculated and tabulated the velocity of sliding from the observed loss in speed between the pulleys in his various experiments.

46 In Tables 1 and 2 appear some of the data thus tabulated by Mr. Lewis. However, instead of tabulating merely the effective tensions measured by him, the centrifugal tensions have here been figured and allowed for, and then the total

tensions thereby obtained subsequently converted into tensions per square inch of cross-section. A column has also been added giving the percentage of average slip due to belt creep alone, as figured by Formula 11.

47 It will be seen that in most of the experiments the velocity of sliding greatly exceeded the average due to the elastic creep alone, and that thus the belt as a whole slid over the pulleys in addition to the elastic creeping, thus showing that the friction corresponding to this creep alone was not enough to produce the pull the belt was called on to perform.

48 The relation between the total average velocity of sliding of the belt on each pulley, and the corresponding coefficient of friction calculated by Mr. Lewis by the formula

$$\text{Ratio of Effective Tensions} = e^{\phi\alpha}$$

and also copied in Tables 1 and 2 of this paper, is plotted in the diagram Fig. 7 and in a similar diagram of the Supplement. On these diagrams is also shown a curve representing the equation

$$\phi = 0.6 - \frac{2}{4 + v} \qquad [12]$$

in which ϕ is the coefficient of friction and v the total average sliding velocity of the belt in feet per minute. These results were obtained from belts that had been in active service, and tested without the application of any belt dressing.

49 As a somewhat conservative average the curve is seen to cover the results obtained with the belts in a normal condition in a highly satisfactory manner.

50 The question now arises, What coefficient of friction ought to be assumed in calculating the pulling power of a belt at any given speed? In view of the foregoing it does not seem right to assume an average coefficient for all belt speeds Nor would it be right to base it on an average total sliding velocity of a belt corresponding to a fixed percentage of the belt speed, for even a very low percentage would mean a very high sliding velocity in the case of a high-speed belt, while a high percentage would mean only a moderate sliding velocity in the case of a slow-speed belt, and it would seem that the speed with which a belt slides over its pulley would principally determine the life of a belt that meets with no accident.

51 After considerable study over the subject, the writer has assumed a variable coefficient of friction expressed by the empirical formula

$$\phi = 0.54 - \frac{140}{500 + V} \qquad [13]$$

in which V is the velocity of the belt in feet per minute

52 Equating Formulae 12 and 13 we get

$$v = \frac{160 + 0.88\,V}{85 + 0.03\,V}$$

as the velocity of sliding on each pulley in terms of the velocity of the belt itself

53 As the percentage of slip between the circumferential speeds of the two pulleys of a belt is twice the percentage of the average total velocity of sliding v of the belt over each pulley, we may now write

$$x = \frac{200\ v}{V} = \frac{200}{V} \cdot \frac{160 + 0.88\,V}{85 + 0.03\,V} \qquad [14]$$

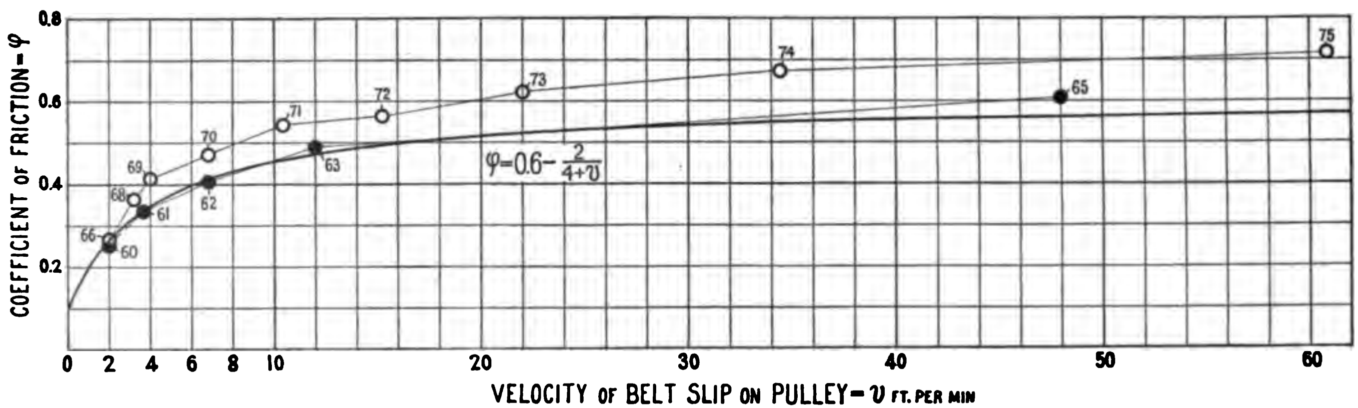

Fig. 7 Plot of Experiments made by Wilfred Lewis, at the works of Wm. Sellers & Co., Philadelphia, in 1885, on a Single Belt $5\frac{1}{2}$ in. Wide by $\frac{7}{32}$ in. Thick, Belt in Ordinary Working Condition

Average total initial tension in experiments 60 to 65 $= t_0 = 81.6$ lb. per sq. in.
Average total initial tension in experiments 66 to 75 $= t_0 = 127.5$ lb.
(The Supplement contains two similar plots.)

as an expression for the percentage of slip corresponding to the coefficient of friction expressed by Formula 13.

54 In Table 3 are listed simultaneous values of ϕ and x as expressed by Formulae 13 and 14 and also, for the sake of comparison, for ϕ figured by the formula

$$\phi = 0.6 - \frac{400}{800 + V}$$

which is the value of ϕ by Formula 12, for $v = \frac{V}{200}$, that is, for a uniform slip of one per cent at all belt speeds.

55 A study of Fig. 7 and the similar figures in the Supplement will show that the variation in the coefficient of friction with the initial tension of the belt is so conflicting as to make it best to leave this out of consideration entirely, and so adhere to the customary assumption that the coefficient of friction is independent of the intensity of the pressure. Therefore, as our whole theory of the variation of the coefficient with the belt speed rests entirely on the formula

$$\text{Ratio of Effective Tensions} = e^{\phi\alpha}$$

this will be used unhesitatingly, in spite of the fact that its absolute validity will be disputed in the Supplement for a number of reasons.

EFFECT OF CENTRIFUGAL FORCE IN A BELT

56 While the effect of the centrifugal force in a fast-running belt seems to have been fully understood by all who have previously treated the subject before this Society, it is still but imperfectly understood by many engineers and mechanics; and has even been treated in a wrong way by certain text-book writers who have followed the work on belting by the late Professor Ruleaux. It has therefore seemed desirable, in the Supplement to the Appendix, to go into details on this part of the subject.

57 Subsequently, the following general formula was developed for the loss in effective tension in a belt, due to its centrifugal force:

$$t_c = \frac{w}{300\,g} V^2$$

in which

t_c = loss in effective tension per square inch of cross-section of belt.
V = velocity of belt in feet per minute.
w = weight of one cubic inch of belting in pounds.
g = acceleration of gravity in feet per second.

Substituting $w = \frac{1}{30}$, as in Par. 25, and $g = 32\frac{1}{6}$, we have more specifically

$$t_c = 0.000003454\ V^2 \qquad [15]$$

which is substantially the same formula as that given in Mr. Nagle's paper, Formula for the Horse-Power of Leather Belts, read at the Hartford meeting in 1881.

FORMULA FOR PULLING POWER OF A HORIZONTAL BELT IN TERMS OF ITS INITIAL TENSION

58 For a horizontal belt on pulleys of the center distance c and one square inch of cross-section we now have, by substituting c for l in Formula 7,

$$t_1 = \left[2\sqrt{t_0} - \sqrt{t_2} + 0.04\, c^2 \left(\frac{t_2^2}{t_0^4} + \frac{1}{t_2^2} - \frac{2}{t_0^2} \right) \right]^2 \qquad [16]$$

$$\text{Ratio of effective tensions} = \frac{t_1 - t_c}{t_2 - t_c} = e^{\phi\alpha} \qquad [17]$$

$$t_c = 0.000003454\, V^2 \qquad [15]$$

$$\phi = 0.54 - \frac{140}{500 - V} \qquad [13]$$

$$p = t_1 - t_2$$

in which formulae

c = center-distance of pulleys, in inches.

t_0 = initial tension. (per square inch of cross-section of belt)

t_1 = tension in tight strand or side. (per square inch of cross-section of belt)

t_2 = tension in slack strand or side. (per square inch of cross-section of belt)

t_c = centrifugal tension; or, more correctly, loss in effective tension due to centrifugal force. (per square inch of cross-section of belt)

p = effective pull. (per square inch of cross-section of belt)

e = basis of Naperian system of logarithms, 2.71828.

ϕ = coefficient of friction between belt and pulley.

α = the lesser arc of contact of belt on pulleys, in radians $= \frac{\pi}{180} \times$ arc in degrees.

V = velocity of belt in feet per minute.

59 However, an attempt to combine these five equations algebraically to obtain an expression of p in terms of t_0, V, and α, leads to an equation solvable by trial only, and for this reason the diagram Plate 1, the use of which was explained in Par. 11 to 24 in the body of the paper, was constructed to effect the solution graphically.

60 For the construction of this diagram Equation 5 was used after substituting c for l as above,

$$\delta_1 = \frac{t_1}{t_0}, \text{ and } \delta_2 = \frac{t_2}{t_0}$$

as in deriving Formula 8 from 7. It thus became

$$\sqrt{\delta_1} + \sqrt{\delta_2} - 2 = 0.04 \frac{c^2}{t_0^{2.5}} \left(\frac{1}{\delta_1^2} + \frac{1}{\delta_2^2} - 2 \right) \qquad [18]$$

and was in that form solved tentatively to obtain a series of points in a series of curves, each representing a certain value of the factor $\frac{c^2}{t_0^{2.5}}$. On the diagram these curves form the bottom field of curves in the middle section.

61 The equation was in each case first solved to obtain an approximate value only of δ_1, in terms of an assumed value of δ_2, by resorting to the approximation δ_2^2 for $\frac{1}{\delta_1^2}$ in the right-hand member of the equation, as shown by Formula 8.

62 Then by substituting this approximate value of δ_1 in the term $\frac{1}{\delta_1^2}$ in the right-hand member of Equation 18 this was again solved for a still closer value of δ_1.

63 For the lesser values of $\frac{c^2}{t_0^{2.5}}$, this closer value of δ_1 differed inappreciably from the first approximation, while for the greatest values plotted on the diagram, the equation was solved twice to get the values actually plotted.

64 However, these greatest values of $\frac{c^2}{t_0^{2.5}}$ never occur in the practical use of belting, and hence the very construction of the diagrams under consideration proved the validity of the approximations δ_2^2 for $\frac{1}{\delta_1^2}$, which is equivalent to the approximation $\frac{t_2^2}{t_0^4}$ for $\frac{1}{t_1^2}$ resorted to in Par. 27 in modifying Equation 5 to Equation 7.

FORMULAE FOR PULLING POWER OF VERTICAL BELTS IN TERMS OF INITIAL TENSION

65 For a vertical belt the relation between the tensions of a belt is expressed by the simple Equation 6, and this can readily be combined with the rest of the equations listed in Par. 58 (in the manner done in the Supplement to the Appendix), which leads to the following formula:

$$p = 4\left(\frac{e^{\phi\alpha}+1}{e^{\phi\alpha}-1} - \sqrt{\frac{4\,e^{\phi\alpha}}{(e^{\phi\alpha}-1)^2} + \frac{0.000003454\,V^2}{t_0}}\right)t_0 \qquad [19]$$

66 This formula is simple enough, though a great improvement over the one derived on the erroneous supposition that the sum of the tensions is constant for all loads.

67 By means of this formula the pulling-power of a belt can easily be determined in terms of its initial tension. However, for a uniform unit initial tension for all speeds, the unit tension in the tight side would vary so much that belts running at different speeds but tightened to a uniform maximum unit initial tension, and allowed to run until this had dropped to a uniform minimum tension, would require re-tightening at greatly different periods.

68 As already pointed out the writer has arrived at the conclusion that the periods at which belts running at different speeds will have to be re-tightened, will be nearly constant if they are all made to do their work at such initial tensions as under full load will result in the same *sum* of the *tension* in the *tight side* and

***one-half* the *tension* in the *slack side* of the *belt*, at the two extremes of the initial tensions, just before and after retightening.**

PULLING POWER OF BELTS IN TERMS OF A CONSTANT SUM OF THE TIGHT TENSION AND ONE-HALF THE SLACK TENSION, AT ALL SPEEDS

Vertical Belts

69 This condition is expressed by the equation

$$t_1 + \tfrac{1}{2} t_2 = A = \text{a constant}$$

Combining this with Equation 17 (in a manner shown in the Supplement) we get

$$p = \frac{(e^{\phi\alpha} - 1)(2A - 0.00001036\, V^2)}{2\, e^{\phi\alpha} + 1} \qquad [20]$$

$$t_1 = \frac{2A + p}{3} \qquad [21]$$

$$t_2 = \frac{2(A - p)}{3} \qquad [22]$$

and

$$t_0 = \frac{4A - p + \sqrt{(4A - p)^2 - 9p^2}}{12} \qquad [23]$$

These formulae are the ones plotted in the diagrams Fig. 1 and 2 in the body of the paper, for $A = 240$ and 160 lb., respectively.

70 By a similar treatment (as shown in the Supplement) we are also able to get an expression for the initial tension in a horizontal belt, which gives results of a high degree of accuracy. This expression is

$$t_0 = \tfrac{1}{4}\left[\sqrt{t_1} + \sqrt{t_2} - \left(\frac{c}{5}\,\frac{t_1 - t_2}{t_1\, t_2}\right)^2\right]^2 \qquad [24]$$

which is evaluated by first determining p by Formula 20, and subsequently t_1, and t_2 by Formulae 21 and 22, as for a vertical belt, paragraph 69.

71 However, while this formula is of great theoretical interest, it is hardly of much practical value; as the initial tension determined by it will differ but little from that determined for a vertical belt by Formula 23, except for belts of extraordinary lengths.

72 One very interesting general conclusion may now be drawn from Formula 24; namely, that while actually doing work two horizontal belts of unequal lengths may be under precisely the same tensions, but this being the case, when idle the longer belt will be under a slightly lower initial tension.

73 It appears, however, that the popular notion that horizontal belts drive a great deal more than vertical belts, is not well founded.

TABLE 1

EXPERIMENTS BY WILFRED LEWIS, AT THE WORKS OF WM. SELLERS & CO., PHILADELPHIA, 1885, ON SINGLE BELT $5\frac{1}{4}$ IN. WIDE BY $\frac{7}{32}$ IN. THICK AND IN ORDINARY WORKING CONDITION WITHOUT BELT DRESSING. BELT SPEED = 800 FT. PER MINUTE. THESE EXPERIMENTS ARE PLOTTED IN FIG. 7 OF THE SUPPLEMENT, WHICH SEE. SEE ALSO HIS PAPER, No. 198, VOL. 2 OF TRANSACTIONS, TABLE 1

Experiment No.	Unit Initial Tension t_0	Unit Tension in Tight Side of Belt = t_1	Unit Tension in Slack Side of Belt = t_2	Percentage of Total Slip	Average Velocity of Sliding in feet per minute = v	Coefficient of Friction by the Formula $\frac{t_1 - t_c}{t_2 - t_c} = e^{\phi\alpha}$	Percentage of Slip due to Belt Creep alone, Formula $z = \frac{100(\sqrt{t_1} - \sqrt{t_2})}{864 + \sqrt{t_1}}$
60	81.6 lb. per square inch	125.33	58.67	0.5	2.0	0.251	0.41
61		131.42	46.58	0.9	3.6	0.336	0.53
62		142.00	42.00	1.7	6.8	0.407	0.62
63		152.41	35.75	3.0	12.0	0.490	0.73
65		179.92	29.92	12.0	48.0	0.610	0.91
66	127.5 lb. per square inch	177.42	77.42	0.5	2.0	0.270	0.52
68		198.25	64.92	0.8	3.2	0.365	0.69
69		208.77	58.67	1.0	4.0	0.418	0.77
70		219.08	50.75	1.7	6.8	0.472	0.87
71		229.50	46.17	2.6	10.4	0.545	0.95
72		244.08	44.08	3.8	15.2	0.569	1.02
73		256.58	39.92	3.5	22.0	0.623	1.10
74		252.42	35.75	8.6	34.4	0.677	1.13
75		283.66	33.67	15.2	60.8	0.719	1.25

TABLE 2

EXPERIMENTS BY WILFRED LEWIS, AT THE WORKS OF WM. SELLERS & CO., PHILADELPHIA, 1885, ON DOUBLE BELT $2\frac{1}{4}$ IN. WIDE BY $\frac{5}{16}$ IN. THICK AND IN ORDINARY WORKING CONDITION WITHOUT BELT DRESSING. BELT SPEED = 800 FT. PER MINUTE. THESE EXPERIMENTS ARE PLOTTED IN FIG. 19 OF THE SUPPLEMENT, WHICH SEE. SEE ALSO HIS PAPER, No. 198, VOL. 2 OF TRANSACTIONS, TABLE 2

Experiment No.	Unit Initial Tension t_0	Unit Tension in Tight Side of Belt = t_1	Unit Tension in Slack Side of Belt = t_2	Percentage of Total Slip	Average Velocity of Sliding in feet per minute = v	Coefficient of Friction by the Formula $\frac{t_1 - t_0}{t_2 - t_0} = e^{\phi\alpha}$	Percentage of Slip due to Belt Creep alone by Formula $z = \frac{100(\sqrt{t_1} - \sqrt{t_2})}{864 + \sqrt{t_1}}$
105	73.5 lb. per square inch	104.9	47.5	0.3	1.2	0.263	0.38
106		123.4	37.5	0.8	3.2	0.395	0.57
107		146.0	32.6	1.7	6.8	0.511	0.73
108		171.5	29.5	4.3	17.2	0.600	0.87
121	283.0 lb. per square inch	403.0	175.9	0.7	2.8	0.267	0.77
124		450.0	137.5	1.5	6.0	0.387	1.07
125		465.0	124.8	2.3	9.2	0.424	1.17
126		482.2	113.5	3.7	14.8	0.469	1.28
127		497.5	99.2	10.1	40.4	0.523	1.39
128	343.5 lb. per square inch	511.3	227.0	0.5	2.0	0.261	0.85
131		557.0	187.2	1.1	4.4	0.350	0.99
133		589.5	162.4	1.8	7.2	0.414	1.30
134		603.0	148.2	2.7	10.8	0.450	1.39
135		618.0	134.0	5.1	20.4	0.490	1.49

TABLE 3

RELATIONS BETWEEN COEFFICIENT OF FRICTION, VELOCITY OF SLIDING AND BELT SPEED. SEE PAR. 54

V = Velocity in feet per minute	Per cent of Slip $= \frac{200}{V} \times \frac{160 + 0.88V}{85 + 0.03V}$	Velocity of Sliding $v = \frac{160 + 0.88V}{85 + 0.03V}$	Coefficient of Friction of $\phi = 0.54 - \frac{140}{500 + V}$	Velocity of Sliding at 1 per cent slip $v = \frac{V}{200}$	Coefficient of Friction of $\phi = 0.6 - \frac{2}{4 + v} = 0.6 - \frac{400}{800 + V}$
0	8	1.88	0.260	0.00	0.100
50	9.432	2.36	0.285	0.25	0.129
100	5.636	2.82	0.307	0.50	0.156
200	3.690	3.69	0.340	1.00	0.200
300	3.010	4.51	0.365	1.50	0.236
400	2.640	5.28	0.384	2.00	0.267
500	2.400	6.00	0.400	2.50	0.292
600	2.227	6.68	0.413	3.00	0.314
700	2.090	7.32	0.423	3.50	0.333
800	1.983	7.93	0.432	4.00	0.350
900	1.889	8.50	0.440	4.50	0.365
1000	1.808	9.04	0.446	5.00	0.378
1200	1.675	10.05	0.458	6.00	0.400
1400	1.566	10.96	0.466	7.00	0.418
1600	1.474	11.79	0.473	8.00	0.433
1800	1.394	12.55	0.479	9.00	0.446
2000	1.325	13.25	0.484	10.00	0.457
2500	1.180	14.75	0.493	12.50	0.479
3000	1.067	16.00	0.500	15.00	0.495
3500	0.974	17.05	0.505	17.50	0.507
4000	0.898	17.95	0.509	20.00	0.517
4500	0.832	18.72	0.512	22.50	0.525
5000	0.768	19.40	0.514	25.00	0.531
5500	0.727	20.00	0.517	27.50	0.536
6000	0.684	20.53	0.519	30.00	0.541
6500	0.646	21.00	0.520	32.50	0.545

DISCUSSION

HENRY R. TOWNE. The earliest investigation of this subject was by General Morin, of the Conservatoire des Arts et Métiers, who gave, in a volume published, I think, about 1850, the results of his experiments to determine the coefficient of friction of belts on pulleys, and algebraic formulae to express the power transmitted under varying conditions. For many years these formulae were accepted universally. General Morin's experiments were made under laboratory conditions.

2 In 1867 I made a series of experiments to determine, under conditions approximating those of actual use, the coefficient of friction and also the tensional strength of commercial belting. These experiments, and a discussion by the late Robert Briggs on the mathematical conditions involved in the problem, were published in the Journal of the Franklin Institute in 1868. Under the title of the Briggs and Towne Experiments, the conclusions thus reached were quoted and accepted for many years, by Professor Rankine, Professor Reuleaux, Professor Unwin, and many other technical writers. A. F. Nagle, in a valuable paper contributed to the Transactions of the Society in 1881 (Vol. 2, p. 91), accepted the results of the Towne experiments as the basis for his discussion of the mathematical problems involved.

3 The Transactions for 1886 (Vol. 7) contained two important contributions to the literature on this subject. One of these is a paper by Professor Lanza (p. 347), which first prominently calls attention to the importance of *speed of slip* as a factor in the transmission of power by belting. The other is a paper by Wilfred Lewis (p. 549) giving the results of a long and elaborate series of experiments in the shops of William Sellers & Co., and demonstrating, among other things, that the proposition first enunciated by General Morin, and accepted unquestioningly by all subsequent authorities, namely, that *the sum of the tensions is constant* $(T_1 + T_2)$, does not hold true in all cases, and is therefore erroneous.

4 The Transactions for 1894 contains another most valuable paper, by Fred. W. Taylor (p. 204), giving the results of his large experience covering many years in the use and observation of belting under the conditions of actual practice. Many new and important deductions based on the investigations of Mr. Taylor are availed of by Mr. Barth in the conclusions and recommendations contained in his paper. One of the most important facts demonstrated by Mr. Taylor relates to the value of *increased thickness* of belts, and the resulting expediency of a larger and more general use of double belts. He was also the first to demonstrate and set forth clearly the economic gain to be derived from the scientific care of belting.

5 Finally, Mr. Barth, availing himself, as he has stated, of the work of his predecessors, especially that of Mr. Lewis and Mr. Taylor, has completed, for the present at least, the study of this problem, which has thus extended over some sixty years, giving us an

elaborate and apparently a conclusive demonstration of the soundness of the mathematical conclusions finally reached, furnishing *working formulae for practical use,* and presenting a most ingenious application of the *slide rule* to the problems involved in the practical use of leather belting.

6 The Society is to be congratulated on including in its roster of membership the names of all those since General Morin who have taken the lead in ascertaining the facts and in determining therefrom the rules which govern the application of leather belting to industrial uses.

7 Mr. Barth's system has now been in use for about two years in the works of the Yale & Towne Mfg. Co., Stamford, Conn., where it has accomplished a substantial increase in economy and efficiency.

WILFRED LEWIS. I am clearly of the opinion that Mr. Barth has discovered and formulated principles of the greatest practical value in the solution of the problems of the transmission of power by leather belting.

2 It is difficult in a paper of this kind to separate the practical from the theoretical without discarding the most valuable part of the undertaking. The laborious work done by the author in order to reach his conclusions, and recorded in the appendix to this paper, is really the basis of the superstructure reared by him and gives the reader some idea of the immense amount of patient research and good sound reasoning employed in building up a complete analysis of the subject.

3 Mr. Barth is the first, I believe, to analyze the peculiar elastic properties of leather, and to demonstrate in a convincing way the effects of these properties in the use of belting under varied conditions. His analysis of the combined effects of elasticity and sag is very original and ingenious, and even aside from the results obtained his methods cannot fail to interest investigators in other fields of research. Difficult and complex problems have been solved by making certain assumptions and approximations that are quite allowable as the means to an end, and it is in these short cuts from the intricate and unwieldly to the simple and practical that he has displayed such remarkable ingenuity. At the same time, for those not enough interested in every step to care to follow a mass of mathematical formulae, Mr. Barth has presented his conclusions in a form available for immediate use.

4 Popular impressions, even though well founded, are often exaggerated beyond reasonable bounds, and while it is true that horizontal belts of considerable length are preferable in the transmission of power to vertical or shorter ones, it will be a surprise, I believe, to engineers, that there really is so little advantage in a long horizontal belt over any length of belt in any position. All this results from the exposure of the fallacy that the sum of the tensions is constant, a belief exploded 23 years ago, although the far-reaching effect of the exposure on the transmission of power by belting has never before been so clearly expounded.

5 The author's treatment, also in the unpublished supplement to the appendix, of the effect of variations in pulley diameter upon the transmission of power, I believe to be absolutely original, and his conclusion that a belt will slip on a driven pulley before it will slip on a driver of the same diameter indicates a subtlety of analysis rarely displayed in our proceedings, and is a fair index of the painstaking care with which the whole paper has been written. Although not perhaps of very great practical importance, as a new discovery, the analysis might well be included in the appendix to the paper, rather than in the unpublished supplement to the appendix.

W. D. Hamerstadt. The writer has been somewhat closely associated with work on pulley and belt drives, and recently has had occasion to compare the results of some carefully conducted experiments with the results which might be expected from the use of formulae as proposed in Mr. Barth's paper. Considering the many variable factors, these comparisons are remarkably favorable, and for average conditions of operation, the relationships which have been established would appear to hold quite true.

2 One almost vital point of consideration in the actual design of belt drives seems to have been touched upon but lightly, however, and then in a manner which, as the author himself has stated, leaves some room for discussion—namely, values of the coefficient of friction to be used in the formulae given, under varying conditions of service. While the value of the coefficient of friction will not affect the theory of belt transmission as given, it will seriously affect the size of drive required to do a given work, and having now a good theoretical basis for work, and being assisted by the observations of others, additional experimental work might well be done for the determination of such values, using as nearly as possible good average leather belting and operating under actual conditions of service.

3 Based partly on the conclusions of Professor Lanza or of Mr. Wilfred Lewis, as given in early papers before the Society, and partly on the very mechanical reasonableness of the thing as he puts it, Mr. Barth assumes that, given a belt and pulley, the value of the coefficient of friction to be used in any case will be determined to a

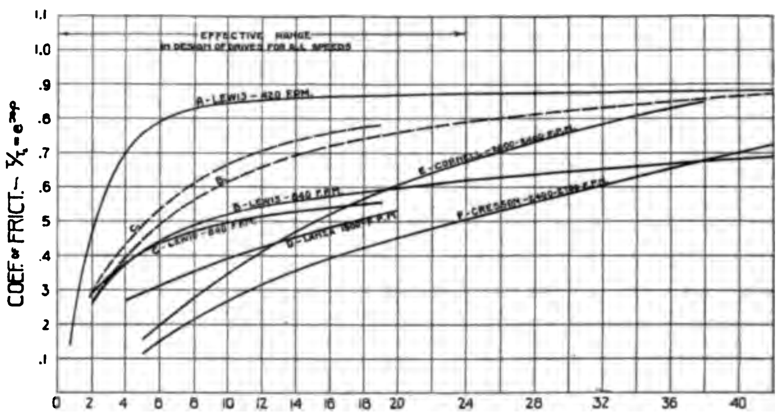

FIG. 1 RELATION BETWEEN COEFFICIENT OF FRICTION AND VELOCITY OF SLIP

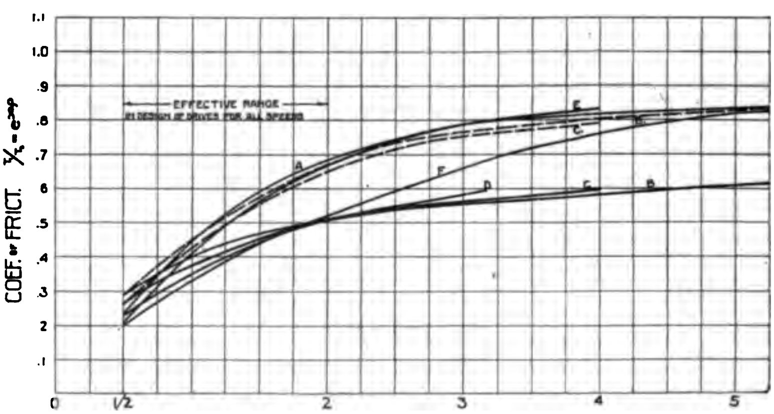

FIG. 2 RELATION BETWEEN COEFFICIENT OF FRICTION AND PERCENTAGE OF SLIP

great extent by the velocity with which the belt slides on its pulley. Taking then a curve representing average relations between these two factors for any convenient speed of belt, values are at once available for the coefficient of friction for any speed of belt and any condition of slip desired.

4 That there exists some ground for such reasoning cannot be

TABLE 1 DATA USED IN MAKING COMPARISONS SHOWN IN ILLUSTRATIONS

Curve	Test	Speed of Belt Ft. per Min.	Belt			Pulley		Remarks
			Kind Leather	Size Inches	Condition	Kind	Diam. Inches	
A	Wilfred Lewis: Table 4, Exp. 209–259..................	420	Oak-tanned Double	4 x $\frac{5}{16}$	New, Dry and Clean	C. 1.	10	Belt Run under Severe Conditions. Tests made to Determine Point of Slip at which Belt Leaves Pulley.
B	Wilfred Lewis: Table 1, Exp. 60–75....................	840	Single......	5½ x $\frac{7}{32}$	Old, Good and Pliable	C. 1.	20	....
C	Wilfred Lewis: Table 2, Exp. 105–135..................	840	Double	2¼ x $\frac{5}{16}$	Old Dry and Clean Quite Pliable	C. 1.	20	....
D	Prof. Lanza: Vol. 7, Trans. West Belt, Exp. 1–11. ...	1500	....	....	....	C. 1.	......	$T_1 + T_2$ Assumed Constant and Corresponding Error Made in Calculated Results.
E	Cornell University: Exp. 1–82	2000–2400	Oak-tanned new, single	5 x $\frac{3}{16}$	Dry and Clean	C. 1.	24	
F	Geo. V. Cresson Co.: Catalog B, 1906, Exp. 1–100........	2400–2700	New, single	3 x $\frac{3}{16}$	Dry and Clean	C. 1.	24	Tests made at Cornell University

NOTE: All of the belts given were in good working condition when tested, being free from oils, grease or dressings. Belts B and C had been subjected to more or less oil in service before testing, however, which accounts in part for lower relative values of their coefficients of friction.

denied, but a brief comparison of results actually obtained from tests performed from time to time, on belts operating at widely different speeds of service, leads one seriously to question its application to practice. Such comparisons rather lead one to expect more nearly correct results when drives are designed on the basis of relative slip between belt and pulley.

5 Fig. 1 and Fig. 2 show a series of curves representing, for a number of different tests, the relation existing, first between values of the coefficient of friction and velocity of slip, and second, between values of the coefficient of friction and percentage of slip. Information regarding the data from which these curves were plotted is given in the table. Letters designate corresponding tests in either set of curves.

6 Referring to Fig. 1, it will be noted that for each different speed of belt there appears to exist a clear and well defined relation between values of the coefficient of friction and velocity of slip; at lower velocities of slip more especially, the value of the coefficient appearing to be higher for slow-speed belts and lower for high-speed belts. Obviously then, any curve representing a relation between values of the coefficient of friction and velocity of slip holds true only for that speed of belt for which it is plotted and cannot be used indiscriminately for all speeds of belts. The effective range of velocities of slip which would be used in the design of belt drives would probably be from 0 to 25 ft. per minute, as indicated in Fig. 1, and the error which might be incurred then in using either of the extreme outside curves shown (even though they do not represent maximum possible range of speeds of belts) would vary from about 70 per cent to values almost infinitely large. The curves B_1 and C_1 indicate the relation that would presumably have held true between values of the coefficient of friction and velocity of slip for belts B and C had those belts been such as to have shown a maximum value of the coefficient equal to that of belts A, E or F.

7 From Fig. 2 it will be noted that for any given speed of belt the same general relation between values of the coefficient of friction and per cent of slip appears to hold true, and belt drives designed on such a basis might then reasonably be expected to give anticipated results at all speeds of operation. When curves B_1 and C_1 are plotted to represent higher values of the coefficient of friction for belts B and C, as in Fig. 1, the similarity in form of these curves is remarkable —the more so as they represent tests performed in some cases over twenty years apart.

8 As an example of the results to be expected when drives are designed on the basis of velocity of slip of the belt, as proposed in the paper, let two extreme conditions of service be taken: one a drive operating at a speed of 400 ft. per minute and at a slip of about 2½ per cent, the other a belt operating at a speed of 5000 ft. per minute and at a slip of about 1 per cent. The slow-speed belt would then have a velocity of slip of 5 ft. per minute on each pulley, the high-speed belt of 25 ft. per minute. Referring to Fig. 7 in Mr. Barth's Appendix, it will be found that for such velocities of slip the values of the coefficient of friction to be used should be respectively 0.38 and 0.53; but the maximum value of the coefficient of friction at even the highest velocities of slip of 60 ft. per minute, as shown, is only about 0.57 and it appears then that the overload capacities of all drives is limited to about that value. This amounts, in the case of the slow and high-speed belts given, to about 50 and 7½ per cent respectively.

9 It is safe to say, however, that fully 80 to 90 per cent of all high-speed drives are used in connection with electrical machinery, and for such work drives must have an overload capacity of at least 50 per cent of their rated capacity. It would be necessary then, for such practice, that belts be originally designed for correspondingly lower velocity of slip, amounting in this case to a velocity of about 5 ft. per minute. The high-speed belt, noted above, at ordinary conditions of service would then operate at a relative total slip between pulley and belt of only one-fifth of one per cent.

10 While the author has had occasion to observe a large number of successful high-speed drives on electrical machines, just the condition of slip here indicated has never been noted, but in almost every case the overload capacity of the drives has been noted as a function of the relative slip between belt and pulley.

11 When it is further considered that, generally speaking, the point at which the belt will leave its pulley is a function of the per cent of slip and is taken independent of the speed of belt, it certainly appears that the relative slip cannot but play an important factor in determining values of the coefficient of friction to be used for such drives.

FRED. W. TAYLOR.[1] The belt is one of the oldest and most commonplace of the elements used in shop practice, so that engineers designing new establishments or remodeling old ones, who wish to be up-to-date, naturally incline toward the use of the electric drive

[1] Discussion abstracted.

rather than the belt. There is no doubt, however, that this has led to the use of the electric drive in many instances where the belt would be far more economical and satisfactory in almost every way.

2 In the average machine shop, for instance, the writer is prepared to say that for more than half of the machines the belt drive can still be used with greater economy and with more satisfactory results than the electric drive; only on the assumption, however, that the belting is systematically cared for. The most serious objection to the belt drive as generally used is the loss of time due to interruption to manufacture when retightening and repairing, and to the loss of driving power and consequent falling off in output, when the belt is allowed to run too slack. Belts can be tightened and repaired at regular intervals after working hours, however, with the use of spring-balance belt-clamps to get the right tension, causing thus practically no interruption to manufacture.

3 As will be explained later, it has been shown by an accurate record kept through a long term of years, that in the average machine shop the average cost per belt per year is $2.25. This includes the original cost of the belt, plus all labor and materials used in maintaining, repairing and cleaning it throughout its life. No similar statistics for the maintenance and renewal of the motor drive seem to be available, but I think no one will contend that the latter can in any way approach this economy.

4 In a great number of cases the electric drive should be used in the machine shop, but in the writer's judgment the burden of proof still rests on the motor drive to show in each case that the economy in delivery and removal of work more than makes up for the extra cost of installation and maintenance, and for the delays incident to repairs, blowing out fuses, etc. In large machines economy lies on the side of the motor drive in many instances, but with almost all small machines the belt drive should still be used. In view of these facts, the belt drive is hardly a back number. In fact, the manager of one pulley manufactory told me recently that even during the dull times his company had been selling from twelve to fifteen thousand pulleys per month.

5 Under the rules still in common use, a large proportion of belt drives are badly designed, and belts are used under heavier tensions than they should be for all-round economy. All who have experimented with belting or who have been interested in the mathematics of belting, will be filled with admiration at the remarkable analysis which Mr. Barth has made of this difficult problem. Even

Mr. Lewis, whose experiments and scientific conclusions have properly been given first place among writings on this subject, tells us in his paper that life is too short to attempt a complete mathematical solution of the problems involved. Yet this is precisely the task which has been accomplished by Mr. Barth.

6 The experiments of Messrs. Briggs and Towne and those of Messrs. Bancroft and Lewis will remain for many years as classic monuments in the development of our scientific knowledge of belting laws; but Mr. Barth's remarkable analysis of the work of former experimenters, supplemented by his accurate though less voluminous experiments on the elastic properties of belting and on the rate and extent of the fall in tension of belts, has rendered his conclusions as to economical speeds and the proper sizes of belts more reliable than those of any previous writer. His final recommendations should be accepted, therefore, rather than those in the papers of Messrs. Towne, Lewis, or the writer.

7 It may be of interest to know how the figure of $2.25, quoted earlier in the paper as the cost per belt per year, was found.

8 In the new machine shop of the Midvale Steel Company, beginning in the year 1884, the writer experimented[1] with all of the belts in the shop, in practical use; and upon the comparative values of the four leading types of leather belting then in common use. This experiment lasted nine years with belting running night and day (equivalent to eighteen years running ten hours per day). Exact records were kept of all items affecting the life and economical use of belting, and at the end of the experiment, among other items, it was found that the average belt cost (under the ordinary belt rules then in use, as, for example, those used on the cone pulleys of the various machines; and on the ten hour basis) $3.34 per belt per year for the first cost plus all labor and materials used in maintenance and repairs. These are double belts, averaging 29 ft. long by 3.8 in. wide.

9 These belts were run under too high tension for economy, however. They lasted on an average 14 years (ten hours per day). The remaining belts in the shop, which proved more economical, lasted on an average not far from 28 years (ten hours per day), and cost per year per belt less than $2.50 for first cost and maintenance,

[1] These experiments are described in a paper entitled Notes on Belting, presented before the Society December 1893, and forming part of Volume 15 of the Transactions.

etc. And this although they were materially larger than the cone belts, averaging 50 ft. long by 4.84 in. wide. The machines in this shop averaged much larger than in the average shop, and an investigation has led me to the conclusion that in the average shop the average belt would be about equal to a 3-in. double belt, 20 ft. long. The first cost plus the maintenance of this belt would not be greater than $2.25 per belt per year.

10 The care of belting should be entirely taken out of the hands of the men who are running the various belt-driven machines, and belts should be systematically retightened at regular intervals, with belt-clamps fitted with spring-balances, each belt having the tightening strain carefully figured in advance. Belting should also be cleaned at regular intervals, and should be softened with the small amount of belt-dressing which is needed to keep it in perfect condition. A laborer can be quickly trained to tighten and care for all the belts in the shop during the noon hours and on Saturday afternoons and at other times when the shop is not running.

11 Two elements of great importance in Mr. Barth's paper are The Influence of Pulley Diameters on the Sum of the Tensions of the Belt and a condensation of the discussion of the formula, Ratio of Effective Tensions $e^{\phi\alpha}$. Not only has this discussion a great theoretical interest, but the conclusions have a distinct practical value.

Charles Robbins.[1] In applying motors in textile mills where the belt has been in use for years and the proposition is essentially that of constant and uniform speed we discovered that the capacity of the spinning frames was largely increased. This led us to make tests upon the loss of speeds, or slip of belts and their lack of uniform operation. The net result in using the induction motor instead of the belt is an increased production of at least seven per cent, and in some instances even ten per cent. Probably some of this increase is due largely to the fact that the belting systems tested were not designed in accord with Mr. Barth's system; but I believe that a system of belts will never approach the uniform and constant speed of an induction motor.

2 The question of efficiency may be classified as (1) the primary efficiency from the engine shaft to the shaft of the driven machine;

[1]Charles Robbins, Westinghouse Electric and Manufacturing Company, East Pittsburg. Pa. Discussion abstracted.

(2) the economies which result from the use of the electric motor drive. These secondary economies, which are undoubtedly the most important, will vary with the class of industry to which the electric motor is applied. It is greatest in those industries where the load-time factor of the installation is lowest and where the inherent characteristics of the electric motor are of greatest value. These characteristics are as follows:

a Ability to adjust the speed according to the demands of the work.

b Absolute certainty of a uniform and constant speed.

3 While these two characteristics may seem to be opposed, they are important factors in the increase of production of different types of machines. As widely separate examples: for a machine tool the readiness with which the speed of a motor may be varied to the right quantity for the work required contributes to its increase of production; on the other hand textile mill service requires an absolutely constant and uniform speed, which is obtained from the induction motor.

4 In determining the value of an electric motor drive the essential point is always the secondary, or accruing economies from its use, rather than the primary economy, although when the primary is added to the secondary the net result will be extremely satisfactory.

GEO. N. VAN DERHOEF.[1] The author's plan of proportioning belts so that the slack will be taken up at approximately regular intervals of time, regardless of speed or power transmitted, is excellent from a theoretical point of view. He is obliged, however, to divide belts into two classes—machine belts and countershaft belts, under different initial tensions, and therefore with different periods between adjustments. I think it will be absolutely necessary to provide more classes. In some cases first cost is of greater importance, and in other cases the expense or inconvenience of taking up belts is the main consideration. With a large belt, running night and day, the stopping of the drive to take up the belt is a serious matter. In the case of many drives, however, this is a matter of small moment.

2 I have had considerable experience with large quarter-twist belts, running from 12 to 20 in. in width, for connecting horizontal and vertical shafts, and have seen results that appear incredible in view of much of the theoretical data published on belting. These belts were under high unit-tension, and always subjected to reverse

[1] Discussion abstracted.

bending over deflecting idlers. Probably one reason for the success of belts of this kind is the automatic regulation, within limits, of the slack-side tension, due to the belt working up and down across the face of the pulley on the vertical shaft. As far as I have observed, belt drives of this kind, when properly designed and erected, have been as satisfactory as horizontal belts with about the same distance between centers.

3 Possibly the larger unit-stresses frequently used necessitate a slight actual slipping of the belt on the pulleys, with some corresponding increase in the coefficient of friction. This should not necessarily be regarded as poor practice, but simply as a factor to be weighed against savings in first cost, friction losses, etc. There seems to be no fundamental objection to slipping within certain limits, provided such slip is a constant quantity. All belts are continually sliding, to some extent, on the surface of the pulleys, due to the theoretical creep caused by the elasticity of the belt. A little more would not necessarily be serious. The surface of a well finished leather belt is such that sliding on a polished iron pulley will not cause much harm provided the heat generated by the slip is dissipated with sufficient rapidity to prevent the temperature of the belt surface from rising too high. This, of course, involves a loss of energy, as do very large belts under low tensions, and the crowning of pulleys. The writer desires to emphasize that due consideration should be given to all the factors involved.

4 Spring belt-clamps should be used wherever practicable, and ought not to be very expensive if manufactured in reasonable quantities. In the majority of cases, however, we shall have to be satisfied with figuring belts properly, and leave the actual initial tension to fate.

5 The idea that the maximum working-stress of a belt should not be determined by its ultimate strength is, I believe, correct. This becomes more apparent in studying transmission ropes. It is a well-known fact that the maximum unit-stress for a manila transmission rope should be of such amount that the side-pressure between the lubricated fibers of the rope will not cause abrasion when the ropes bend over the sheaves, and the fibers slide on one another. Probably some such internal action takes place in the case of leather belts. In transmission ropes the ultimate strength bears a greater ratio to the proper maximum working stress than is the case with leather belts. Manila rope is therefore a very safe transmitting band.

6 The constant lengthening of belts in service has its counterpart in ropes. Where a rope is simply carried around two sheaves, as

in the separate rope system, the general equation of the rope is without question similar to that which the author has shown to be true of leather belts.

7 The continuous system of rope transmission, with its automatic tension carriage, has the slack-side tension maintained at a minimum. This is one of the fundamental reasons why the continuous system can transmit the same amount of power at the same rope speed and with the same rope life, with less rope than is possible with the separate wrap system. A few years ago the continuous system was looked upon by most engineers with considerable scepticism; its enormous development in the last quarter of a century is due simply to its basis on absolutely sound mathematical principles.

WALTER C. ALLEN. My contribution to the discussion will relate to the practical results obtained from the installation of an improved method of caring for belting, rather than to the technical phases of the question. In this connection a brief description of the working out of the improved system in the works of the Yale & Towne Mfg. Co. may prove interesting.

2 The problem of transmitting large amounts of power by means of belting is not a serious one with us, as our power is for the most part transmitted electrically; each room is provided with one or more motors, and the power is distributed from them through line and countershafts to the machines. The great majority of our belts are small, and many of them run at high speeds. Altogether we have about 4800 belts, so that their proper maintenance is an important and somewhat difficult problem.

3 Early in 1905, at Mr. Barth's suggestion we undertook an investigation of our belting and the methods employed in its upkeep, as a result of which we decided to adopt a system of caring for belting recommended by Messrs. Taylor and Barth. For the sake of brevity I have divided my notes into comparative statements, of the conditions before and after the adoption of the new method as affecting each element of this important subject. It may seem that the conditions existing before the installation of the new plan were distinctly bad, but I venture to say that they were as good as those in many manufacturing establishments at the present time, if not better. The improved conditions, however, are so infinitely superior to the old that by comparison the latter appear extremely antiquated and crude.

4 *Tensions.* Under the old plan we had no means of knowing

with any accuracy the tension of a belt. It was left to the individual judgment and experience of those doing the repairing, so that inevitably the tensions of the belts varied in proportion to the variation of judgment of the repair men.

5 The first step in the reorganization was the building of a belt bench and the provision of tension scales such as are shown in Fig. 6. These are used now altogether for the determination of tensions.

6 *Records.* Under the old regime we had no records whatever of our belts.

7 Under the new plan we have a record of each belt showing its location; its type, i. e., whether open or crossed, countershaft or machine belt; kind of leather; thickness, width and length. These records also show for each belt the dates of inspection.

8 *Organization.* Under the old plan our millwrights cared for the heavy belts, but the repairing was done only when the belt gave way, or stretched so that it failed to transmit the necessary power. The small machine belts were cared for by the individual machine operators, many of whom knew absolutely nothing about belting, and in some cases our investigations showed that ignorant operators had attempted to tighten a belt by cutting out a piece, and, finding that they had cut out so much that the belt would not go over the pulleys, were then compelled to cut out still more and set in a piece in order to make the belt long enough to do the work. In these cases also the belts were not repaired until they actually gave out through breakage or failed to give the necessary pull.

9 Under the new plan a gang of four men do absolutely nothing else but inspect belting and attend to the repairs and retightening. A belt room has been provided in which is an annunciator, and a series of push buttons are arranged at the telephone central, so that in case of an accident to a belt the foreman or gang boss can call the belt man easily. In a plant as large as ours the annunciator results in a great saving of time.

10 A tickler system was installed by means of which the belt gang are notified regarding the belts to be inspected each day. After the inspections are made these tickler cards are returned to the office where the proper records are made and the ticklers put back for the next inspection.

11 These belt men take their lunch hour from 11 to 12 o'clock, working during the noon hour, and are thereby enabled to repair many belts which could not be repaired when the works are running, without loss of time to other employees.

12 *Fastening.* Under the old plan there was no fixed rule regarding the fastening, rawhide lacing and belt hooks being used indiscriminately. Under the present plan Jackson wire lacing, put into the belts by means of a machine, is universally used. For continuous belts, under the old plan we used a kind of glue which took from three to ten hours to set satisfactorily. Under the present plan we are using a special glue which will set hard in thirty minutes. This also results in a saving of time in the case of an accident to continuous belts.

13 *Belt Dressing.* Under the old plan comparatively little belt-dressing was used, but in many cases rosin was used through ignorance of the fact that it causes the belting to deteriorate rapidly. We now use entirely Plomo belt-dressing, which is extremely useful and tends to prolong rather than to shorten the life of the belt.

14 *Reclamation of Belts.* Under the old plan no reclamation was attempted, but at the present time we reclaim a considerable amount of belting each year. Belting damaged on the edges is cut down and used for narrower belts, short pieces are scarfed and glued together and the oil is taken out of oily belting and the belts used over again.

15 *Kind of Belting.* Several kinds of belting were used under the old plan, but we have gradually standardized our belting until at the present time practically nothing but a high grade of oak-tanned belting is used.

16 *Cost of Up-Keep.* Of course there was no method of determining the cost of maintenance under the old plan. Our records show that during the year 1906 the labor-cost of maintaining our belting system was 96 cents per belt. During 1907 it was 73 cents and during 1908, 45 cents. This decrease has of course been due to the increased efficiency of the men doing the work and to the fact that experience has indicated where inspection periods could be lengthened out, and also to the fact that the belting is now in such condition that expensive breakdowns seldom occur.

17 The foregoing statements describe briefly the various features of the old and the new plans; a summary of the advantages of the new plan follows:

a Decreased cost of belting. The cost for the year 1907 was only about 60 per cent of that for 1906, despite the fact that we installed more new machinery in 1907 than in 1906.

b Increased efficiency of machines, due to the fact that the tensions are maintained much more uniformly than formerly.

c Continuous production by both men and machines, due to decreased interference due to belt-breakdowns.

d Uniform type of belt lacing, decreasing danger to employees.

e Decreased cost of maintenance.

f Under the present plan the cost of maintenance appears as a separate item where it can be watched and compared with that of previous periods to determine the relative economies, while under the old plan the figures were combined with a mass of others so as to make it impossible to determine how much it had cost.

CONCLUSION

18 When we first commenced to install the new system we had all sorts of trouble as is generally the case with any new thing. The plan was opposed by foremen, gang bosses and workmen, each of whom had an idea that the new tensions were entirely wrong, and that the machines would never do the work properly, unless they could adjust the belting according to their individual ideas. One of the best evidences of the value of the present plan is that this antagonism has entirely disappeared, and what was at first considered by many an interference and a hindrance is now accepted as a help and is believed to be entirely satisfactory by those competent to hold an opinion.

Mr. Taylor. The original experiments at the Midvale Steel Works were started in 1884; 17 years later, when all the machinery in that shop was taken out, one of the belts, which was of the type of those run under proper rules, that is, approximately the low tension suggested by Mr. Barth, had run all that time night and day under heavy tension. During this time it had required tightening only nine times, and at the end of the equivalent of 34 years of ten-hour service that belt came off its pulleys and was immediately put to work on another machine, in good condition. This instance of the life of a belt properly taken care of and properly tightened will be a surprise to the man accustomed to see a belt go out of use in from two to five years. This statement has just been determined.

Dwight V. Merrick.[1] As I am interested in chain drives, I will draw attention to some experiments made by Hans Renold, Ltd., of Manchester, England, and embodied in a pamphlet issued May 1908, comparing the relative efficiency of chain and belt drives on automatic machines. Mr. Renold claims that with the chain drive the output was increased 20 per cent, fewer drills and parting tools were used, and a better finish was obtained on the work. He says: "The tool did its work *unflinchingly* at every part of the revolution of the spindle—no more and no less." He further states that the wear and tear on the spindle and countershaft bearing was considerably reduced. These statements were so striking that the Link-Belt Company, with which I am associated, decided to make further tests. In one of these which I was detailed to make I maintained a constant feed and speed and used the same tools with each drive, and in all cases the tool was used until it became necessary to re-grind, the object being to cut off as many pieces or drill as many holes as possible before this condition was reached. The tool when chain-driven did considerably more work than when belt-driven. I quote from my report as follows:

2 These tests were made on a 3 in. by 36 in. Jones and Lamson turret lathe, with "blue chip steel" cutting-off tool $\frac{3}{16}$ in. wide, cutting off cold-rolled shafting $2\frac{1}{2}$ in. diameter, feed 0.012 in. per revolution.

3 Care was taken in forging, treating and grinding the several tools used, to insure uniformity in their cutting qualities; but to obviate the possibility of the results being affected by the cutting qualities of the different tools, each tool was used with both drives.

4 One of the tools cut off 16 pieces when chain-driven before it became necessary to re-grind, and only 9 pieces when belt-driven. The cutting speed in both cases was 94 ft. per minute, feed 0.012 in. per revolution, and another tool cut off 8 pieces when chain-driven against 5 when belt-driven. In this latter case the cutting speed was 130 ft. per minute, feed 0.012 in. for chain and belt.

5 As the cutting periods in the above test were so short, two more series of tests were made with longer continuous periods. These tests were made on a drill press with new $\frac{7}{8}$ in. carbon steel drills in a soft cast-iron block, 3 in. thick. The same drill was used on both drives, and was carefully and uniformly ground for each test.

6 One of the drills when belt-driven drilled 31 holes before it became necessary to re-grind, but when chain-driven the same drill drilled 57 holes; the cutting speed in both cases was 62 ft. per minute,

[1] Dwight V. Merrick, Link-Belt Mfg. Co., Nicetown, Philadelphia, Pa.

TABLE 1

RESULTS OF EXPERIMENTS ON A 3-IN. BY 36-IN. JONES & LAMSON TURRET LATHE 0.012 IN FEED PER REVOLUTION.

Mark on Tool.	Kind of Drive	No. of Pieces	Metal Cut by Tool Inches	Cutting Speed Ft. per Minute	Condition of Tool	Time Minutes	R.P.M. of Spindle
2	Belt	6½	8.125	94	Ruined	5.91	143
	Chain	16	20.	94	Ruined	14.70	143
	Belt	9	11.25	94	Ruined	7.15	143
5	Chain	7⅞	9.843	128	Ruined	4.81	196
	Belt	4⅞	6.093	134	Ruined	2.72	203
4	Belt	1	1.25	134	Good	0.51	203
	Belt	¼	0.312	151	Ruined	0.12	231
	Chain	1	1.25	126	Good	0.54	193
	Chain	¾	0.937	151	Ruined	0.39	231
1	Belt	1	1.25	129	Good	0.53	197
	Belt	½	0.625	146	Ruined	0.15	223
3	Chain	1	1.25	129	Good	0.53	197
	Chain	1	1.25	149	Fair	0.44	228
	Chain	⅜	0.468	195	Ruined	0.15	299

NOTE: A higher cutting speed was obtained by the chain drive.

TABLE 2

RESULTS OF EXPERIMENTS ON A DRILL PRESS WITH NEW ¾-IN. DIAMETER CARBON STEEL DRILLS, 0.018 IN. FEED PER REVOLUTION, IN A SOFT CAST-IRON BLOCK, 3-IN. THICK

Mark on Drill	Kind of Drive	Holes Drilled Number	Metal Cut by Drill Inches	Cutting Speed R.P.M.	Condition of Drill after Drilling Holes	Time Minutes	R.P.M. of Spindle
1	Belt	31	93	62.2	Started to ruin	18.91	273
1	Chain	57	171	60.5	Corner rounded, needed grinding, starting to ruin	35.91	264
2	Chain	37	111	62.2	Starting to ruin	22.57	273
2	Belt	20	60	62.2	Starting to ruin	12.20	273

NOTE: A great many more holes were drilled by the chain drive.

TABLE 3

RESULTS OF EXPERIMENTS ON THE SAME DRILL PRESS AS IN TABLE 2 WITH NEW $\frac{23}{32}$-IN. DIAMETER CARBON STEEL DRILLS, 0.018 IN. FEED PER REVOLUTION, IN A VERY HARD CAST-IRON BLOCK, 3 IN. THICK

Mark of Drill	Kind of Drive	Holes Drilled Number	Metal Cut by Drill Inches	Cutting Speed R.P.M.	Condition of Drill after Drilling Holes	Time Minutes	R.P.M. of Spindle
3	Chain	17	51	28.0	Starting to ruin	19.84	148.8
3	Belt	14	42	28.4		15.40	151.4
4	Chain	17	$49\frac{1}{2}$	28.0	Started to run on 17th hole $1\frac{1}{2}$ in. deep	18.48	148.8
4	Belt	13	39	28.2	Started to ruin	14.44	150.0

NOTE: More holes were drilled by the chain drive, but the percentage of gain was not anywhere near as great as in Table 2.

feed 0.018 in., and another drill at the same speed and feed drilled 37 holes when chain-driven, against 20 when belt-driven.

7 The other series of drill tests was made on the same drill press, with $\frac{23}{32}$ in. carbon steel drills on a very hard cast-iron block, 3 in. thick. One of the drills when chain-driven drilled 17 holes before it became necessary to re-grind, against 14 holes when belt-driven; the cutting speed in both cases was 28 ft. per minute, feed 0.018 in. per revolution, and another drill did 17 holes, chain-driven, against 13 belt-driven, same feed and speed as above.

8 The results were so gratifying that further tests are being made on four similar automatic machines, at our plant in Indianapolis, two fitted with belt drives and two with chain drives. The same feeds and speeds will be maintained with each drive throughout the series of tests, but a variety of tests will be made to establish the maximum efficiency of both belt and chain drives, to the best of our ability. The results will all be tabulated and published in a pamphlet in the near future.

9 The accompanying tables contain the tabulated results of my experiments.

F. A. WALDRON. After listening to this paper, one naturally asks the question, What is its commercial value? Mr. Allen has answered this very well, but I will give a little of my own experience with the system.

2 At the plant of the Yale & Towne Company, most of the responsibility for the condition of belts, prior to the author's going there, was placed with me and I am willing to take any criticisms. I became an ardent advocate of Mr. Barth's work on belts, however, particularly because of the practical results obtained.

3 After leaving the Yale & Towne Company, I had occasion to purchase a Barth bench and spring-balance and apply the elements of the system without spending a large amount in replacing countershafts. I established the system of varying tensions on different machines. A light countershaft would not stand as heavy tension on the belt as the author originally prescribed. Tensions on belting, lengths, taking up, etc., were recorded. A record of complaints received in the millwright department for a specified number of countershafts and machine belts had been kept, and for ten days or two weeks before installation something like 150 complaints came in. After complete installation of the Barth bench and scales and the Barth system, the complaints dropped to 80 for two weeks, and six weeks later to 35, showing the commercial results of systematic care of belts.

4 Belts as low as $1\frac{1}{2}$ in. wide, and some heavy double belts three to four inches wide, were the limits on size.

5 This system was installed almost at the cost of my reputation, and on leaving that concern, I supposed that the belt bench and bench-scales would be relegated to the scrap heap. Having an opportunity to put in a belt-bench and scales elsewhere, however, I wrote the firm asking if they did not want to sell the bench and scale and they said "no."

A. A. Cary. I was much interested in Mr. Allen's remarks concerning the employment of Mr. Barth's system and formulae for the selection and proper application of belts to drive the numerous machines at the Yale & Towne plant, but explanation of one essential point is needed to show how this can be practically accomplished.

2 As I understand, one important factor required in the formula used to determine the proper initial tension to which each belt must be subjected when put in place, is the horse-power to be transmitted by that belt. It has been stated that 4000 belts are used in this plant, operating perhaps one-half that number of machines. I would like to know the method employed to determine the power requirements of each of these machines so as to obtain the required value of this factor when the formulae are used.

3 If we merely *guess* at the power required, we depart from the exact scientific method of determining information in our belt problems and recede toward the "rule of thumb" method, as a formula is no more exact than is the value of the most uncertain quantity employed in its solution. If Mr. Barth can give us any "short cut" method for determining the power required by machines to be driven by belts, he will funish information that will give his formulae a very practical value.

A. F. NAGLE. This paper does not pretend to present new facts, but sets forth, in mathematical formulae and diagrams, data obtained by Messrs. Lewis, Taylor and others. It also diagrams some simple arithmetical computations. As a work of mathematical study and diagrammatic representation, the paper is admirable, but as a practical aid to a busy engineer, it seems to me too complicated. The only part which holds my attention is the diagram in Fig. 3, giving the horse power of belts at different velocities, and of two types spoken of here as countershaft and as main drive belts, but more commonly designated as "single and double thickness." The reason for this distinction is that while the stress in the net solid body of the leather is taken to be the same in each case, in "single" belts the joint is a butt joint and is laced. This cuts away more of the belt than where the belt is of double thickness, lapped and cemented, or riveted; the difference being in the character of the joint rather than in the strength of the belt.

2 For comparison then, we can take Mr. Barth's estimate of the relative strength of these belts, as 160 to 240 or 1.0 to 1.50. Mr. Towne found these to be as 1.0 to 1.82, and in my early studies, I was inclined to adopt this ratio; later, however, I have used the ratio of 275 to 400 or 1.0 to 1.45.

3 The belt problem is very far from being one of pure mathematics. As in most engineering problems, there is about 5 per cent of scientific knowledge involved, and fully 95 per cent of good judgment based upon experience. We rarely know the exact power to be transmitted except in the case of prime movers. The arc of contact, the velocity, and the stress we are willing to put upon the leather, are all easily determined, but we cannot decide upon the coefficient of friction by formula. A new leather belt upon an iron pulley may not have a coefficient of friction of 25 per cent, while the same belt, well worn and well groomed, will give 65 per cent in a clean, dry room; put the same belt in a wet place, like a tannery, or a dusty place, like a stone-crushing plant, and we have an entirely different coefficient.

4 It seems to me that the designing engineer, even though he understands the mathematics of the belt problem, if ignorant or unappreciative of the practical conditions under which the belt works, will be liable to make a mistake. On the other hand the engineer familiar with the conditions, but ignorant of the mathematics involved, is also liable to error in his conclusions. A cautious man will endeavor to err on the safe side, feeling no doubt as did our venerable ex-President John Fritz, who when remonstrated with for making some machinery needlessly strong, replied, "If I do, nobody will ever find it out."

5 On general principles, it is of course desirable to work belts, like other members of a machine, with large coefficients of safety, but engineering in its last analysis is a question of finance and we must "hew as close to the line" as possible. Mr. Towne found the ultimate strength of laced belts to be 200 lb. per inch width ($\frac{7}{32}$ in.) thick, and used $\frac{1}{3}$ of this, or $66\frac{2}{3}$ lb. as a safe working stress. Mr. Towne also found a coefficient of friction of 42 per cent to be safe. The general practice of the day has been quite close to these factors, but if I understand his diagrams correctly, Mr. Barth has departed far from them.

6 In 1881 I read before the Society a paper giving for the first time, I believe, a belt formula which took cognizance of the effect of centrifugal force. The data used therein were based principally upon Mr. Towne's experiments. The results obtained were well within the safe limits of previous practice for low speeds, but at high speeds my formula showed the deviation. Common formulæ gave results (see Kent, Mech. Eng. Pocket Book, p. 879) as follows:

For single belt 1 in. wide,

Velocity	(1)	(2)	(3)	(4)	Nagle
600 ft. per min.	1.09 h.p.	0.55 h.p.	0.60 h.p.	0.82 h.p.	0.73 h.p.

Barth gives only 0.40 h.p.

For double belt,

Common Formula	Nagle	Barth
1.17 h.p.	1.24 h.p.	0.68 h.p.

7 For the purpose of giving a clear conception of Mr. Barth's deviation from the others, I repeat my formula here:

$$h.p. = \frac{C.\ V.\ t\ w\ (S - 0.012\ V^2)}{550}$$

C is a constant expressing the adhesion of the belt upon the pulley under a unit of stress of belt. Its value is expressed by the equation

$C = 1 - 10^{-0.00758fa}$ where a is the arc of contact and f the coefficient of friction. The other quantities are as follows:

V = velocity in feet per second.
S = stress upon leather per square inch, which I have taken at 275 lb. for laced and 400 lb. for riveted belts.
t and w are the thickness and width respectively in inches.
550 ft. lb. = horse power per sec.

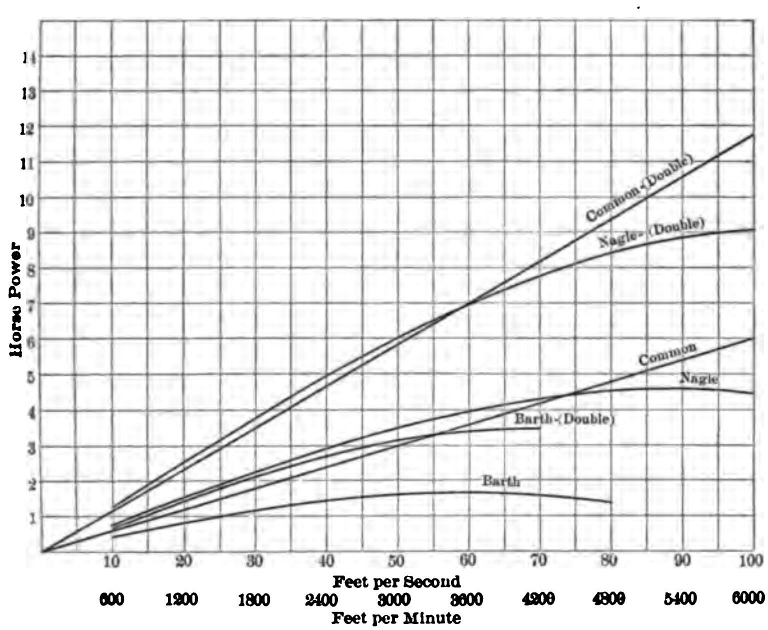

Fig. 1 Comparison of Different Belt Formulæ, Based Upon Belts 1 In. Wide and $\frac{7}{32}$ In. Thick for Single and $\frac{1}{4}$ In. Thick for Double Belts

8 To illustrate the solving of this equation, let a = 180 deg. and f = 0.40, then

$$180 \times 0.40 \times 0.00758 = 0.54576$$
$$10^{-0.54576} = \log 10 \times 0.54576 = 1 \times 0.54576$$

0.54576 is a logarithm of which 3.513 is the number. This being a minus-coefficient, we must take its reciprocal or 0.284; subtracting this from 1, we get 0.716. The result could have been obtained by subtracting the log 0.54576 from 1, giving 1.45424, and this gives 0.2846 as its number direct.

9 In Kent's Mechanical Engineering Pocket Book, p. 878, tables are given based on this formula, which facilitate its use. I wish to call attention to the wide divergence of Mr. Barth's conclusions from those commonly used as well as from my own, as plotted in Fig. 1 herewith. I have reduced his figures to the same thickness as mine, namely $\frac{3}{16}$ in. for single and $\frac{1}{4}$ for double.

10 This work has been done somewhat hastily and I hope the author will check it at least so far as relates so the interpretation of his diagram. If my work is correct, I am puzzled to understand why his tables of belt horse power differ so much from mine.

Prof. Wm. W. Bird. I feel very much pleased and highly complimented to see the results of Mr. Barth's mathematical analysis of my experiments on belt creep. On a few points, however, I am still in doubt in regard to his general conclusion. I believe:

a That the elasticity of a belt varies with the velocity and that at very slow speeds the sum of the tensions would remain constant, while at high speeds, if it were not for the centrifugal force, the sum would increase practically with the load. If this is true, a belt will improve with higher speeds and will not reach a maximum at 4000 ft. per minute as shown in Fig. 3.

b That in determining the size of a belt for a given load, the diameter of the smaller pulley should be considered. A belt will do relatively less on a small pulley than on a large one, other conditions being the same.

c That the crowning of the pulleys should be considered as it affects the life of the belt.

d That the carrying capacity of a belt should not be given in terms of square inches of cross section, as a double belt with one square inch of cross section will not transmit as much as a single belt of the same cross section.

2 I have recently made some tests on compound or rider belts and have been somewhat surprised at the relative movements of the main and rider belts with various pulley ratios, from which I have concluded that if these belts were glued together, as they would be in a double belt, numerous internal stresses and strains must be set up when the belt passes over a pulley, especially over a small one.

3 The crown is a very serious matter on a small pulley, as the following figures will show. Take a 4 in. pulley with a crown of 0.2 in. in the diameter, if the belt wraps 180 deg., the length in the

middle of the belt will be 0.31 in. greater than on the sides; this means a stretch of 0.05 in. per inch or 1000 lb. stress on middle fiber, taking modulus of elasticity as 20,000 lb. The belt must slip or be ruined, for this stress does not include load or initial tension and is in itself enough to stretch the belt beyond the elastic limit. The slipping necessary to adjust this stress must influence the friction and slipping of the belt as a whole.

4 I would like to have Mr. Barth answer a question which I have been asked a great many times,—why does the sum of the tensions in a belt increase with the load? I would also like to have him calculate with his slide-rule the size of a belt for the following conditions: 20-h.p. motor, 6-in. pulley, 1200 r.p.m., to drive a shaft 12 ft. away at 200. Would the same belt last as long if the drive were reversed, that is, a shaft running at 200 r.p.m. driving a generator with 6 in. pulley at 1200? I would like also to ask Mr. Barth or any engineer present whether he would use the same size belt with the 6 in. pulley as a driver as with it as a driven, and with the same size of belt; and whether in this case it would last longer, other conditions being equal.

5 Anyone who has undertaken an investigation of the belt problem knows that it is almost impossible to keep conditions constant—humidity, oil in the belt, surface of pulley, etc., seem to change without notice and complicate the work.

6 I wish to congratulate Mr. Barth on his efforts to advance the theory of the transmission of power by leather belting, and to agree with Mr. Lewis in the conclusion of his paper presented in 1886, "That there is still need of more light on the subject."

PROF. C. H. BENJAMIN. I have been asked to contribute to the discussion of Mr. Barth's paper; technically, I am afraid I can not criticise it or add to it, for it leaves but little more to be said. Sentimentally, I can not but regret the gradual disappearance of our *terra incognita*, both geographical and mechanical. Time was when large areas on the map bore the encouraging legend "Unexplored Wilderness" or "Great American Desert" and left room for the free play of the imagination. Today you miss those fascinating areas and are tied down to realities.

2 Not many years ago, the grinding of a lathe tool was an interesting experiment, attended with much uncertainty, and the matter of feeds and speeds offered an alluring field for investigation. Mr. Taylor has spoiled all that for us and now our imagination is worked by slide-rule.

3 Time was when the possibilities of belting were vague in outline and when coefficient of friction, slack tension and belt creep were rather shadowy phantoms. It was pleasant then to speculate on what the belt would do and how long it would do it and the man with the longest memory had the advantage. But now, Mr. Lewis, Mr. Bird and Mr. Barth have taken all the romance out of it and another illusion succumbs to the deadly aim of the slide-rule.

4 Perhaps I take a malicious pleasure in noting that one or more factors of the problem are still out of harness and a trifle intangible. Our old friend, the coefficient of friction, is in hiding under the belt sporting with those other elusive fairies, modulus of elasticity and belt-creep. After all, what does it matter? Aside from the interesting theoretical questions involved, what we need to know is, first, how wide a belt to use at a certain speed to transmit a certain power,—Mr. Taylor has answered this question. Second, how tight to lace or cement that belt that it may do the work for a reasonable time without relacing,—Mr. Barth has told us that.

5 I began experimenting on belts 25 years ago and have been at it more or less since. With a fixed pulley and a slipping belt, I found no difficulty in proving $\phi = 0.42$ after Rankine, but when I built a belt machine and tested belts under running conditions, ϕ lost all its constancy and might as well have been called x. Working backwards from the measured tension and using the old formula, I found ϕ to vary with the load, the speed, the kind of pulley, the age of the belt, the weather and the dominant political party—in fine, to be so mysterious and intangible a quantity as to be useless for practical purposes.

6 The sum of the tensions also varied in a manner that did not admit of rational explanation. And right here let me say that the reasons for Mr. Barth's assumption of constancy for $(t_1 + \frac{1}{2} t_2)$ are hardly clear to me. Why call that constant which is not constant? Why call anything constant except as it is shown to be so by measurement? This is not said in criticism but for the sake of information.

7 There is one aspect of the paper that deserves special attention and that is the recognition of the fact that a belt is an elastic connector with a variable length and variable tensions. Most writers on the subject have treated belting as if it were a non-extensible element which could be exactly represented on paper and whose behavior was capable of exact mathematical analysis. A belt in action is almost like a thing alive, squirming, lengthening, shortening, its tension changing back and forth with a variable modulus of elasticity

and a lag in its changes due to its contact with the pulley and the short time intervals. A belt must be tested to be appreciated and theory must wait upon experiment.

8 I fully appreciate the value of Mr. Barth's analysis and can see that his methods will result in marked economies in establishments where many large belts are used and where conditions are pre-determined. In the smaller shop, where conditions vary, and in isolated cases with differing sorts of pulleys, differing kinds of belt, new and old, I feel that each case will have to be settled on its own merits. Until more experiments are recorded, the average machine-designer or millwright will have to be guided largely by his own judgment and experience in determining the width and tension of each belt. Let us have more experiments.

H. K. Hathaway.[1] To the scientist and machine-tool designer the value of Mr. Barth's paper will unquestionably be immediately apparent, but the writer feels that the paper does not represent with sufficient clearness features of the problem that are of inestimable value to the engineer concerned with running a shop. Assuming that the designer takes care of the sizes of belts required, and the speed at which they should be run, in accordance with the conclusions of such eminent authorities as Mr. Taylor, Mr. Barth, and Mr. Lewis, a great deal is lost unless the shop-man properly cares for the maintenance of such belts. As Mr. Barth has pointed out, the care and maintenance of belting in the great majority of shops is done by rule of thumb, and left entirely to the judgment of the shop millwright or the workman operating the machine.

2 The efficiency of a belt-driven machine largely depends upon the tension of the belts being properly maintained at a point above the minimum initial tension at which they will transmit the power required. This point Mr. Barth has only slightly touched on, whereas the writer feels that this subject should have occupied a section fully as large as the body of the paper presented.

3 If a machine stands idle during working hours while the belt is being repaired or tightened it produces nothing during that time, and there is a distinct loss to the manufacturer. If a machine stands idle for one-half hour out of ten hours working time there is a loss of 5 per cent in the output of that machine, and if in a shop having 100 machines, 10 machines out of the 100 lose one-half hour each day

[1] H. K. Hathaway, Tabor Mfg. Co., Philadelphia, Pa.

on account of repairs to belts it amounts to a loss of 0.5 per cent on the total output of the shop. This feature, however, is probably not so bad as the loss in output due to the machine belts being run so loose that they cannot begin to take the feeds, speeds, and depths of cut for which the machines are designed and that the tools will stand.

4 The writer has had considerable experience with the system of maintenance of belting mentioned in Par. 67 of Mr. Barth's paper, and will describe it briefly.

5 Almost every foreman or superintendent, in attempting to bring up the speeds of his machines to something like what he knows to be possible, has found that such attempts usually result in the belt's slipping or breaking, or the lacing giving out, and knows that where the care of belts is left to the man on the machine, only in a very few cases can the belts be depended upon to do the maximum amount of work. If, therefore, the maximum feed, speed and depth of cut are to be prescribed and used, as is done by the aid of Mr. Barth's slide-rules under the Taylor system, it is essential that belts of the best quality and of the proper proportions be used, and that they be kept in first-class condition and at the proper tension, so that they can be relied upon to give the pull required. It is also necessary that all repairing, tightening, and inspection of belts be done outside of working hours that there may be no loss of output from interruption to manufacture. In order to accomplish these objects the following system has been evolved.

6 A record is kept for each belt in the shop on the form shown as Fig. 1, on which are given all standard data for each belt in question.

7 When a new belt is to be put on, or an old belt to be inspected or tightened, the special belt fixer's bench developed by Mr. Gulowsen is used, together with the belt-tension scales referred to by Mr. Barth. With this apparatus it is possible for one man to remove, tighten and replace almost any belt in from six to eighteen minutes. In putting on a new belt, or tightening an old one, the drums or pulleys on the belt bench are set by means of a steel tape to correspond with the distance over the actual pulleys, as previously determined, and shown on the belting record as "Length over Pulleys." A roll of belting, of the proper width and thickness, is next placed in the open drum, and passed through one pair of clamps of the belt scales around the drums or pulleys and through the other pair of clamps of the belt scales. The clamps are then tightened on the belt and the belt drawn up by means of the screws until the spring

BELTING DEPARTMENT.............

BELT SYMBOL

Location..	Length over Pulleys........................	Max. Velocity..............
Purpose..	Width..................................	Min. Velocity..............
Kind..	Thickness................................	Max. Tension.............. } ON EACH SPRING BALANCE
Maker's Name................................	Cross Section in sq. in........................	Min. Tension..............
Date Put in Use................................	How Fastened............................	Cost of Belt........................
Date Taken Out................................	Belt Grease to be used........................	Cost of Maintenance..................
Life of Belt................................	Belt Dressing to be used......................	Total Cost........................

Date Re-Tightened	Tension in Pounds Indicated by Each Spring Balance		Amount Taken Out	Date Tension Should be Taken Again	Date Cleaned and Greased	Date to be Cleaned and Greased Again	Date Dressing put on During Service	General Remarks about Work Done on Belt, Condition of Belt, etc.	Workman's Name	Man's Time Required to do Work	Cost of Work
	Before Tightening	After Tightening									

Fig. 1 Belt Record for Individual Belts

balances between the two pairs of clamps record the tension required, after which the belt is cut off so that the two ends will come together, and the belt is laced on a belt-lacing machine and put on its pulleys.

8 A memorandum, which also serves as the belt fixer's order and time card, giving him all necessary instruction, is then placed in what is called the "tickler," a portfolio having a compartment for each day of the year, under the date on which the belt will probably require re-tightening, and on that day it will be removed from the tickler, together with the memoranda for any other belts requiring attention, and sent to the belt fixer for attention during the noon hour and after quitting-time.

9 These belts are then removed from their pulleys, taken to the belt bench and tested to ascertain whether they require tightening; if the tension is found to have fallen to approximately the minimum, they are drawn up to the maximum tension as previously described. a piece is cut from one end, the belt is re-laced and put back in place and these facts are noted on the belt fixer's memorandum, which is then returned to the planning department, and entered on the belt record; and a new memorandum placed in the tickler under the date on which the belt will again require attention. Notices for scraping, cleaning and greasing the belts at proper intervals are also placed in the tickler.

10 The length of time a belt will run before the tension will fall to the minimum at which it will pull all that is required, has been determined from experiments, and a belt seldom requires attention before the time set for re-tightening; when this does occur, however, a belt-dressing which does not injure the belt, but which will enable it to pull properly until noon or the end of the day, is applied, and the memorandum is removed from the tickler and another placed under its next date for re-tightening.

11 The system described accomplishes four things of vital importance to economical production:

- *a* Freedom from interruption to production from having to repair belts during working hours, by having all belts systematically inspected and all breakdown and slippage anticipated and prevented before they occur.
- *b* Possibility of using the maximum feeds, speeds and depths of cuts at all times.
- *c* Increase in life of the belt owing to all belts being of the proper dimensions and properly laced and spliced and run at the proper tension.
- *d* Reduction of cost of maintenance to a minimum.

12 Mr. Barth's belting slide-rule is used in determining the dimensions of the belts, the maximum and minimum tensions. The writer can speak from experience of the great value of the belting slide-rule in solving the belting problems that confront the shop engineer, and while the mathematical features of Mr. Barth's paper

Out In		Order Number D L
Department Day Rate..............................		Man's Time............................
Max. Tension	Min. Tension	BELT SYMBOL

Cleaned and Greased............................Grease Used........................

Dressed While in Use ....Dressing Used......................

Amount Taken Out.............................Length Put In......................

Length of Splice...............................Cement Used.......................

Tension in Lbs. Indicated by Each Spring Balance { Before Tightening.............................. After..

Workman's Name...Man's No.........

Entered in		
Pay Sheet	Cost Sheet	Belt Record

....

DAY WORK time note

FIG. 2 BELT FIXER'S ORDER AND TIME CARD

are unquestionably interesting to many, the writer feels that, like himself, many will be glad to accept Mr. Barth's figures without question provided they can have the slide-rule.

13 It is a fact that in the average shop very few belts become unfit for use through legitimate wear, but rather through accidents or improper care. Where the care of the belts is left to the workman, the belts are usually far too loose, and when a belt slips it is less

trouble for the workman to reduce his speed, feed, or depth of cut, or as a last resort to use rosin to make the belt pull. This use of rosin will ruin any belt in a very short time.

14 Very few machinists or even foremen know how to tighten or lace a belt properly, the amount to be taken out being usually guessed at, and a great deal of time is lost through the machine's standing idle while the cutting and trying is going on. The writer has seen a good machinist run a cone belt, which he had made too tight, on "cross cones," i. e., on steps not in line with each other, with the result that it twisted itself up like a corkscrew and was practically ruined.

15 Another cause of premature ruin of belts is improper lacing, the ends not being cut square and the lacing on one side stretching more than the other, causing the belt to run crooked.

16 Cemented splices, when properly made, give the best results. Machine lacing, using a spiral wire lacing, while not so good as a cemented splice, is very satisfactory, however, and more convenient, and takes less time for putting on and taking off belts for the purpose of testing and tightening on the belt bench. A belt joined by a cemented splice must be tested and spliced in position, which is not so convenient as on the belt bench, especially in the case of overhead belts. Even where cemented splices are used the belt bench is convenient for cutting new belts, or re-tightening to a length giving the proper tension, and for repairs. Only one wire joint is used in any belt, splices being made if a section becomes damaged so that a new piece must be set in. The average belt if cared for under this system will last from six to eight years.

17 The tension on a new belt falls very rapidly, and our present practice is to tighten it after 24 hours, then in 48 hours, then in one week, then in two weeks, and so on doubling the length of intervals until it gets to three months; from this point we must ascertain by trial for each belt how much greater the intervals may be. This of course depends upon the severity of service the belt is called upon to perform as well as the quality of the belt.

Prof. Wm. S. Aldrich. In the first place, the academic discussion of the constancy of the sum of the belt tensions under all loads is finally set at rest. Now that we really know what is what, by the invaluable series of experiments referred to, the wonder is that this fallacy of the constancy of the sum of the belt tensions is so persistent.

2 It was doubt of this position that led the writer to analyze for himself the experiments on belting then available, those of Wilfred Lewis and J. S. Bancroft, undertaken for Wm. Sellers & Co., and of Professor Lanza, of the Massachusetts Institute of Technology. Both of these were recorded in papers read before the Society, and published in Vol. 7 of the Transactions. It is remarkable that these classic experiments have been before the world thus long, and yet so little studied and respected, and, as far as the writer is aware, have not been superseded by experiments in their special field with more modern apparatus. Until they are superseded, Mr. Barth's conclusions must stand, a remarkable instance of the deductive reasoning by which it would seem that engineering progress must be made.

3 On the other hand, Mr. Barth has built up, in characteristic fashion, from theoretical considerations more or less influenced by a knowledge of the phenomena of belt-transmission, combined with the physical properties of belting, certain new and helpful relations that must govern in the future. Such is his "new theorem of the relations of the tension in a belt," that "under any variation of the effective pull of a belt, the sum of the square roots of the tensions in the two strands remains constant, as against the old fallacious supposition that the sum of these tensions remains constant." (Appendix, Par. 26). Therefore,

$$\sqrt{T_1} + \sqrt{T_2} = 2\sqrt{T_0} \qquad \text{(1)}$$

4 Now, if we can obtain a similar relation for the difference of the square roots of the tensions, then we shall have at once, by the usual formula for the product of the sum and difference of two quantities, the difference of their squares; that is, in this case, the difference of the squares of the square roots of the tensions which is the difference of the tensions, or the pulling power sought.

5 This much needed "difference of square roots of tensions" has been indicated by Mr. Barth (Appendix Par. 14), "on the strength of the experiments made by Mr. Lewis and himself, namely, that within the limits of ordinary working tensions of a belt, the difference between the lengths of a belt at different tensions is proportional to the difference between the square roots of those tensions." We thus have,

$$L_1 - L_2 = K(\sqrt{T_1} - \sqrt{T_2}) \qquad \text{(2)}$$

in which K is a constant, dependent upon the material of the belt, and determined by experiment on the belt.

6 Combining with Equation 1, we have, as already indicated

$$T_1 - T_2 = 2 \sqrt{T_0}\,(L_1 - L_2)\,\frac{1}{K} \qquad\text{(3)}$$

It seems to the writer that this might possibly be a helpful deduction, though it may be without much practical application; so that knowing the initial unit tension T_0 and the lengths of the belt under the tensions T_1 and T_2, together with the constant K, its pulling power $(T_1 - T_2)$ is known. It seems, therefore, necessary to know the difference in the lengths of the belt, due to differences in the belt tensions, that is, to the different driving powers under which it is expected to operate the belt, or in other words, to calibrate the belt-performance for this use.

7 It may be remarked, in passing, that the constant K is to be found from the experiments of Mr. Lewis, as analyzed by Mr. Barth (Appendix, Equation 3),

$$L_t = L\left(1 + \frac{\sqrt{t}}{864}\right) \qquad\text{(4)}$$

in which L_t equals the length of belt under the unit tension t when its slack length is L. From this, by analogy with the above Equation 2, we have,

$$K = \frac{L}{864} \qquad\text{(5)}$$

8 It will no doubt appear that the writer is still inclined to let the arc of contact and the coefficient of friction of belts take care of themselves, notwithstanding the keen discussion that has centered about the fourth conclusion in his paper, referred to by Mr. Barth; namely,

> "(4) The ratio of the tensions of a belt-transmitting power cannot be calculated with any degree of accuracy by means of the well-known belt formula:
>
> $$\frac{T_1}{T_2} = e^{\phi\alpha} \qquad\text{(6)}$$
>
> involving the arc of contact α and the coefficient of friction ϕ."

9 This relation is no doubt a guide and a help, indicating the way the ratios of belt tensions are most likely to be involved. But

it certainly requires a radical modification to adapt it to any reliable use in predetermining the ratio of tensions for lacing up belts for given pulling power. Mr. Barth has wrought out these modifications with excellent results, judged by the adaptability of his slide rules, and the closeness of approximation to actual conditions (within the limits assigned) of the assumptions upon which they are based; namely (Par. 44), "that for the driving belt of a machine the *minimum initial tension* must be such that when the belt is doing the maximum amount of work intended, the *sum of the tension on the tight side of the belt and one-half the tension on the slack side will equal 240 lb. per square inch of cross section for all belt speeds;* and that for a belt driving a countershaft, or any other belt inconvenient to get at for re-tightening or more readily made of liberal dimensions, this sum will equal *160 lb.*"

10 Here, then, is a definite acceptance of things as they are, and a straightforward assumption involving additive relations of belt tensions of leather belting, as it is made and used, and conformable to experience, rather than their ratios agreeable to theoretical formula, involving coefficient of friction and arc of contact. This latter relation (6) is as elusive as the traction-coefficient in railroad work; and engineers probably will have their own opinions about each until some genius can predetermine what coefficients of friction are to be expected in every instance, and so properly introduce the friction for dynamic conditions into a formula based entirely upon a consideration of statical relations.

THE AUTHOR. In reading the unexpectedly numerous discussions of this paper, the author is pleased to note the general appreciation of it as a contribution to the literature of its kind, but regrets the assumption by two or three of the discussors that he considers the paper final in its application of the theories developed. All that is claimed is that he has taken practical advantage of the experimental data at his disposal, and has taken the pains to do mathematical justice to them, deriving therefrom excellent results in the scientific running of machine tools whose belts have been tightened and worked according to the rules thus established.

2 While the author feels guilty, therefore, of narrowing to a considerable extent the scope within which Professor Benjamin's imagination may still run rampant, so far as the behavior of a leather belt goes, he fully agrees that further experiments are needed in order to determine the coefficient of friction under all the variable conditions

under which belts are called on to drive; and yet more, in order to settle conclusively whether the coefficient of friction is a function of the velocity of slip, as he has assumed, or of the percentage of slip, as indicated by Mr. Hamerstadt's study of numerous experiments at different belt speeds, though the latter seems contrary to the mechanical principles involved in the phenomenon of slip.

3 For the special benefit of Professor Bird, the writer will even say, that while all he knows about belting could probably be reduced to a pamphlet three times the size of his paper, a good-sized volume would probably be required to hold all he does not know but would like to know about belting, and a small library would be required to record all he does not care a straw to know about the subject.

4 But while the writer agrees with Mr. Hamerstadt as to the desirability of further experiments and will look forward to these with the keenest interest, he does not see the force of his argument about the necessary overload capacity of a high-speed belt, on a motor with an overload capacity; surely we need only make the belt big enough to take care of the overload as a normal load, and be satisfied to have it unnecessarily large for the rated capacity of the motor; just as a bridge intended for a light normal load must still be made strong enough for any anticipated occasional extra load. Trouble arises only when we do not know how to design a bridge properly, or when we get an occasional extra load which we have had no reason to anticipate.

5 Though the writer had not expected to be forced to express himself on the question of belt-drives *versus* electric motor drives, he will say, in view of Mr. Robbins' remarks, that he believes that during the past decade hundreds of thousands of dollars, if not millions, have been more than wasted by the substitution of motor-drives for belt drives. Such a change has often been advantageous, of course, and is occasionally recommended to his clients by the writer: the trouble has been that the enormous investments of electrical manufacturing establishments have forced the electrical salesman more than any other to create a demand for his product, so that not only has he allowed his enthusiasm to run away with him, but he frequently has recommended his product against his own biased judgment; persuading the incompetent shop manager or superintendent to accept his product as a remedy for a small output that is in reality due to a complication of causes that could be cured only by the application of a number of remedial measures.

6 The writer believes, however, that a reaction against this indis-

criminate electrification of machine shops has already set in, aside from the influence of the industrial depression, and that the electric drive will be installed, in the near future, only when conditions make it unquestionably more advantageous than the belt-drive.

7 As touched upon by Mr. Van Derhoef, the elastic properties of transmission-rope are probably similar to those of leather belts, and it seems to be in order for someone to ascertain them by the necessary experiments, and subsequently to apply this knowledge by use of the writer's methods.

8 The writer values Mr. Allen's statements of the advantages derived from the adoption of the Taylor system, as introduced by the author in the Yale & Towne Mfg. Co.'s plant, where 4800 belts are thus taken care of. Mr. Allen, in conjunction with Messrs. Taylor, Hathaway and Waldron, has thus supplemented the scant attention paid in the paper to the aspect of the subject most practically important. The reason for this omission is that in his work with belting the writer has derived by far the greatest personal satisfaction from the solution of the mathematical problems involved, and he has been unable to eliminate entirely the personal interest.

9 It is not possible to answer here Mr. Cary's question as to how to estimate the horse-power required to drive each machine in a large plant, but the writer will be pleased to give him, and anybody else interested enough to pay a visit to Philadelphia, an idea of how it is done, by means of slide-rules especially constructed for the purpose.

10 As a further answer to Professor Bird's various statements and questions, the writer will only say that on a more careful reading of the paper, as well as the Appendix and Supplement, he will find most of them answered. For instance, the most valuable mathematical developments in the Appendix and Supplement answer the question why the sum of the tensions of a belt increases with the load; and study of this will help him to formulate for himself an answer to his non-mathematical questioners. As to the effect of crownings a mall pulley, the writer heartily agrees with Professor Bird in a general way, though surprised to note with what confidence the latter estimates the difference between the tensions in the middle and edge-fibers of a belt running over such a pulley.

11. The author is sorry that the considerable trouble to which Mr. Nagle has gone to make comparisons with his own earlier formulae for the horse-power transmitted by leather belting, is based on a misunderstanding of the fundamental basis of the author's work.

12 As stated in the paper, the author bases his figures on a certain

tension per square inch of belt, independent both of the strength of the belt itself and its thickness, and of the strength of the lacing, except that the latter must be in excess of the maximum tension brought to bear on a belt while delivering power. The author, therefore, makes no distinction between a single and a double belt, but merely considers the tension per square inch of section, as it has not been definitely proven that the coefficient of friction depends materially on the area of the surface presented by the belt against the pulley.

13 As Mr. Nagle somehow has assumed that the two horse-power curves in Fig. 3 are meant respectively for a single and a double belt, whereas they stand for something totally different, it unfortunately follows that the comparisons made by him of his own and the author's ideas as to what power a belt should be counted on to transmit, have completely miscarried.

14 Mr. Nagle says that we cannot decide upon the coefficient of friction by formula. This is unquestionably so, but it is also true that, having roughly decided, by one means or another, what we wish to count on as the coefficient of friction at any one velocity of a belt, we may to great advantage make an empirical formula to represent a perfectly-graded change of this coefficient with the velocity; and only by so doing can we effect a mathematical solution of the belt problem that is an improvement on the unquestionably wrong assumption of a coefficient independent of the velocity of the belt, such as 0.42, originally recommended by Mr. Towne, or 0.28, recommended by the late Professor Ruleaux.

15 The author fully agrees with Mr. Nagle that "a new belt on an iron pulley may not have a coefficient of friction of as much as 0.25, while the same belt, well worn and well groomed, will give 0.65 in a clean, dry room;" and, more than that, knows that this elusive quantity will vary all the way from almost 0 to 1.50. However, just because the author is a practical and practicing engineer, though very fond of a little pure mathematics in the handling of practical engineering problems, he has adopted *something* as a standard, this something being a variable lying happily between the great extremes, instead of being merely a single average between the extreme values.

16 The author is not at all disappointed because a perfectly new belt will not give the output required, at its minimum tension, without the resort to a temporary application of some good adhesion-producing belt-dressing; nor on the other hand, when a "well worn and groomed" belt at times is capable of giving the output required, at a little less initial tension than the one he aims at maintaining by the

means more fully described in the discussions submitted by Mr. Allen and Mr. Hathaway.

17 Mr. Nagle also remarks that we rarely know the exact power to be transmitted except in the case of prime movers, which no doubt is true, so far as the work of most engineers is concerned. However, in the author's practice at least, the maximum output of every belt put up on any machine is known; simply because he personally sets the limit, and has means of seeing that the same is never exceeded.

18 Mr. Nagle refers to his paper read in 1881 as the first one to recognize the effect of centrifugal force in a belt. A correct formula, however, for the loss of effective tension in a belt, due to its centrifugal force, was given by Weisbach, at a much earlier date. This fact does not detract, of course, from the value of Mr. Nagle's paper, in which, probably for the first time, this matter was presented in a manner that made it readily available to the busy, practical engineer.

19 As regards Mr. Nagle's suggestion that the data have not been presented in a sufficiently handy form for the busy engineer, the author believes Mr. Nagle has failed to appreciate the slide-rule illustrated in Fig. 5, which contains these data in a form which for handiness leaves tables and diagrams far behind, while he at the same time is not ready to admit that there is anything the matter with the various diagrams that give the same data.

SAFETY VALVES

No. 1231

SAFETY VALVES FOR LOCOMOTIVES

By Frederic M. Whyte, New York

Member of the Society

It is the purpose of this paper to present some ideas about safety valves for steam boilers and particularly for locomotive boilers.

2 Just how the capacity of the first valve used on a steam boiler was determined, or what relation this capacity had to the generating capacity of the boiler, may be recorded somewhere in history, but it is doubtful if either fact was recorded or even determined. So far as locomotive work is concerned, the same ignorance prevails today; but there is good promise that this ignorance will soon be dispelled. In marine work certain formulae have been devised for calculating the sizes of safety valves, and these formulae have been accepted, more or less blindly, it is thought.

3 The general practice in locomotive work has been to determine in an "offhand" way the size and number of safety valves to be used, and former practice has guided the determination entirely. If a larger boiler is to be used the valve capacity may not be increased, depending upon the judgment of the person whose duty it is to determine the capacity. Again, the capacity has been indicated in an indifferent manner, being expressed as a "size," meaning the diameter of something more or less uncertain; while the other dimension, the lift, which is necessary to give an indication of the capacity, is entirely ignored.

4 But to know the exact capacity of the available valves is not sufficient; it is quite as important to know how much steam is to be released and in what length of time it should be released. It will be comparatively easy to determine the capacity of safety valves, if indeed the elaborate tests which have already been made—data

Presented at the New York monthly meeting (February 1909) of The American Society of Mechanical Engineers.

from which it is hoped may be presented in the discussion[1]—have not already solved part of the problem; more difficult will be that part of it which is concerned with the quantity of steam to be released and the rate of the release. The subject is of mutual interest to the valve manufacturer and the user,—the design of the valve for capacity and wear to be worked out by the manufacturer; and the capacity which is to be used, both in volume and in number of valves, and the rate of release, to be determined by the user with the assistance of the manufacturer.

ESSENTIALS OF A SAFETY VALVE ON A LOCOMOTIVE

5 The design of the valve will include the diameter of the controlling opening and the passages leading to it from the steam volume, as well as those leading from it to the atmosphere, the shape and material of the seat, the amount of lift of the valve, and the shape and material of the valve face, the spring and its protection, the adjustment, the muffler, if one is to be used, and the action of the valve in lifting and in seating.

6 It will not be necessary to discuss the diameter of the controlling opening, and of the passages to and from it, in view of the suggestion here made that instead of indicating the capacity of a valve in a very rough way by the diameter of some opening, the capacity be expressed in pounds of steam at certain pressures. The shape of the seat and of the valve face may or may not be of importance; but this will be referred to again. The material in the seat and face will naturally be that which will best withstand the effects of the flow of steam over them, and the possible pound of the valve when seating.

7 The reliability of the spring and the effect of heat upon it are very important points. Adjustments should be readily made, but on the other hand to get out of adjustment should be practically impossible. The capacity of the muffler need not be questioned, except in extreme designs, but the indicated capacity should be that of the valve complete, with or without muffler, according to the intended use of the valve; then it is important only that it deaden sufficiently the noise of the escaping steam.

8 The action of the valve in lifting and in seating, the desirability of a forewarning that the maximum pressure is about reached, and the operating conditions which bear upon this question of fore-

[1]These data are given in the paper "Safety Valve Capacity" which follows. They were presented as a discussion and afterwards published in The Journal as a paper under the above title.—EDITOR.

warning, are correlative. With any kind of steam-generating plant it ought to be quite sufficient if those immediately responsible for the quantity produced, and for its use, know what is available; in stationary and in marine work this is generally true, and steam gages can be placed within view of those who should know what the pressure is at any time. Unfortunately in locomotive work, however, it has become perhaps desirable that others than those within view of the gage in the cab know something about the steam pressure, and inasmuch as the fireman is willing, and sometimes anxious, that they should know, he takes the only means at hand to inform them when he thinks that the results of his labors are good, and "fires against the pop" so that everybody within hearing or sight of the valve knows by the escaping steam that the fireman is doing his duty. If when a train is ascending a grade the conductor at the rear sees steam escaping from the valve he knows the train will get up the grade; on hard grades he will watch for the only indication which can be given him, and the fireman tries to present this indicator, the escape from the valve, the "white feather."

9 Numerous similar examples might be mentioned, but assuming that such an indicator of steaming conditions has grown to be a necessity, undesirable as it may be, how can it be produced at the least expense? Surely not with a valve from 2½ to 4 in. in diameter and open to its full capacity. Two devices, at least, are available to give the indication at a lower cost than the full open valve: the "simmering" valve, which will open slightly for two or three pounds about the normal maximum, then open full, and just reversing this order in seating; the other, the small pilot valve, which will open at two or three pounds pressure below the working valve. The first method will have some bearing on the kind of metal to be used in the valve seat and valve face and possibly upon the shape of the exterior edge of the valve and the opposing surface of the seat. The second method means the addition of the small valve, an additional cost for which there will be no need if the first method can be developed successfully.

RELATION OF VALVE CAPACITY TO STEAM-GENERATING CAPACITY

10 There remains for consideration the relation of valve capacity to steam-generating capacity, and the unit capacity of the valves which will make up the total valve capacity. The fact that in locomotive work the total valve capacity has not been as great as the maximum steam-generating capacity should be ample proof that

such valve capacity is not necessary. The reason for this is, of course, that on account of using the exhaust steam for producing the forced draft, when the demand for steam from the boiler is reduced or entirely cut off, the forced draft is automatically reduced or cut off, and the generating capacity is reduced so that it is not necessary that the safety valves release the full generating capacity. Probably it will be largely a question of opinion what per cent of the total generating capacity the valve ought to have, although it is possible that as attention is centered upon this question some more or less positive solution of it may result.

11 Having fixed upon the per cent of the generating capacity to be provided for in the valves it will be necessary to determine the desirable unit capacity of the valves. Some States require that each locomotive boiler have at least two valves. Starting with this condition, consideration of the maintenance of the valves indicates that they should be duplicates and therefore that each have a capacity equal to one-half the required generating capacity. If a number of boilers of different capacities are to be considered, then the smaller ones would probably be provided with the same valves as the larger ones for the purpose of duplication. There are some large boilers for which three valves might be necessary, because the necessary capacity in two units might make the valves abnormally large for construction purposes. Also it is worth while to consider whether undesirable results would come about from opening almost instantaneously an escape of steam from the boiler to the atmosphere. No suggestions are offered on this, but it is hoped that something bearing on the question may be developed in the discussion.

12 It is suggested that instead of setting the several valves on a boiler at different pressures, all the valves be set at the same pressure, with the idea that each of them will operate frequently enough to keep all in working condition, rather than run the risk of one valve being found out of condition when it is required for action.

13 It is probable that some time it will be found desirable to consider the minimum distance above the steam releasing surface of the water at which the safety valve seat may be placed.

No. 1232

SAFETY VALVE CAPACITY

By Philip G. Darling, New York

Associate Member of the Society

The function of a safety valve is to prevent the pressure in the boiler to which it is applied from rising above a definite point, to do this automatically and under the most severe conditions which can arise in service. For this, the valve or valves must have a relieving capacity at least equal to the boiler evaporation under these conditions. If it has not this capacity, the boiler pressure will continue to rise, although the valve is blowing, with a strain to the boiler and danger of explosion consequent to over-pressure. Thus, with the exception of the requisite mechanical reliability, the factor in a safety valve bearing the most vital relation to its real safety is its capacity.

2 It is the purpose of this paper to show an apparatus and method employed to determine safety valve lifts, giving the results of tests made with this apparatus upon different valves; to analyze a few of the existing rules or statutes governing valve size; and to propose a rule, giving the results of a series of direct capacity tests upon which it is based; to indicate its application to special requirements; and finally its general bearing upon valve specifications.

3 Two factors in a safety valve geometrically determine the area of discharge and hence the relieving capacity:—the diameter of the inlet opening at the seat and the valve lift. The former is the nominal valve size, the latter is the amount the valve disc lifts vertically from the seat when in action. In calculating the sizes of valves to be placed on boilers, rules which do not include a term for this valve lift, or an equivalent, such as a term for the *effective* area of discharge, assume in their derivation a lift for each size of valve. Nearly all existing rules and formulae are of this kind, which rate all valves of a given nominal size as of the same capacity.

4 To find what lifts standard make valves actually have in prac-

Presented at the New York monthly meeting (February 1909) of The American Society of Mechanical Engineers.

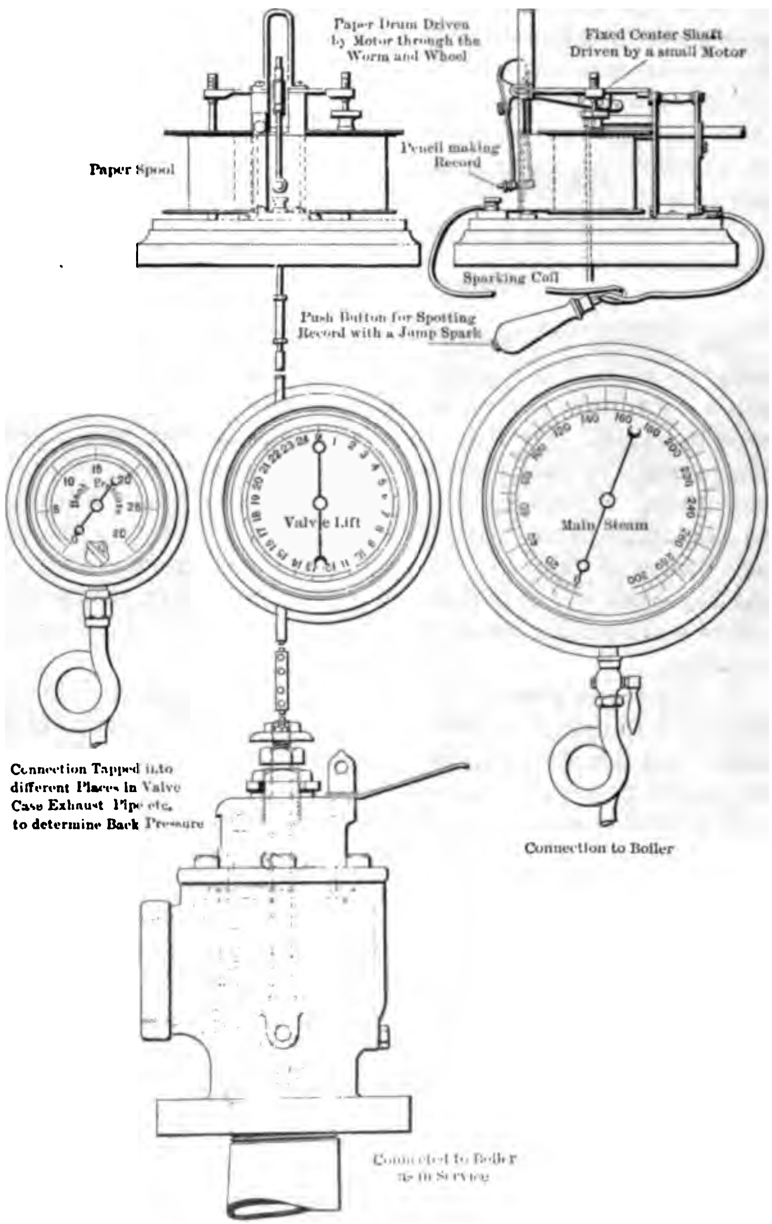

FIG. 1 SAFETY VALVE LIFT-RECORDING APPARATUS

tice, and thus test the truth or error of this assumption that they are approximately the same for the same size of valve, an apparatus has been devised and tests conducted upon different makes of valves. With this apparatus not only can the valve lift be read at any moment to thousandths of an inch, but an exact permanent record of the lift during the blowing of the valve is obtained, somewhat similar to a steam engine indicator card in appearance and of a quite similar use and value in analyzing the action of the valve.

5 As appears in Fig. 1 the valve under test is mounted upon the boiler in the regular manner, and a small rod is tapped into the top end of its spindle, which rod connects the lifting parts of the valve directly with a circular micrometer gage, the reading hand of which indicates the lift upon a large circular scale or dial. The rod through this gage case is solid, maintaining a direct connection to the pencil movement of the recording gage above. This gage is a modified Edson recording gage with a multiplication in the pencil movement of about 8 to 1, and with the chart drum driven by an electric motor of different speeds, giving a horizontal time element to the record. The steam pressures are noted and read from a large test gage graduated in pounds per square inch, and an electric spark device makes it possible to spot the chart at any moment, which is done as the different pound pressures during the blowing of the valve are reached. The actual lift equivalents of the pencil heights upon the chart are carefully calibrated so that the record may be accurately measured to thousandths of an inch.

6 In testing the motor driving, the paper drum is started and the pressure in the boiler raised. The valve, being mounted directly upon the boiler, then pops, blows down and closes under the exact conditions of service, the pencil recording on the chart the history of its action.

7 With this apparatus, investigations and tests were started upon seven different makes of 4-in. stationary safety valves, followed by similar tests upon nine makes of muffler locomotive valves, six of which were 3½ in., all of the valves being designed for and tested at 200 lb. The stationary valve tests were made upon a 94-h.p. water-tube boiler made by the Babcock & Wilcox Co. (See Fig. 2.) The locomotive valve tests were made upon locomotive No. 900 of the Illinois Central R. R., the valve being mounted directly upon the top of the main steam dome. (See Figs. 4 and 5.) This locomotive is a consolidation type, having 50 sq. ft. of grate area and 2953 sq. ft. of heating surface. Although a great amount of addi-

Fig. 2 Valve-Lift Apparatus as Used with the Stationary Test Boiler at Bridgeport, Conn.

tional experimenting has been done, only the results of the above tests will be quoted here. These lift records show (with the exception of a small preliminary simmer, which some of the valves have) an abrupt opening to full lift and an almost equally abrupt closing when a certain lower lift is reached. Both the opening and closing lifts are significant of the action of the valves. (See Fig. 3.)

8 The results of the 4-in. iron body stationary valve tests summarized are as follows: of the seven valves the average lift at opening was 0.079 in. and at closing 0.044 in., or excluding the valve with the highest lifts, the averages were 0.07 in. at opening and 0.037 in. at closing. The valve with the lowest lifts had 0.031 in. at opening and 0.017 in. at closing, while that with the highest had 0.137 in. and 0.088 in. Expressing the effective steam-discharge areas of the

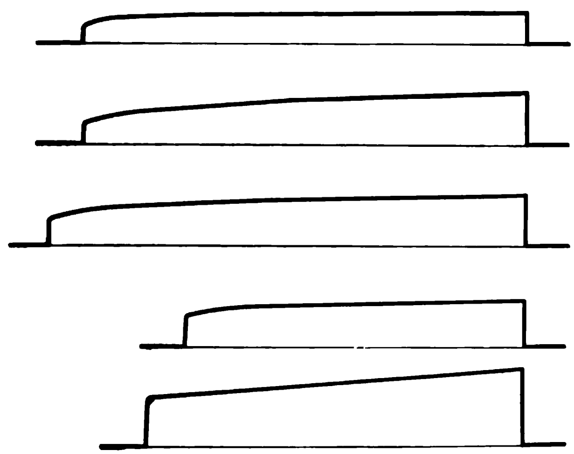

FIG. 3 TYPICAL VALVE-LIFT DIAGRAMS

valves taken at their opening lifts as percentages of the largest obtained, the smallest had 31.4 per cent, the next larger 40.8 per cent, and the next 46.6 per cent. Of the six 3½-in. muffler locomotive valves the summarized lifts are as follows: average of the six valves, 0.074 in. at opening and 0.043 in. at closing. Average, excluding the highest, 0.061 in. at opening and 0.031 in. at closing. The lowest lift valve had 0.04 in. opening and 0.023 in. closing; the highest, 0.140 in. opening and 0.102 in. closing. As percentages of the largest effective steam-discharge area, the smallest was 36.4 per cent, the next larger 39.8 per cent, and the next 46.4 per cent. In both the stationary and locomotive tests, the lowest lift valve was flat-seated, which is allowed for in the above discharge area percentages.

9 The great variation—300 per cent—in the lifts of these standard valves of the same size is startling and its real significance is apparent when it is realized that under existing official safety valve rules these valves, some of them with less than one-third the lift and capacity of others, receive the *same* rating and are listed as of equal relieving value. Three of these existing rules are given as an illustration of their nature: the United States Supervising Inspectors

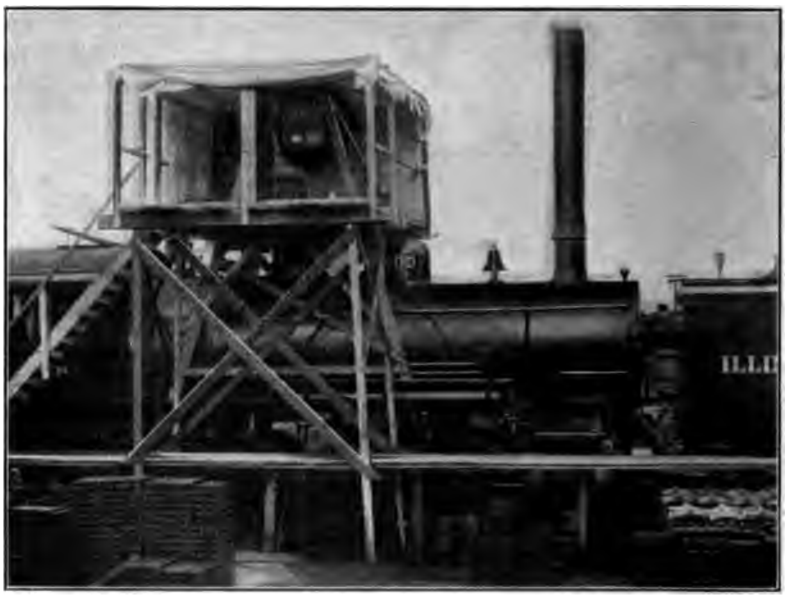

FIG. 4 VALVE LIFT APPARATUS AS ERECTED FOR LOCOMOTIVE TESTING AT BURNSIDE, ILL.

Rule, that of the Board of Boiler Rules of Massachusetts, and the Boiler Inspection Rule of Philadelphia.

RULE OF UNITED STATES BOARD OF SUPERVISING INSPECTORS

$$A = 0.2074 \times \frac{W}{P}$$

A = area of safety valve in square inches per square foot of grate surface.
W = pounds of water evaporation per square foot of grate per hour.
P = boiler pressure (absolute).

10 In 1875 a special committee was appointed by the United States Board of Supervising Inspectors to conduct experiments upon safety valves at the Washington Navy Yard. Although the pressures used in these experiments (30 and 70 lb. per square inch) were too low to make the results of much value today, one of the conclusions reported is significant:

Fig. 5 Detail of Lift Apparatus at Burnside, Ill., Showing Locomotive Valve

a That the diameter of a safety valve is not an infallible test of its efficiency.

b That the lift which can be obtained in a safety valve, other conditions being equal, *is* a test of its efficiency.

11 The present rule of the board, as given above, formulated by L. D. Lovekin, Chief Engineer of the New York Shipbuilding Co., was adopted in 1904. Its derivation assumes practically a 45-deg. seat and a valve lift of $\frac{1}{32}$ of the nominal valve diameter. The discharge area in this rule is obtained by multiplying the valve lift $\frac{D}{32}$ by the valve circumference $(\pi \times D)$ and taking but 75 per cent of the result to allow for the added restriction of a 45-deg. over a flat seat. The 75 per cent equals approximately the sine of 45 deg. or 0.707. This value for the discharge area, i.e., $\left(0.75 \times \pi \times \frac{D^2}{32}\right)$, is substituted directly into Napier's formula for the flow of steam, $w = a \frac{P}{70}$. Thus in the valves to which this rule is applied the following lifts are assumed to exist: 1-in. valve, 0.03 in.; 2-in. valve, 0.06 in.; 3-in. valve, 0.09 in.; 4-in. valve, 0.13 in.; 5-in. valve, 0.16 in.; 6-in. valve, 0.19 in.

12 Referring back to the valve lifts as given in Par. 8, it is seen that the highest lift agrees very closely with the lift assumed for 4-in. valves in the rule of the Board of Supervising Inspectors. If the lifts of valves of different design were more uniformly of this value, or if the rule expressly stipulated either that the lift of $\frac{1}{32}$ of the valve diameter actually be obtained in valves qualifying under it; or, if not, that an equivalent discharge area be obtained by the use of larger valves, the rule would apply satisfactorily However, the lowest lift valve actually has but 25% of the lift assumed for the 4-in. valves in the rule; the next larger less than 50%; while the average lift of these valves, excluding only the highest, is but 50% of the assumed lift in the rule for 4-in. valves.

MASSACHUSETTS RULE

$$A = \frac{w \times 70}{P} \times 11$$

A = area of safety valve in square inches per sq. ft. of grate surface.

w = pounds of water evaporation per square foot of grate surface per second.

P = boiler pressure (absolute) at which valve is set to blow.

13 One of the most recently issued rules is that contained in the pamphlet of the new Massachusetts Board of Boiler Rules, dated March 24, 1908. This rule is merely the United States rule given above with a 3.2 per cent larger constant and hence requiring a valve larger by that amount. The evaporation term is expressed in pounds per second instead of per hour and two constants are given instead of one, but when reduced to the form of the United States rule it gives $A = 0.214 \times \frac{W}{P}$. Figuring this back as was done above with the United States rule, and taking the 75 per cent of the flat seat area as there done, this rule assumes a valve lift of $\frac{1}{31}$ of the valve diameter instead of the $\frac{1}{32}$ of the United States rule. This changing of the assumed lift from $\frac{1}{32}$ to $\frac{1}{31}$ of the valve diameter being the only difference between the two rules, the inadequacy of the United States rule just referred to applies to this more recent rule of the Massachusetts Board.

PHILADELPHIA RULE

$$a = \frac{22.5\,G}{p+8.62}$$

a = total area of safety valve or valves in square inches.
G = grate area in square feet.
p = boiler pressure (gage).

14 The Philadelphia rule now in use came from France in 1868, where it was the official rule at that time, having been adopted and recommended to the city of Philadelphia by a specially appointed committee of the Franklin Institute, although this committee frankly acknowledged in its report that it "had not found the reasoning upon which the rule had been based." The area a of this rule is the effective valve opening, or as stated in the Philadelphia ordinance of July 13, 1868, "the least sectional area for the discharge of steam." Hence if this rule were to be applied as its derivation from the French requires, the lift of the valve must be known and considered whenever it is used. However, the example of its application given in the ordinance as well as that given in the original report of the Franklin Institute committee, which recommended it, shows the area a applied to the nominal valve opening. In the light of its derivation, this method of using it takes as the effective discharge area the valve opening itself, the error of which is very great. Such use, as specifically stated in the report of the committee above referred to,

assumes a valve lift at least $\frac{1}{4}$ of the valve diameter, i.e., the practically impossible lift of 1 in. in a 4-in. valve. Nevertheless, this is exactly the method of use indicated in the text of the ordinance.

15 The principal defect of these rules in the light of the preceding tests is that they assume that valves of the same nominal size have the same capacity and they rate them the same without distinction, in spite of the fact that in actual practice some have but $\frac{1}{3}$ of the capacity of others. There are other defects, as have been shown, such as varying the assumed lift as the valve diameter, while in reality with a given design the lifts are more nearly the same in the different sizes, not varying nearly as rapidly as the diameters. And further than this, the lifts assumed for the larger valves are nearly double the actual average obtained in practice.

16 The direct conclusion is this, that existing rules and statutes are not safe to follow. Some of these rules in use were formulated before, and have not been modified since, spring safety valves were invented, and at a time when 120 lb. was considered high pressure. None of these rules takes account of the different lifts which exist in the different makes of valves of the same nominal size, and they thus rate exactly alike valves which actually vary in lift and relieving capacity over 300 per cent. It would therefore seem the duty of all who are responsible for steam installation and operation to leave the determination of safety valve size and selection no longer to such statutes as may happen to exist in their territory, but to investigate for themselves.

17 The elements of a better rule for determining safety valve size exist in Napier's formula for the flow of steam, combined with the actual discharge area of the valve as determined by its lift. In Steam Boilers, by Peabody & Miller, this method of determining the discharge of a safety valve is used. The uncertainty of the coefficient of flow, that is, of the constant to be used in Napier's formula when applied to the irregular steam discharge passages of safety valves, has probably been largely responsible for the fact that this method of obtaining valve capacities has not been more generally used. To determine what this constant or coefficient of flow is, and how it is affected by variations in valve design and adjustment, an extended series of tests has recently been conducted by the writer at the Stirling Department of the Babcock & Wilcox Co., at Barberton, Ohio.

18 A 373-h.p. class K, No. 20 Stirling boiler, fired with a Stirling chain grate, with a total grate area of 101 sq. ft., was used. This

boiler contained a U-type superheater designed for a superheat of 50 deg. fahr. The water feed to this boiler was measured in calibrated tanks and pumped (steam for the pump being furnished from another boiler) through a pipe line which had been blanked wherever it was impossible, with stop valves and intermediate open drips, to insure against any leakage. The entire steam discharge from the boiler was through the valve being tested, all other steam connections from the boiler being either blanked or closed with stop valves beyond which were placed open drip connections to indicate any leakage. A constant

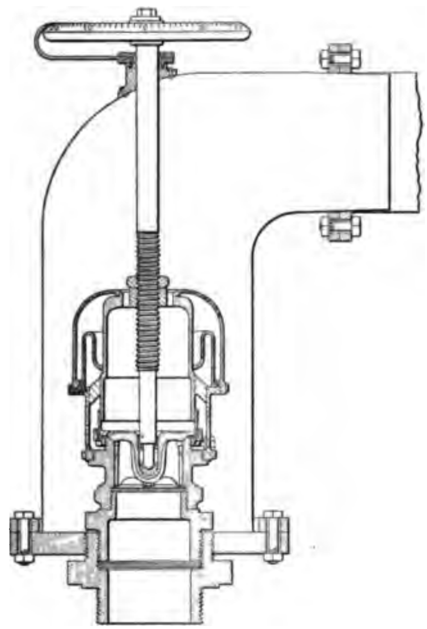

Fig. 6 Arrangement of Valve with Micrometer Spindle Used in the Direct-Capacity Testing at Barberton, Ohio

watch was kept throughout the testing upon all points of the feed and steam lines, to insure that all water measured in the calibrated tanks was passing through the tested valves without intermediate loss.

19 The valves tested consisted of 3-in., 3½-in. and 4-in. iron stationary valves, and 1½-in., 3-in. and 3½-in. locomotive valves, the latter with and without mufflers. These six valves were all previously tested and adjusted on steam. Without changing the position of the valve disc and ring, the springs of these valves were

then removed and solid spindles, threaded (with a 10-pitch thread) through the valve casing above, inserted. Upon the top end of these spindles, wheels graduated with 100 divisions were placed. Fig. 6 shows the arrangement used with the locomotive valves, the spindle and graduated wheel being similar to that used with the stationary valves. By this means the valve lift to thousandths of an inch was definitely set for each test and the necessity for constant valve lift readings, with that source of error, eliminated.

20 In conducting the tests three hours' duration was selected as the minimum time for satisfactory results. Pressure and temperature readings were taken every three minutes, water readings every half hour. A man stationed at the water glass regulated the feed to the boiler to maintain the same level in the boiler during the test; other men were stationed, one at the water tanks, one firing and one taking the pressure and temperature readings. Pressure readings were taken from two test gages connected about 4 in. below the valve inlet, the gages being calibrated both before and after the series of tests was run and corrections applied. In all 29 tests were run, of which 15 were 3 hours long, 4 were 2½ hours, 3 were 2 hours, and 7 of less duration.

21 Tests numbered 1 to 5 were preliminary runs of but one hour or less duration apiece, and records of them are thus omitted in Table 1, which gives the lifts, discharge areas, average pressure and superheat, and the steam discharge in pounds per hour of each of the other tests. The discharge areas in the valves with 45-deg. seats are given by the following formula which is easily derived geometrically:

$$a = 2.22 \times D \times l + 1.11 \times l^2$$

where

a = effective area in square inches
D = valve diameter in inches
l = valve lift in inches

In tests 8 and 23, where the width of valve seat was 0.225 in. and 0.185 in. respectively, and the valve was thus slightly above the depth of the valve seat, the area was figured for this condition.

22 As previously stated, the application of these results is in fixing a constant for the flow of Napier's formula as applied to safety valves. The formula is $w = a \dfrac{P}{70}$ in which w equals pounds discharged per second, P equals the absolute steam pressure behind

the orifice or under the valve and a equals the effective discharge opening in square inches. This may be stated as $E = C \times a \times P$; in which E equals the pounds steam discharge per hour and C equals a constant. The values of E, a and P being given for the above tests, C is directly obtainable.

23 Figuring and plotting the values of this constant indicate the following conclusions:

a Increasing or altering the steam pressure from approximately 50 to 150 lb. per square inch (tests 14 and 10) does not affect the constant, this merely checking the applicability of Napier's formula in that respect.

b Radically changing the shape of the valve disc outside of the seat at the huddling or throttling chamber, so-called, does not affect the constant or discharge. In test 15 the valve had a downward projecting lip (as in Fig. 1), deflecting the steam flow through nearly 90 deg., yet the discharge was practically the same as in tests 10 and 14, where the lip was cut entirely away (as in Fig. 6), giving a comparatively unobstructed flow to the discharging steam.

c Moving the valve adjusting ring through much more than its complete adjustment range does not affect the constant or discharge. (Tests 16 and 17.)

d The addition of the muffler to a locomotive valve does not materially alter the constant or discharge. There is but 2 per cent difference between tests 10 and 13.

e Disregarding the rather unsatisfactory 1½-in. and 3-in. locomotive valve tests, the different sizes of valves tested show a variation in the constant of about 4 per cent when plotted to given lifts.

f There is a slight uniform decrease of the constant when increasing the valve lifts.

24 The variations indicated in the last two conditions are not large enough, however, to impair materially the value of a single constant obtained by averaging the constants of all the 24 tests given. The selection of such a constant is obviously in accord with the other four conditions mentioned. This average constant is 47.5, giving as the formula $E = 47.5 \times a \times P$. Its theoretical value for the standard orifice of Napier's formula is 51.4, of which the above is 92½ per cent.

25 To make this formula more generally serviceable, it should be expressed in terms of the valve diameter and lift, and can be still

further simplified in its application by expressing the term E (steam discharged or boiler evaporation per hour) in terms of the boiler heating surface or grate area. For the almost universal 45-deg. seat the effective discharge area is, with a slight approximation, $L \times$ sine $45 \times \pi \times D$, in which L equals the valve lift vertically in inches and D the valve diameter in inches. Substituting this in the above formula gives $E = 47.5 \times L \times \text{sine } 45 \times \pi \times D \times P$, or $E = 105.5 \times L \times D \times P$.

26 The slight mathematical approximation referred to consists in multiplying the ($L \times$ sine 45) by ($\pi \times D$) instead of by the exact value ($\pi \times D + \frac{1}{2}L$). To find directly the effect of this approximation upon the above constant, the values for E, L, D and P from the tests have been substituted into the above formula and the average constant re-determined, which is 108.1. The average lift of all the tests is 0.111 in. Plotting the constants obtained from the above formula in each test, as ordinates, to valve lifts, as abscissae, obtaining thus the slight inclination referred to in Par. 23 *f*, and plotting a line with this inclination through the above obtained average constant 108.1, taken at the 0.111-in. average lift, gives a line which at a maximum lift of say 0.14 in. gives a constant of just 105. At lower lifts this is slightly larger. Hence 105 would seem to be the conservative figure to adopt, as a constant in this formula for general use, giving

$$E = 105 \times L \times D \times P$$

This transposed for D gives:

$$D = 0.0095 \times \frac{E}{L \times P}$$

Note that the nominal valve area does not enter into the use of this formula and that if a value of 12, for instance, is obtained for D it will call for two 6-in. or three 4-in. valves. For flat seats these constants become 149 and 0.0067 respectively.

27 The fact that these tests were run with some superheat (an average of 37.2 deg. fahr.) while the majority of valves are used with saturated steam, would, if any material difference exists, place the above constants on the safe side. The capacities of the stationary and locomotive valves, the lift test results of which are summarized in Par. 8, have been figured from this formula, taking the valve lifts at opening, and in pounds of steam per hour are as follows:

Of the seven 4-in. iron body stationary valves, the average capacity at 200 lb. pressure is 7370 lb. per hour, the smallest capacity valve (figured for a flat seat) has a capacity of 3960 lb., the largest 12,400 lb., and of the six 3½-in. muffler locomotive valves at 200 lb. pressure, the average capacity is 6060 lb. per hour, the smallest 4020 lb., the largest 11,050 lb.

28 To make the use of the rule more direct, where the evaporation of the boiler is only indirectly known, it may be expressed in terms of the boiler heating surface or grate area. This modification consists merely in substituting for the term E (pounds of total evaporation) a term H (square feet of total heating surface) multiplied by the pounds of water per square foot of heating surface which the boiler will evaporate. Evidently the value of these modified forms of the formula depends upon the proper selection of average boiler evaporation figures for different types of boilers and also upon the possibility of so grouping these boiler types that average figures can be thus selected. This modified form of the formula is

$$D = C \times \frac{H}{L \times P}$$

in which H equals the total boiler heating surface in square feet and C equals a constant.

29 Values of the constant for different types of boilers and of service have been selected. These constants are susceptible of course to endless discussion among manufacturers, and it is undoubtedly more satisfactory, where any question arises, to use the form containing term E itself. Nevertheless the form containing the term H is more direct in its application and it is believed that the values given below for the constant will prove serviceable. In applying the formula in this form rather than the original one, containing the evaporation term E, it should be remembered that these constants are based upon average proportions and hence should not be used for boilers in which any abnormal proportions or relations between grate area, heating surface, etc., exist.

30 For cylindrical multi-tubular, vertical and water-tube stationary boilers a constant of 0.068 is suggested. This is based upon an average evaporation of 3½ lb. of water per square foot of heating surface per hour, with an overload capacity of 100 per cent, giving 7 lb. per square foot of heating surface, the figure used in obtaining the above constant.

TABLE 1 SAFETY VALVE CAPACITY TESTS

Run at the Stirling Works of the Babcock & Wilcox Co., Barberton, Ohio November 30 to December 23, 1908

Test Number	Duration of Test	Size and Type of Valve	Adjustment	Valve Lift	Pressure	Superheat	Discharge per Hour	Discharge Area*	Remarks
	Hours			Inches	Lbs. per Sq. In.	Deg. F.	Lbs. of Steam	Sq. In.	
6	3	4-in. R. F. Iron Stationary	Regular Adj., Exh. Piped	0.0695	151.7	43.6	5120	0.6226	No Back Pressure
7	3	"	"	0.139	145.4	45.1	8600	1.255	Back Pressure 2 lb.
8	3	"	"	0.180	135.7	49.2	11020	1.704	Back Pressure 3 lb., Max. Lift
9	3	"	"	0.1045	149.4	41.9	7290	0.9400	Back Pressure 1 lb.
10	2½	3½-in. Locomotive Type R	Regular Adj. Without Muffler	0.140	146.7	39.0	8685	1.109	Open Locomotive Valve
11	3	"	"	0.070	152.5	38.C	4670	0.5493	Open Locomotive Valve
12	3	"	"	0.105	150.3	41.2	6790	0.8280	Open Locomotive Valve
13	3	" Type S	With Muffler	0.1395	146.3	38.1	8400	1.106	Muffler Valve in this and following Locomotive Tests
14	2	" "	"	0.140	52.2	51.3	3620	1.109	Test at Low Steam Pressure
15	2½	Same Except with Lipped Feather	"	0.140	146.4	39.0	8600	1.109	Different Type of Valve Disc
16	3	4-in. R. F. Iron Stationary	Reg. Adj., Exh. Piped	0.140	138.5	42.3	8770	1.265	No Back Pressure, Repetition of Test 7
17	3	"	Adj. Ring One Turn $\frac{1}{16}$-in. above Reg. Position	0.140	142.0	50.1	8900	1.265	Back Pressure 3 lb. Adj. Ring Position Ch'ng'd

18	2	1½-in. Locomotive Type S	Regular Adj., with Muffler	0.107	140.8	23.0	2515	0.4272	Unsatisfactory tests; Valve too Small for Boiler
19	1	"	"	0.060	151.2	None	1550	0.2038	
20	2½	"	"	0.075	146.3	None	2025	0.2560	
21	2½	"	"	0.075	147.7	None	1975	0.2560	
22	1½	3½-in. R. F. Iron Stationary	Regular Adj., Exh. Piped	0.070	146.8	42.6	4320	0.5493	No Back Pressure
23	3	"	"	0.140	139.9	43.6	8360	1.136	No Back Pressure
24	3	"	"	0.105	141.6	48.7	6300	0.8280	No Back Pressure
25	3	3-in. R. F. Iron Stationary	"	0.130	140.1	48.4	6370	0.8846	
26	3	"	"	0.100	142.8	45.6	5160	0.6770	
27	2	"	"	0.070	142.4	29.5	3705	0.4716	
28	3	3-in. Locomotive Type S	Regular Adj., with Muffler	0.130	138.4	48.7	7060	0.8846	
29	3	"	"	0.090	139.3	43.9	4950	0.6084	

* The valves all having 45-deg. bevel seats these areas are obtained from formula

$$a = 2.22 \times D \times l + 1.11 \times l^2$$

except as in tests 8, 23, where the valve lift is greater than the depth of the valve seat, in which case the following formula is used:

$$a = 2.22 \times D \times d + 1.11 \times d^2 + \pi \times D \times (l - d)$$

a = Discharge area. l = Valve lift.
D = Valve diameter. d = Depth of valve seat.

31 For water tube marine and Scotch marine boilers, the suggested constant is 0.095. This is based upon an overload or maximum evaporation of 10 lb. of water per square foot of heating surface per hour.

32 For locomotives the constant 0.055 was determined experimentally as explained below. Special conditions to be considered in locomotive practice separate it from regular stationary and marine work. In the first place the maximum evaporation of a locomotive is possible only with the maximum draft obtained when the cylinders are exhausting up the stack, at which time the throttle is necessarily open. The throttle, being open, is drawing some of the steam and therefore the safety valves on a locomotive can never receive the full maximum evaporation of the boiler. Just what per cent of this maximum evaporation the valve must be able to relieve under the most severe conditions can only be determined experimentally. Evidently the most severe conditions obtain when an engineman after a long, hard, up-hill haul with a full glass of water and full pressure, reaching the top of the hill, suddenly shuts off his throttle and injectors. The work on the hill has brought the engine steaming to its maximum and the sudden closing of throttle and injectors forces all the steam through the safety valves. Of course the minute the throttle is closed the steaming quickly falls off and it is at just that moment that the most severe test upon the valves comes.

33 A large number of service tests have been conducted to determine this constant. The size of valves upon a locomotive has been increased or decreased until one valve would just handle the maximum steam generation, and the locomotive heating surface being known the formula was figured back to obtain the constant. Other special conditions were considered, such as the liability in locomotive practice to a not infrequent occurrence of the most severe conditions; the exceptionally severe service which locomotive safety valves receive; and the consequent advisability of providing a substantial excess valve capacity on locomotives.

34 As to the method of applying the proposed safety valve capacity rule in practice, manufacturers could be asked to specify the capacities of their valves, stamping it upon them as the opening and closing pressures are now done. This would necessitate no extra work further than the time required in the stamping, because for valves of the same size and design, giving practically the same lift, this would have to be determined but once, which of itself is but a moment's work with a small portable lift gage which is now manu-

factured. The specifying of safety valves by a designing engineer could then be as definite a problem as is that of other pieces of apparatus. Whatever views are held as to the advantages of high or low lifts, there can be no question, it would seem, as to the advantage of knowing what this lift actually is, as would be shown in this specifying by manufacturers of the capacity of their valves. Further, as to the feasibility of adopting such a rule (which incorporates the valve lift) in statutes governing valve sizes—this would involve the granting and obtaining by manufacturers of a legal rating for their valve designs based upon their demonstrated lifts.

35 This paper has dealt with but one phase of the subject of safety valves in order that its conclusions might be drawn more clearly. The apparatus and tests shown indicate that the lifts and capacities of different make valves of the same size and for the same conditions vary as much as 300 per cent, and that there is therefore the liability of large error in specifying valves in accordance with existing rules and statutes, because these rules as shown rate all valves of one size as of the same capacity, irrespective of this variation. A simple rule is given, based upon an extended series of direct capacity tests, which avoids this error by incorporating a term for the valve lift. Finally, the method and advantage of applying this rule in practice have been briefly indicated.

No. 1233

DISCUSSION UPON SAFETY VALVES

The subject of safety valves was extensively discussed at the New York monthly meeting, February 1909, when the preceding papers by Frederic M. Whyte and Philip G. Darling were presented. The discussion was continued at the Spring Meeting at Washington, May 1909. Full reports of both discussions were published in The Journal for April and June 1909. In what follows, synopses only are given of the most important engineering data presented upon the proportions of safety valves.

LUTHER D. LOVEKIN [1] discussed the rules for marine work adopted by the Board of Supervising Inspectors. In 1903 the regulations concerning safety valves prescribed by the United States Board of Steamboat Inspectors were investigated by Mr. Lovekin, for the purpose of formulating new rules. The rule in use was based on grate surface without regard to the amount of coal burned in a given time. The rules finally prepared by Mr. Lovekin and adopted by the Board of Supervising Inspectors are based on evaporation and on Napier's formula for the flow of steam. The formula for the required area of discharge for a valve is derived as follows:

Let

P = absolute pressure
W = weight discharged per hour in pounds
A = area valve opening in square inches
d = diameter of valve in inches
a = area of valve of diameter d

From Napier's rule

$$W = \frac{360\,A\,P}{7}$$

For safety valve practice allow 75 per cent of this and restrict the lift of valve to $\frac{1}{32}$ diameter. Then

$$W = \frac{270\,P}{7} \times \frac{\pi d^2}{32} = 4.821\,Pa$$

$$a = 0.2074\,\frac{W}{P}$$

[1] Chief Engineer, New York Shipbuilding Co., Camden, N. J.

If W represents the weight of water in pounds evaporated per square foot of grate surface per hour the above formula will give the area of valve required per square foot of grate surface.

2 A table of safety valve sizes was prepared by the aid of this formula, but the board failed to state in their rules that the sizes were based on a lift of 1/32 of the diameter. In commenting upon the above formula, Mr. Lovekin brought out the following points:

3 The clear area between the valve and its seat (due to having a lift equal to 1/32 of its diameter) is only about 1/11 of the area corresponding to the nominal diameter found by the formula. Therefore it would seem that the inlet from the boiler to the safety valve need be equal in area only to the free area between the safety valve and its seat. This would reduce the opening in the boiler to about 1/11 of the area used at the present time. Experiments have shown, however, that a free entrance from the boiler to the safety valve is absolutely necessary to prevent chattering. Just what this relation is has not been determined. It would depend entirely on the length of the nozzle or pipe connecting the safety valve to the boiler. In most cases, safety valves are bolted either directly to the boiler or to a casting which is bolted directly to the boiler, so there would be very little gain in reducing the inlet nozzle to a safety valve. If safety valves are connected to the outlets of dry pipes to boilers, it is advisable to have at least 25 per cent excess area through the slots in order to prevent excessive pressures in the boilers.

4 Some rules insist on an outlet area equivalent to the full bore of the safety valve, which seems inconsistent if we have only 1/11 of the area for the steam to pass through at the valve seat. An outlet from a safety valve equal to 1/2 the nominal area of the valve would no doubt suffice in all cases. Most of the United States battleships are equipped in this manner. While the United States cruiser Tennessee was on trial the main engines were stopped suddenly. All the the safety valves responded instantly and though the steam pressure went up to 10 lb. above popping point, no trouble was experienced, proving that a combined area of outlet pipes equal to 1/2 the area of the safety valves was sufficient.

Albert C. Ashton said that while the tests show what pop safety valves would accomplish under certain favorable conditions, they have not clearly demonstrated that high-lift valves so made are a success on all applications. They certainly have shown many failures in locomotive service during the past year and must still be classed as an experiment.

2 Safety valves should never give such a large and sudden relief as to affect the water level in a boiler, neither should they close so suddenly as to cause a shock to the boiler by the quick stoppage of the flow of steam. High-lift valves which do this are not as practical as lower-lift valves which give a somewhat slower and easier relief.

3 The tests which Mr. Darling has explained show an average lift of 1/8 in. for the high-lift valves which is about double the lift of standard valves. Such high lift seems to be excessive, although there may be some virtue in making valves with a little higher lift than the common standard of 1/16 in.

A. B. Carhart, speaking on the proper rating of safety valves and their relation to boiler capacity, said that the limit of diameter of valves for stationary boilers should be 5 in., and for locomotives 3½ in.; common practice is in accord with this. Valves of 1 sq.in. discharge area are the largest advisable for locomotives. A total discharge area of 2 sq. in. for locomotives having 35 sq. ft. grate area, and of 3 sq. in. for the largest ones having 50 sq. ft. grate area, has been demonstrated to be amply sufficient. The capacity might be divided as follows: (*a*) muffled valve with close adjustment; (*b*) reserve valve regulated for reasonably greater discharge; and (*c*) an emergency valve as the ultimate protection against explosion, the other two simply to limit the working pressure under ordinary conditions. Valves generally discharge more steam than engineers are willing to permit. The strain on the boiler is dangerous when the opening is too large and sudden; and if water is lifted, it chokes the relief through the safety valve and endangers the cylinders.

2 A smaller valve with high lift is not the equivalent of a valve of larger seat diameter and less lift with the same discharge area; the smaller valve gives a smaller percentage of steam discharge, there is greater danger of sticking of the guide wings in opening and more trouble from pounding of the seat and leaking, and the outlet area becomes too large in proportion to the inlet, causing chattering and ineffective relief to the boiler, besides requiring an undesirable excessive spring compression. The lift should not exceed 0.08 in. for locomotive valves, and 0.10 in. for stationary valves used at lower pressures; prudence and economy would reduce rather than increase this limit. Many valves of high lift have been produced in past years, but all have been withdrawn because of rational objections developed in their use.

3 Every valve has a wide range of lift, which can often be varied from 0.04 in. to 0.10 in. by simple adjustment, and to still greater limits by a change of springs. In valves as commonly made, limited lift is a matter of preference, not of necessity; and such valves are giving entire satisfaction in service, with demonstrated safety under all conditions, under the present rules and ratings.

4 All internal work that must be extracted from the escaping steam, to accomplish high lift of the disc, reduces the velocity and efficiency of the relief and requires an undue throttling of the outlet, strangling the discharge instead of relieving the boiler. This condition is described in the early patent to Richardson, of January 19, 1869, showing the over-lapping regulating ring.

5 Napier's formula was used as the basis for calculating safety-valve areas as long ago as the tests made by the United States Board of Supervising Inspectors of Steam Vessels in 1875; and reports of tests of safety valves made at the Massachusetts Institute of Technology can be found in Peabody and Miller's text book on Steam Boilers printed more than a dozen years ago, showing lifts of 0.07 in. and 0.08 in., with an efficiency of 95 per cent of the calculation by Napier's formula as there applied.

E. A. May[1] spoke on the proper method of rating safety valves for low-pressure boilers. A safety valve on a low-pressure boiler is rarely called upon to exhaust all the steam-generating capacity, due to several conditions:

a In the majority of heating plants, the full amount of radiation is almost always in service, caring for a large percentage of the steam generated, and even when the radiation is nearly all cut out there is still circulation through the piping.

b Practically every steam boiler used in low-pressure work, which rarely calls for gage pressure in excess of 2 lb., has its damper regulator which, when properly set, checks combustion when 2-lb. pressure is reached.

c Chimney conditions in the majority of heating plants make it almost impossible to drive the boiler to its maximum steam-generating capacity, i.e., the maximum capacity obtainable with every condition exactly right.

[1]Mechanical Engineer, American Radiator Company, Chicago.

2 In practically all house installations at least two of these conditions exist, and in a majority all three, so that we would have to select a valve out of all proportion to actual requirements in order to exhaust all the steam which might be generated by the boiler under its full steam generating capacity under ideal conditions.

3 This brings us to a consideration of maximum capacity and how it is established: whether (*a*) by the heating surface of the boiler alone; (*b*) by the grate surface; (*c*) by the fuel-carrying capacity; (*d*) by the rate of combustion; or (*e*) by all combined. Scarcely any two manufacturers of low-pressure house-heating boilers agree in this particular. One may rate solely on the area of heating surface, another on the grate surface, and still another on the amount of fuel the grate will carry, but the writer believes that none of those factors should be considered alone.

4 In view of the wide variation in methods employed by manufacturers in ratings of boilers, as well as in the rules employed by users of safety valves, it would be difficult to select a proper size valve based on grate dimensions only. If valve manufacturers would indicate, in addition to the size of the valve, its capacity at different adjustments for exhausting steam, it would help materially. Valves could in fact be designed and sold on their exhaust capacity without regard to specific size, i. e., owing to variation in design, one valve might have a larger diameter with a lesser lift than another, while their capacity for exhaust might be identical.

5 The simplicity of this method will be appreciated by anyone considering the rules and formulæ in effect at present. If the law specified, however, that for a certain evaporative power or rating of boiler a certain exhaust capacity should be maintained in the valve each manufacturer could determine for himself the proper valve to use.

F. J. Cole quoted from a letter of an English locomotive builder stating that the "Ramsbottom" duplex safety valve is almost universally adopted there. It was introduced in 1858 and made 3 in. in diameter. Notwithstanding that boilers have since nearly doubled in capacity and pressures have been increased 50 per cent, this size is still used, which shows the disregard of proportion of safety valve to any other part of a locomotive boiler.

2 It is probable that general foreign practice for locomotive safety valves is systematized no more than in England or America. On account of the peculiar conditions governing the draft of loco-

motives the same necessity does not exist for safety valve regulation as in the case of marine or stationary boilers, the action of the exhaust automatically taking care, in large measure, of the generation of steam.

3 Mr. Cole stated that he is in favor of a thorough investigation looking towards the formulating of definite and authoritative rules for the application of safety valves to locomotives, and invited attention to the following suggestions for their preparation:

- *a* The diameter, number and kind of safety valves to be based on their capacity for discharging pounds of steam per second at different pressures.
- *b* The maximum amount of steam which the safety valves may be required to discharge when the throttle is suddenly closed after the fires have been urged to their maximum rate, to be based on the square feet of equated heating surface, so that the relative values for evaporation for various kinds of heating surface, whether of firebox, water tubes for supporting arch brick, long and short boiler tubes between the limits of 10 and 21 ft. in length, and values for different spacing of boiler tubes, will be taken into consideration. Or, what would be simpler, some approximation of average value of heating surface, equated to account for difference in length and spacing of tubes; the firebox heating surface in this case to be considered as a certain percentage of the whole for all sizes of locomotives.

4 A great diversity of practice exists in the spacing of flues in locomotive boilers. The variation in length ranges in common practice from 10 ft. to 21 ft. These two conditions make the use of unequated heating surface somewhat unreliable as an absolute guide for the amount of water evaporated. It is evident in two boilers having the same diameter and the same length between flue sheets that one will contain a much larger amount of heating surface if the flues are spaced 11/16 in. apart than the other if they were spaced 1 in., and both these figures are within the limits of what is accepted as good practice. Furthermore the heating surface of flues of the same diameter and, say, 11 ft. long, will be much more effective per square foot than if the flues were 21 ft. long. Firebox heating surface is, of course, very much more efficient than tube heating surface, and the water-tube heating surface for supporting firebricks is more efficient than the ordinary boiler tubes.

5 Tests show that the evaporation of boilers is somewhat independent of the tube spacing, and probably is more in direct relation to the cubical contents, as it is a matter of common knowledge that the steaming capacity of boilers does not vary in direct proportion to the amount of heating surface if a great variation exists in the spacing of the tubes.

6 The evaporation per square foot of heating surface in locomotives is a variable quantity, ranging from 6 lb. or even less to 15 or 16 lb. per square foot of heating surface per hour. On the authority of Professor Goss, from Purdue University tests, it may be stated that the evaporation in a very general way, and the draft produced by the blower and exhaust in terms of inches of water, will be approximately as follows:

1-in.	draft	will	evaporate	3.0	lb.	per	foot	of	heating	surface	per hour
2-in.	"	"	"	6.0	"	"	"	"	"	"	" "
3-in.	"	"	"	8.2	"	"	"	"	"	"	" "
4-in.	"	"	"	10.0	"	"	"	"	"	"	" "
5-in.	"	"	"	11.4	"	"	"	"	"	"	" "
6-in.	"	"	"	12.6	"	"	"	"	"	"	" "
7-in.	"	"	"	14.0	"	"	"	"	"	"	" "
8-in.	"	"	"	15.0	"	"	"	"	"	"	" "

Dr. Chas. E. Lucke stated that another element in the safety valve question, of minor importance, perhaps, is the time element. He had experimented for many years with rapidly rising and rapidly falling pressures, and believed that increase in pressure in a chamber may go momentarily far beyond what a safety valve is set for. Because this excess is only momentary and measured in fractions of seconds, it should not be considered of no consequence; it is indeed of far more consequence, as a suddenly applied load cannot be resisted by the metal under stress as well as a steady load.

2 If then by any romote series of circumstances the pressure in the boiler suddenly rises, as it may, the time element will enter in, the pressure will go higher than the safety valve is set for, before the valve opens, and will suddenly stress the entire structure. This subject should be stu died e perimentally, with the others involving the steady rate of steam discharge, to discover if it is of any consequence in practical safety valve work. Although Dr. Lucke had never seen the pressure rise in a steam boiler in this way, he believed it could so rise, producing the effects described.

JESSE M. SMITH thought Dr. Lucke had touched on a point which needed investigation. Another point along the same line is the danger of having a safety valve too large, particularly if it be of the "pop" kind. If a boiler be stored with water at a temperature corresponding to 150-lb. pressure, and that pressure be suddenly reduced, a portion of the water will instantly flash into steam and the boiler may be greatly strained and may explode. There is danger from having a safety valve too large as well as from having it too small.

2 Those who have had to do with the investigation of boiler explosions, and particularly those being questioned with regard to them in the courts, will realize the necessity for rules based upon scientific investigation and reason, instead of rules having no special reason for their existence except that they have existed for hundreds of years or more.

GARLAND P. ROBINSON stated that the commission with which he is connected had collected reliable data on about 7500 locomotive boilers, and he had recently calculated the valve capacity of 1000 of these boilers for the purpose of finding the average practice for safety-valve equipment. The greatest variations were noted. For instance, boilers using 180-lb. pressure with valves of $\frac{1}{16}$-in. lift have two 3-in. valves to take care of the evaporation from 1750 to 3350 sq. ft. heating surface. It was also found that two $2\frac{1}{2}$-in. valves were used to take care of 900 to 1900 sq. ft. of heating surface. These cases represent whole classes and not individual boilers. Therefore it would appear that no rule has been followed to determine the size of valve required.

2 The heating surface, all things considered, is the best unit of measurement for determining the size of safety valves for locomotive boilers. In his opinion a formula based on the heating surface and providing for 50 per cent of the maximum evaporation of the boiler, will give satisfactory results for locomotives. A formula for size of safety valves for locomotive boilers can be derived in the manner shown in Mr. Darling's paper on Safety Valve Capacity.

3 For locomotive valves with 45-deg. valve-seats, Mr. Robinson used the formula

$$D = 0.05 \frac{\text{heating surface}}{L \times P}$$

and for locomotive valves with flat valve-seats, the formula

$$D = 0.033 \frac{\text{heating surface}}{L \times P}$$

4 He had checked 1000 boilers and found the average constant to be 0.0441 for present practice. Included in the 1000 boilers, however, are a number which evidently have valves of insufficient size, as the constant in their case is only 0.024. Eliminating this class of boilers, the constant for average practice is about 0.05, as given in the formula.

H. C. McCarty[1] said that his experience had clearly proved that safety valves with *unusual discharge*, resulting from *increased lift* of valve, cause a violent disturbance in the water level, especially on the large modern locomotive boilers, and in proportion to this disturbance is the volume of water passing the throttle valve, and hence to the steam chest and cylinders, increased. Railroads will be relieved of many expensive repairs by reversing these conditions, and thus produce the driest steam possible for the engine. To this end, the throttle valve, as is well known, is located at the highest possible point in the boiler. Further to secure greater locomotive efficiency in this direction, the safety valve should be at as high a point on the boiler as clearance limits permit, and with an independent short connection of ample dimensions to the boiler.

2 Mr. McCarty said his further observations have been that the location of the valve on a boiler has much to do with the normal crest of the water. Air-brake shocks in train and similar effects, in conjunction with high-lift valves, have been a frequent cause of locomotive failures through the combination of undesirable conditions, all of which cause a greater volume of water to pass through the throttle valve and safety valve.

3 In the speaker's experience in locomotive service he had never had even a suggestion of the necessity or the advisability of increasing the lift of the valve; on the contrary, the reverse condition, from a service standpoint, presents itself. The possible limited economy in first cost of a slightly smaller valve having increased lift, to accomplish increased discharge, compared with the next larger size valve with normal lift, is deceptive, as the short life and expensive maintenance of the high-lift valve make it not only an expensive burden to the railroads, but an unreliable device.

M. W. Sewall said that if the evaporative capacity of the boiler and the delivery capacity of the safety valve were adapted to each other, no difficulty need arise from the use of high-lift valves. The areas

[1] President Coale Muffler and Safety Valve Co., Baltimore, Md.

of approach to the valve and discharge from it should then be such as practice has already shown to be essential. The usual diameters have been mentioned in the discussion as if they could not be changed, and the high and low lifts have been spoken of as related to those diameters. As the high-lift valve has a greatly increased discharge capacity, however, it should be reduced in diameter and an entirely new adaptation of diameters to "pounds of steam discharged" should be made. A manufacturer could then adopt any desired combination of diameter and lift and the valves would be rated on the pounds of steam delivered per second.

A. A. Cary, in discussing safety-valve springs, said that the ratio between the pitch diameter of the spring and the diameter of the wire composing it should not be less than 5 to 1, but 7 to 1 is a better minimum proportion. A pop-valve spring should not be wound to a smaller proportion than 7 to 1, and with such a spring coiled to a smaller ratio he had found a considerable breakage resulting.

2 One matter deserving careful attention in the design of pop-valve springs is the shape of the section of wire used. Unquestionably, the best and safest wire for springs is that of round section. The principal stress occurring in the wire of a helical spring is that of torsion, and in a wire of square section the greatest fiber stress occurs at the corners of the square, which are the most distantly removed from the center of the section.

3 The only advantage gained by the use of square-wire springs is a slight reduction of the space required for a spring having the same resistance to compression.

4 The most durable of all is the helical spring designed to resist extension, known as an extension spring. When this spring is properly applied, the load is carried directly along the line of the spring's axis, thus doing away with the "buckling" which so frequently imposes harmful bending strains (in addition to the torsional strain) in the wire composing compression springs. The use of compression springs for pop valves has become almost universal, but there is no reason why extension springs of good design cannot be used for this purpose.

F. L. DuBosque thought that the formula in the U. S. marine laws has the serious defect that some of its factors are left to the opinion of any one of a great number of persons concerned in its use. The factor W is made up of two quantities, the calorific value of the fuel and the amount burned per square foot of grate surface, and the value of these factors can with reasonable judgment be varied so as to vary the size of the safety valve at least 50 per cent. It is now impossible for a designer to specify the size of a safety valve on a marine boiler without first obtaining from the United States inspector his opinion on the value of these two factors, notwithstanding the fact that the inspector who is compelled to decide this question cannot possibly have as much information to assist him as the designer.

2 This new formula, therefore, has not in any way improved the Rules of the Steamboat Inspection Service and, as pointed out above, has added only a complication. As to results produced by it, it is easy to see that by selecting proper proportions for the two factors that make up W,—and these factors both may be within reasonable limits,—the same result will be obtained as by the old formula. The old formula at least gave the designer a certain basis to work on, and if he was designing his work with the proper regard for safety he had the privilege of deviating from the formula if he felt it did not provide a valve of large enough size.

3 This new formula is also similar for cylindrical and water-tube boilers. Practical operation shows that a safety valve on water-tube boilers should be much smaller than on cylindrical boilers of equal evaporative power. A sudden release of steam pressure in a water-tube boiler with its limited water-line area causes more damage by lifting the water within the boiler than can be caused by a moderate increase in steam pressure.

L. D. LOVEKIN, in replying to the remarks by Mr. DuBosque, said that he was not aware of the trouble he had caused marine engineers, and still further, he saw no reason for such trouble. He had discussed the matter fully with a number of engineers and showed them the new formula which he proposed submitting to the Board, and all agreed that his formula was based upon common sense.

2 Any safety valve based on one square inch of opening for three square feet of grate area for a Scotch boiler, and one square inch of opening for six square feet of grate area for a water-tube boiler, is absurd, and yet this was the formula used by the United States Inspectors for many years.

3 Nelson Foley, of England, states that safety valves may be made capable of lifting, say $\frac{1}{8}$ of their diameter; that a high lift is useless and may be an evil if anything gives way; that the waste steam pipe, when not under the Board of Trade rules, may be equal in area

to the opening with the lift just mentioned, i. e., the area of the waste steam pipe would be one-half the gross cross-sectional area of the valve.

4 Our United States Navy Steam Engineering Department, with all their experience in connection with boilers, have agreed with several prominent authorities abroad on a lift of $\frac{1}{4}$ the diameter of the valve. It does not follow, however, because a valve has provision for a lift equal to $\frac{1}{4}$ of its diameter, that it ever lifts this amount. It is simply a provision in case the valve is required to be lifted by the safety-valve hand-operating gear usually provided on all ships.

5 The area of waste steam pipe on all our recent naval vessels is made $\frac{1}{2}$ the gross cross-sectional area of the valve, which accords with the statements of Mr. Foley.

6 It is a coincidence that while the present rule might give an excessive lift on sizes above $4\frac{1}{2}$-in. diameter, it averages up closely to the sizes recommended by many manufacturers for valves below $4\frac{1}{2}$-in. diameter. The rate of evaporation of 180 lb. in the present rule almost coincides throughout with the Board of Trade formula for safety valves under natural draft.

7 It appears therefore that the Board of Trade thought it wise to keep all boilers under natural draft at the same rate of evaporation, i. e., all boilers worked under natural draft are assumed to be capable of evaporating 180 lb. of water per square foot of grate surface, which seems to be a safe maximum rate for any marine boiler under natural draft.

8 When forced draft is used, under the Board of Trade regulations, the area of the safety valve must not be less than that found by the following formula.

$$A \times \left(\begin{array}{l}\text{estimated consumption of coal} \\ \text{per square foot of grate, in} \\ \text{pounds per hour} \div 20\end{array}\right) = \text{area of valves required.}$$

A equals the nominal area of the valve, based on its diameter, as found from the table of safety valve areas under the Board regulations.

9 The results of experiments on safety-valve lips illustrated in the figures may be of interest to the members of the Society. These experiments were made by Nelson Foley to determine the effect of adjusting the lip on safety valves.

Nathan B. Payne believed the most important point brought out by the discussion was that there is no proper standard of measurement

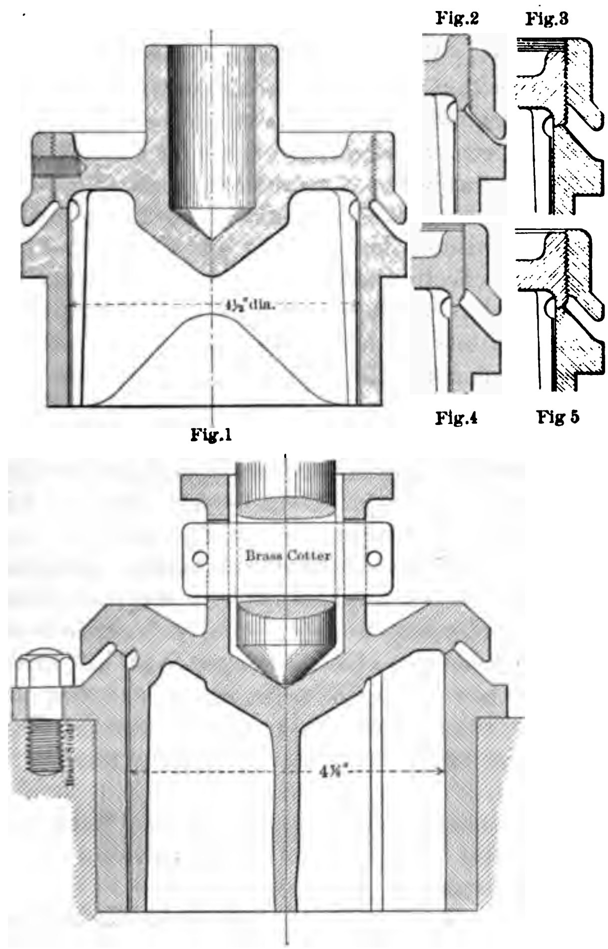

Fig. 1 Results of Experiments on Safety-Valve Lips

VALVES ROSE AT 81 LB. AND LIFTED ABOUT $\frac{3}{32}$ IN. RATIO OF VALVE AREA TO GRATE AREA ½ SQ. IN. TO 1 SQ. FT. FIG. 1 VALVE CLOSED IN ½ MIN. AT 79 LB., AND VIBRATED CONSIDERABLY. FIG. 2 BLEW STEADILY, WITHOUT CLOSING. FIG. 3 CLOSED IN 1 MIN. AT 80 LB. PRESSURE DROPPED STEADILY. FIG. 4 SAME AS IN FIG. 3. CLOSED IN ¾ MIN. AT 79½ LB. LESS VIBRATION THAN IN FIG. 1.

for the safety valve's capacity. Whether a high-lift or a low-lift valve is selected, what we must have is some way of measuring the relieving capacity. When we buy a 4-in. valve, for instance, we want to know whether that valve has relieving capacity for a 100-h.p. or a 200-h.p. boiler, or what size it is suited for.

2 We have been thinking with regard to the relieving capacity of safety valves that we need consider only one dimension, but it is absolutely impossible to determine the amount of relieving capacity in a given time without knowing the lift of the valve off the seat, so as to get the effective area of opening. The question for the user to decide is how much relief he can get from a given make and size of valve. If one maker offers a safety valve having $\frac{1}{32}$-in. lift, and another offers a $\frac{1}{8}$-in. lift, each should state how many pounds of steam per hour his valve will relieve.

H. O. Pond said the question of high-lift and low-lift valves seemed to be one simply of capacity. If the low-lift valve will deliver a certain number of pounds of steam at a given pressure and temperature, and its capacity under these conditions is known, this is the principal thing required. The same test applies in the case of the high-lift valve, the essential point, however, being to know how much steam the valve will discharge. Undoubtedly a high-lift valve will give more capacity than a low-lift valve having the same diameter of opening. This being so, we could use a smaller valve of the high-lift type, which would be an advantage in many ways.

F. L. Pryor summarized the results obtained from tests made to obtain the blowing-off pressure of safety valves when tested with water and with steam.

2 A standard 4-in. pop safety valve, set for 125 lb., was mounted on a 4-in. pipe and so connected that either steam pressure or water pressure could be admitted to the valve. One set of tests was made over a period of 15 days, the test of one day being with steam and the following day with water. In a second series of tests, the valve was tested at three different settings on the same day, viz. 104, 131 and 159 lb. The third series of tests was made with the valve at a number of different settings, from 105 to 165 lb., one measurement being made directly after the other and no precaution taken to insure that the valve had returned to its normal temperature after the preceding test, except that before operating with water pressure a considerable amount of water was flushed through the valve.

3 The results obtained in all the tests were in practical agreement, and indicated that the blowing-off pressure with steam and with water did not differ to any great extent, although the pressure to blow off with water was higher than with steam.

4 In the case when the valve was allowed to cool 24 hours, the water pressure required to open it was about 3½ lb. higher than the steam pressure. In the tests where the valve was cooled thoroughly with water, the pressure with water was about 3 lb. higher than with steam. In the rapid change test the water pressure amounted to about 2.6 lb. more than the steam pressure.

5 In all tests the steam and water pressure record was that at which the valve was in full operation. In the case of the steam pressure test there were two distinct points below full open pressure which could also have been noted: when the valve began to leak, which occurred about 2 lb. below the final blowing-off pressure, and when the rate of flow suddenly increased, which was about 1 lb. below maximum.

A. B. CARHART, speaking on the proportions of safety-valve parts, said that the specifications which require valve seats to be made of non-corrosive metal, and the rules which compel every valve to be tried and lifted by the lever every day, aim to overcome the ever-present danger that the valve may stick upon its seat and fail to open at the critical moment. But the greatest cause of the sticking of the valve, when it does occur, is not corrosion of the seat face, but the binding friction of the disc-guides against the side of the well or throat of the valve. This cocking or binding effect can be decreased by any modification of design which will reduce the diameter of the cylindrical guide, or which will bring the guiding surface close to the plane of the seat, both of which will reduce the moment of the friction or cocking stress.

2 Any device which reduces the lift of the disc and the spring movement to the least possible amount will also reduce the eccentric spring action and its effect, and any valve design which contemplates an unnecessarily large lift or compression disadvantageously magnifies this effect.

3 An early and still common form of safety valve has the seat opening beveled at an angle of 45 deg. The effective steam passage is therefore measured by the sine of 45 deg., which is approximately only 0.7 of the actual compression of the spring when the valve opens, so that the spring must necessarily compress about 1½ times

the effective lift. Even this does not always afford a free passage for the steam to the air, where there is vertical overlap of the regulating ring against the lip of the disc in order to increase the lift against the greater pressure of the shortening spring.

Edw. F. Miller presented the following method of obtaining the valve discharge area based on the rate of fuel consumption. The weight of steam flowing through an orifice with a slightly rounded entrance may be figured quite accurately by Napier's formula (sometimes called Rankine's formula). Its accuracy for commercially dry steam has been shown by tests made under pressures varying from 30 to 150 lb. There are a number of papers on this subject in the earlier volumes of the Transactions. According to the formula the weight of steam discharged per second through an orifice with slightly rounded entrance is $\frac{FP}{70}$ where F is the area of the orifice in square inches and P is the pressure in pounds absolute on one square inch.

2 The discharge per second through an orifice with a sharp edge at the entrance, as would be the case in a safety valve, has been found from actual tests on valves to be 0.95, the amount figured from this formula. The opening needed in a safety valve may be figured as follows:

G = grate area.

R = rate of coal consumption per square foot of grate per hour.

9 = probable evaporation per pound of coal under actual conditions.

$$\frac{G \times R \times 9}{3600} = \text{weight of steam made per second.}$$

Equate this to the preceding expression and solve for F:

$$\frac{G \times R \times 9}{3600} = 0.95 \frac{F \times P}{70}$$

$$F = \frac{G \times R \times 9 \times 70}{3600 \times P \times 0.95}$$

3 The area of the opening through a safety valve is equal to the inner circumference of the seat times the effective lift. For a valve with the seat at an angle the effective lift is equal to the lift multiplied by the cosine of the angle the seat makes with a horizontal.

4 For a 45-deg. angle the effective lift is 0.707 × lift. Calling D the inner diameter of the valve, the opening is

$$\pi \times D \times \text{lift} \times 0.707$$

Substituting this for F,

$$\pi D \times \text{lift} \times 0.707 = \frac{G \times R \times 9 \times 70}{3600 \times P \times 0.95}$$

If the lift of the valve is $\frac{1}{10}$ in.,

$$D = \frac{G \times R \times 9 \times 70}{3600 \times P \times 0.95 \times \pi \times 0.707 \times 0.1} = \frac{GR}{P \times 1.206}$$

5 If the lift is 0.05 instead of 0.10, then the valve diameter D is doubled. Doubling the pressure will make the same valve with the same lift take care of double the weight of steam. For illustration:

Grate area = 2^r.
Coal consumption = 18 lb. per square foot per hour.
Pressure = 120 lb. absolute.

$$D = \frac{25 \times 18}{120 \times 1.206} = 3.1$$

Pressure, 150 lb. absolute.
Grate area, 50 sq. ft.
Coal consumption, 25 lb. per square foot per hour.

$$\frac{50 \times 25}{150 \times 1.206} = D = 6.9 \text{ in.}$$

A valve as large as this would be replaced by two of equivalent capacity.
The circumference = 3.14 × 6.9

Two smaller valves of diameter $\frac{6.9}{2} = 3.45$ will give the same circumference and the same discharge with the same lift.

George H. Musgrave[1]. The function of the safety valve is twofold: (*a*) it gives notice of the highest pressure permissible; (*b*) it gives the alarm that more water or less fuel is needed. He had been told by engineers in the marine service, that through the use of safety valves with excessive lift and quick discharge, their engines had been plugged by taking over water. He had known of numerous occasions in locomotive service where there have been very disastrous results.

[1]General Sales Agent, Star Brass Mfg. Co., Boston, Mass.

2 If the same principle is to be introduced in high-lift locomotive safety valves that is now used in injectors to raise water, what is to prevent the syphoning of the water to the throttle valve, and its flowing through the dry pipe and into the cylinders? From his long experience, originally in locomotive service, afterwards in marine and stationary service and at the present time, on safety valves for all uses, he would suggest that the medium-discharge type is the safest and most satisfactory valve to use. Any valve that will materially disturb the water level and have a tendency to raise it is dangerous.

M. W. Sewall, speaking on exact regulations for valve proportions, said that the public authorities and insurance companies should establish means of regulation in regard to the following:

a Flange diameters for various rates of discharge.
b Requirements as to minimum discharge of pounds of steam per second within given ranges of pressure.
c Requirements as to non-corrosive seats or other operating parts, strength of parts, means of operation by hand, and security against being put out of adjustment by ill-disposed persons.

Albert C. Ashton was opposed to high-lift valves, since they open and close so suddenly as to injure the boiler and its connected fittings, as well as the valve itself.

2 He thought that any revised rule for the size of pop safety valves should not prescribe a capacity of relief that could be obtained only with a high-lift valve, as suggested by Mr. Darling and Mr. Lovekin, who approve of a lift equal to $\frac{1}{32}$ of the diameter of the valve.

A. F. Nagle presented the following table on the size of safety valves for boilers of a given power, based upon the following data:

a A boiler horsepower is the term used to express the evaporation of 34.50 lb. of water per hr. from and at 212 deg. Fahr.
b A spring safety valve can and should be depended upon to lift $\frac{1}{32}$ of its diameter.
c The flow of steam follows closely Napier's formula, reduced to $92\frac{1}{2}$ per cent by Mr. Darling's experiments (Par. 24).

d The formula used in the computation is h.p. = $0.0951\ D^2P$, transformed from Mr. Darling's formula, where

h.p. = boiler horsepower.
D = diameter of valve in inches.
P = absolute steam pressure.

2 In using this table, allowance must be made for what is likely to be the maximum horsepower of the boiler and not its normal rating. Fifty per cent overload is not unusual, and double the rating, while not impossible, is not liable to pass through the safety valve.

HORSEPOWER OF BOILERS AND SIZE OF SAFETY VALVES

Steam Pressure Pounds	Safety Valves					
	2 in.	2½ in.	3 in.	3½ in.	4 in.	4½ in.
100	44	68	98	134	175	221
125	53	83	120	163	213	269
150	63	98	141	192	251	318
175	72	113	162	221	289	366
200	82	128	184	250	327	414
225	91	142	205	280	365	462
250	100	157	227	309	403	510

Note.—Roughly every 4 lb. of coal burned per hour represents one boiler h.p.

Jerome J. Aull thought that the proposed rule should include a term for a fixed lift rather than a variable one, for the reason that with the latter would result a hopeless confusion of safety-valve openings in boilers of the same size. Thus under Mr. Darling's rule a boiler of a certain size might be provided with a safety-valve connection varying from 2½ in. to 4 in. in diameter, depending upon the make of valve specified. It would be far more convenient and satisfactory to standardize safety-valve connections so that any valve having the required capacity could be used. To do this it would be necessary that the valves themselves be standardized within certain set limits and this could be done only by a body of disinterested engineers, properly authorized to investigate the subject.

2 Mr. Aull condemned high lift as it made the seats and spring-bearings subject to a severe pounding action; there is more danger of chattering; close adjustment is not possible; there is danger of lifting water; and the boiler seams are sometimes strained to the opening point.

Philip G. Darling, in closing, dealt with the many different values at present being advocated for safety-valve lifts. Recent articles place this maximum limit variously at 0.05 in., 0.06 in., 0.08 in., 0.09 in., and 0.14 in., for the same size valves.

2 It is well known by those in touch with foreign manufacturers that valve lifts, spring compressions and other valve elements which are radically different from what has been the general practice in this country, are being used successfully, and in some places universally.

3 Two cases will illustrate this. The springs on 3¾-in. triplex valves of the Thornycroft design, used widely in English marine practice, are not only of the exposed type, but have, when set for a designed pressure of 250 lb., a compression of 4 in. These are regular safety valves of the same principles as our own duplex valves. To those who would condemn a compression of ½ in. to ⅝ in. as radically high and unsafe this instance should be suggestive and help to broaden their conceptions of the possibilities of safety-valve design. Again, in *London Engineering*, February 26, 1909, reprinted in *Power*, March 30, 1909, J. H. Gibson tells of exceptionally good results obtained in a valve having 0.21-in. lift. He says: "We think we are justified in the assumption . . . . that anything tending to reduce the size of these important fittings (safety valves), which have been growing to abnormal proportions of late, is a step in the right direction."

4 High lift is not synonymous with excess safety-valve capacity. A boiler's evaporation absolutely determines the necessary safety-valve capacity. In a given boiler the pounds of steam per hour which the valve should be able to relieve can be definitely figured and all that is further needed, in making the correct valve specification, is the capacity of the safety valves.

5 It is not a question of lift for itself, but of requisite relieving capacity, and if this is obtained with a 3-in. instead of a 4-in. or 4½-in. valve there is a positive, real advantage, not only in original cost but in the maintenance and better action of comparatively small rather than large valves.

6 It is thus not a uniformity in the lifts of different valves which the engineering public should demand, but rather the practice of stating relieving capacities, based on the actual lifts existing in the valves themselves. If the capacities were stamped upon the valves, as already done by one maker, it would give a rational basis for use in the application of safety valves to boilers.

7 It has been objected that capacities thus published could not be verified without actual capacity runs, such as the Barberton tests recorded in the paper, on the ground that in some valves the effective area of discharge at the seat, upon which the formula is based, is not the smallest discharge area; or even if it is, that there is a material throttling or holding back of the steam flow. Valves containing the original Richardson adjusting ring have been cited as designs in which this choking occurs.

8 In order to secure information upon this matter prior to conducting the direct capacity tests at Barberton referred to in the paper, the effective discharge areas at the seat and at the most contracted passage between the lip and adjusting ring were figured and plotted for the different valves tested at different lifts. Further, a 3½-in. valve was constructed having this Richardson ring and projecting disc lip design, and for the same valve another disc and ring in which the projecting lip was cut entirely away. In the former the discharging steam was deflected through practically 90 deg., and in the latter the steam had a free straightaway passage. These two designs were radically different and fairly represented the extremes of what on the one hand seemed to be a choked or impeded steam discharge passage and on the other a free open one.

9 The most effective discharge areas of the two taken at the seat and at the most contracted part of the passage between the lip and rings are given in the table in square inches for different lifts.

TABLE 1 EFFECTIVE DISCHARGE AREAS

Valve with Projecting Lip and Richardson Ring			Valve without the Lip	
Lift	At Seat	Most Contracted Point Beyond Seat	At Seat	Most Contracted Point Beyond Seat
0.02	0.16	1.20	0.16	2.01
0.06	0.47	1.40	0.47	2.14
0.10	0.79	1.76	0.79	2.29
0.14	1.11	2.27	1.11	2.43

These areas, taken with Napier's formula, give a method of figuring the theoretical pressure existing in the "throttling chamber" under the disc lip; that pressure being to the boiler pressure as the effective discharge area at the seat is to the most contracted area between the lip and ring beyond. The highest pressure thus indicated in the throttling chamber is less than 50 per cent of the corresponding boiler pressure. This pressure in the "throttling chamber" being the discharge pressure of steam passing over the valve seat, and the full flow of Napier's formula being practically unaffected by any discharge pressure less than 60 per cent of the original or boiler pressure, the theoretical conclusion is that the discharge from neither of these valves would be affected by the disc design or discharge areas outside of the valve seat.

10 In replying to the references made to disastrous results to boilers such as the opening up of seams and fittings due to the sudden release or cutting off of steam by the safety valve, Mr. Darling discussed the sudden change of pressure due to opening and closing of throttle valves and blow-off valves, concluding that the shock to the boiler from this source would far exceed that due to the closing of the safety valve.

11 The larger the safety valve compared with the boiler the greater the shock to the boiler due to its action, if such shock exists. A 5-in. valve mounted directly upon a 94-h.p. test boiler would increase or accentuate this tendency to strain over say a 3½-in. valve on an 800-b.h.p. locomotive surely 12 or 13 times. Yet with a most sensitive boiler pressure test gage graduated to pounds and mounted upon this 94-h.p. test boiler, absolutely no recoil of the gage hand upward either at the opening or closing of a 5-in. valve is perceptible. It would seem that some increase of pressure such as would be indicated upon the gage would be positively necessary to transmit a strain to the boiler.

12 Two cases had recently come to his notice in which locomotive safety valves had loosened from their spud connections and had blown off while the boiler was under its full steam pressure. One was a 3½-in and the other a 4-in. valve, which therefore opened areas of 9.6 and 12.6 sq. in., respectively, while the maximum corresponding safety-valve discharge area could be but a little over one square inch. Yet no damage to the boilers was experienced. The blowing off of 2-in. locomotive whistle connections had been cited as a not infrequent occurrence. The steam-relief in such accidents is of course more sudden than with a safety valve, and the fact that this opening of ten to twelve times the maximum discharge area of the corresponding safety valves results in no further inconvenience than the replacing of the fittings raises some question as to the actual disaster impending in the use of valves having a discharge area of but 1 sq. in.

No. 1234

A UNIQUE BELT CONVEYOR

By E. C. Soper, Detroit, Mich.

Member of the Society

It is quite possible that a description of a belt conveyor a quarter of a mile long, and requiring more power to operate empty than loaded, will be interesting to some of the members and since its installation and operation are at variance from the prescribed rules of conveyor design, we beg to submit the following:

2 The belt conveyor was built during the summer of 1908 in one of the large portland cement plants of the South. It consists of a 24-in. 8-ply canvas belt in two sections, one section about 1000 ft. between centers, and the other with 1100 ft. between centers, its function being to convey the shale used in the manufacture of the cement, from the shale quarry to the plant. The shale deposit is located on a mountain about 247 ft. above the shale storage tanks, as shown in profile, Fig. 1. The two sections intersect at an angle of 140 deg. 40 min., so that the blasting from the limestone quarry does not interfere with the operation of the belt. The belt conveys the shale around the limestone quarry, as shown in plan, Fig. 1.

3 The belt is flat and carried by rollers, the top row having 4 ft. between centers and the return idlers 12 ft. between centers. Guide rollers are placed with about 40 ft. between centers along both upper and lower belts. (See Fig. 2.) The majority of manufacturers of belt conveyors recommend the maximum length between centers of a single belt to be about 700 ft. to 800 ft.

4 Referring to Fig. 1, the belt conveys the material down-hill, and to this fact is due the apparently parodoxical results in power required to operate, shown in Tables 1 and 2.

5 Because of the extreme length of the belt, and the fact that there is no roof or other covering, it was necessary to install some system for taking care of the expansion and contraction, in addition

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to the ordinary stretch of the belt, which is taken up in the majority of installations by 24-in., 36-in. or 48-in. takeups, according to length of belt. A set of 36-in. takeups, (Fig. 3) was installed at the upper end of each of these belts to maintain alignment and equal

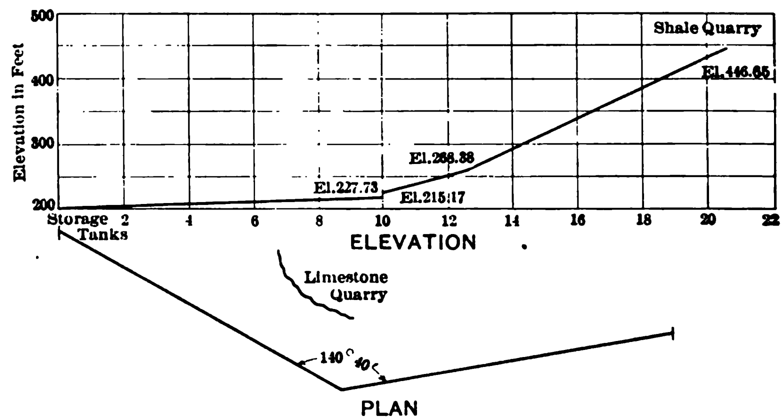

FIG. 1 PROFILE SHOWING ELEVATION AND PLAN OF CONVEYORS

tension on each edge of the belt. The system installed acts as a tension carriage and makes it less often necessary to cut out the

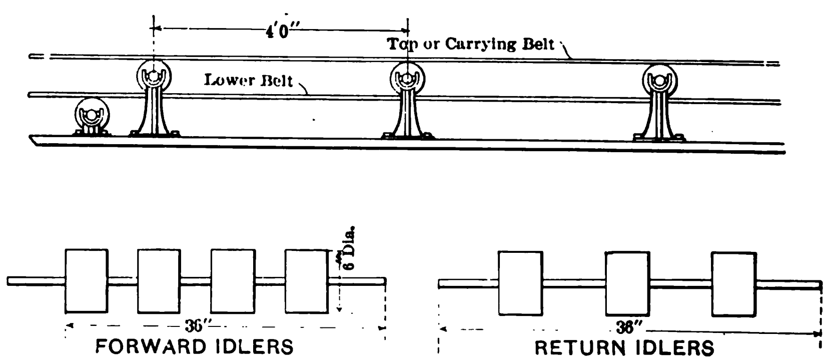

FIG. 2 DETAILS OF FORWARD AND RETURN IDLERS

slack in the belt, and in cool and wet weather the belt adjusts itself, the increased tension due to contraction raising the weight in the tower. A 10-h.p. motor drives each section. The lower section has a 6-ft. drop and requires approximately 5.1 h.p. to operate empty and 5.1 h.p.

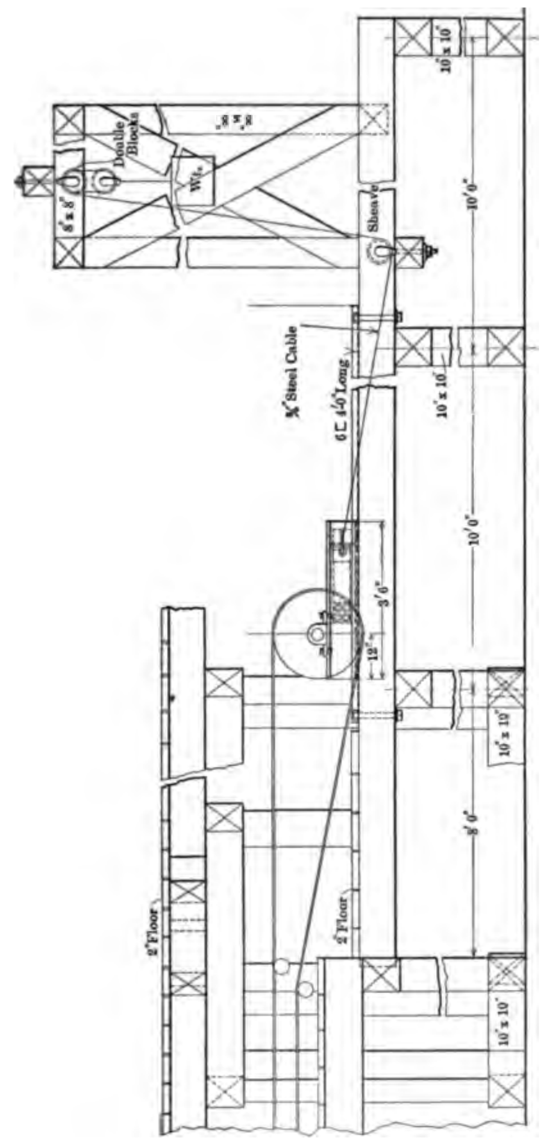

Fig. 3 Details of Take-up Arrangement for Belts

Fig. 4 View of Discharge from Upper to Lower Section

Fig. 5 Looking Down on First or Lower Section

Fig. 6 Side View of Upper Section

to carry a load of 1200 lb., as shoveled by ten men. (See tests which follow.) The discharge from the upper to the lower sections through a chute is shown in Fig. 4. There is no spilling of material at any point of the travel, and pieces of shale a cubic foot in size are carried. The upper section is driven, contrary to practice, at the upper end, the pull being on the lower or slack side of the belt, but in this case, due to the pull of gravity on the top side, the belt was found to work better with the pull on the lower side.

FIG. 7 GENERAL VIEW SHOWING BOTH BELTS

6 The several halftones give views of the belt taken from different points. In clearing a way through the woods, the poles obtained were utilized for trestling and the planking was obtained from the scrap pile of concrete-form lumber.

7 Fig. 6 is a side view of the lower end of the upper section, showing the two depressions in the belt, and though these depressions do not conform closely to the prescribed radius of 300 ft., there is no lifting of the belt from the carrying idlers.

8 Power tests were made on the two sections after the belt had been operating a few days, with the following results: the speed of

the belt of the lower section, which has a grade of 2.4 per cent for 665 ft., or $0.024 \times 665 = 16$ ft., was 146 ft. per min.; the belt was driven by a 10-h.p. direct-current Westinghouse motor, and was loaded 2.2 lb. per ft. for a distance of 550 ft., or 1210 lb.; this load fell 16 ft. in 5 min. Then

$$\frac{1210 \times 16}{5} = 3520 \text{ ft.-lb. of work exerted by load}$$

or,

$$\frac{3520}{33{,}000} = \frac{1}{11} \text{ h.p. (approx.) helping to pull the belt.}$$

When the belt was loaded as above, a test of the motor showed that 16 amperes, 239 volts, or 5.1 h.p., were required. There was no appreciable difference in the ammeter and voltmeter readings, when belt was empty or loaded, as in test.

9 When the belts were installed, after trying them out and ascertaining how easily they could be operated, a sprocket was placed on the tail-shaft of the lower section and also one on the head-shaft of the upper section, and the two sprockets were connected by a vertical quarter-twist chain. The idea was to drive both belts by a 10-h.p. motor at the head of the lower belt section, after all shafts had become well seated in the bearings and the stiffness had disappeared from the belt and it was in good operating condition. This was also necessary in order to take up the slack in the upper section when starting, and *the speeds were such that the top side of the belt ran 3 ft. per min. faster than the lower side.* The results of a series of tests are given in Tables 1 and 2.

TABLE 1 POWER TESTS OF BELTS UNDER CONDITIONS NOTED IN TEXT

Time (A.M.)	Volts	Amperes	Watts	H.P.	Notes
9:50	207	12	2484	3.3	Belts chained together
10:08	210	12	2520	3.3	Eight men loading
10:09	208	14	2912	3.9	Connecting chain off, 10-h.p.
10:11	210	14	2940	3.9	motor only
10:15	200	14	2800	3.7	Gradual increase in electrical load
10:20	200	15	3000	4.0	due to decrease in shale load
10:35	200	16	3200	4.2	

Note: Low voltage due to very small mains and long distance (2500 ft.).

TABLE 2 SECOND SERIES OF TESTS

Time (P.M.)	Volts	Amperes	Watts	H.P.	Notes
2:00	194	16	3104	4.1	Empty. Connected to lower belt by chain
2:15	180	16	2886	3.8	All readings at motor and not including line loss
2:25	182	18	3275	4.4	
2:35	186	18	3348	4.4	
3:45	195	14	2730	3.6	Loaded by seven men
3:50	185	19	3515	4.7	Loaded as before, but with connecting chain off. 10-h.p. motor only

Note: Readings taken on motor at upper end of upper belt-section.

Initial and Operating Costs

10 Tables are given herewith upon the first cost of the equipment (Table 3) and the cost of operation and maintenance (Table 4). Table 4 is based upon a capacity of 200 tons conveyed in ten hours. Inasmuch as the capacity is directly proportional to the speed, if it was desired to increase the capacity of the conveyor, it would only be necessary to increase the travel of the belt per minute, and from experience, it is quite possible that by doubling the load the power required to operate would be reduced 50 per cent.

11 The operation costs given in Table 4 are taken from actual practice. Doubling the capacity per day and assuming above costs to be approximately the same reduces the actual cost of conveying to $0.0038 per ton. Interest and depreciation, $0.0063, or a total of $0.0101.

TABLE 3 COST PER FOOT OF COMPLETED BELTS INCLUDING ELECTRICAL MOTORS, TRESTLING, ETC.

Materials	Total Cost	Cost per Ft.
Lumber	$ 496.34	$0.238
Belt	5361.52	2.58
Castings	1435.77	0.69
Electrical equipment, including two 10-h.p. motors	637.11	0.316
Miscellaneous: nails, bolts, screws, iron, etc.	193.20	0.093
Labor	962.20	0.46
	$9106.16	$4.37

Note: Length of first section, center to center, 998 ft.; second section, 1082 ft.; total, 2080 ft.; takeup, 15 ft.

Cost of castings includes machine work, etc.

12 Regarding the operation of the belt: after the stiffness had disappeared there was very little slipping at the head or drive pulleys, and there was sufficient lubrication in the shale itself to form a waterproof covering about ⅛-in. thick on the belt, thereby protecting it not only from wear but from the action of the elements, and proving a very good dressing to keep the belt pliable. Because of the slow speed, etc., there are very few repairs necessary to the belt, and in this instance, being coated as described above, the belt should last several years.

TABLE 4 COST TO OPERATE AND MAINTAIN BELT CONVEYOR

		PER 10 HR. DAY	PER TON
Power			
10 h.p. at \$0.004 per h.p.-hr		\$0.40	\$0.002
Labor			
Boy oiling, etc	\$0.75		
Taking up slack once in 7 days, 2 men, 3 hr. at \$0.20 per hr	0.171	0.92	0.0046
Supplies			
Belt Lacing	0.10		
Waste, Resin, etc	0.10	0.20	0.001
Total		\$1.52	\$0.0076
Oil (no charge, using waste oil from large crushers).			
Interest, etc.			
Interest, Depreciation, Renewals, 10 per cent on investment of \$9200		2.52	0.0126
Grand Total		\$4.04	\$0.0202

DISCUSSION

T. A. Bennett. Mr. Soper's paper, while giving a practical description of a certain installation, hardly seems to warrant the word "unique." There are many conveyors in regular practice just as long—conveyors which run downhill—in fact, conveyors that need a brake. As for size, there is a 36-in. movable belt conveyor in New York over a thousand feet long, used for filling in the refuse from the city on Riker's Island. The driving arrangement mentioned, from the receiving end of the conveyor, is also common practice.

2 Regarding the maximum length of conveyors, with a flat belt, as in this instance, the limit would be merely the cost of installation as practically any tensile strength desired can be obtained by increasing the number of plies of the belt. With a troughed belt the limit of length would be the tensile strength of the thickest belt that would conform to the trough of the idlers. This limit approaches somewhat the limits the author mentions, although such belts have been put in, as above, for lengths of a thousand feet or more.

3 The take-up has been in use for over five years in belt-conveyor practice. There is one installation at Bilbao, Spain, handling iron ore, which runs down an incline of 13 deg. and needs a brake, and has a counterweighted take-up working in a vertical plane. The take-up and drive are located on the return belt near the foot of the incline.

4 The tonnage of the conveyor is so small that the cost of maintenance per ton is also misleading. The wear of a belt is occasioned by the material coming in contact with it when delivered to it. A narrow stream of material permits each particle to come in contact with a small proportion of the total width, whereas a wide stream utilizes the full width of the belt and furthermore carries a large part of the material on top of the belt without ever touching it. I believe the capacity of this conveyor is something like 20 tons per hour, whereas such a conveyor should easily handle 200 tons per hour. Therefore the average cost for maintenance of the belt per ton carried is high.

HARRINGTON EMERSON. The last words in this paper are "the belt should last several years." In the last line of Table 4, it is stated that interest, depreciation and renewals amount to 10 per cent on an investment of $9200. Now, if the belt is to last only a few years, 10 per cent is not sufficient to cover interest, depreciation and renewal. Assuming that the belt lasts four years, the depreciation account alone would be $2300. That would increase the cost per ton from $0.02 to about $0.035.

FRED J. MILLER. It might be well to consider that there are other things that constitute part of this plant as well as the belt. As I understand, it is stated that the belt may last only a few years, but the rest of the plant may last enough longer than ten years to make 10 per cent a fair total charge for depreciation.

THE AUTHOR.[1] The belt conveyor in question has now been in operation about eighteen months, in which time less than $3 in repairs has been expended. The belt itself shows little wear and should last ten years. Of course the driving and carrying mechanisms will last indefinitely under ordinary conditions, as there is little wearing of the working parts, due to the slow speed of the belt.

2 The installation, as stated previously, though ample for a capacity of 200 tons per hour, is required to carry not over 20 tons per hour, and certainly the cost per ton for maintenance and other charges is out of proportion to what it would be were the belt carrying anything like full load.

3 The motor at the receiving end of the upper belt has been taken out and the belt has been driven for several months by the "lower" 10-h.p. motor, the "upper" belt being driven by the "quarter-twist" chain mentioned in the paper. At the time of writing the paper, this was to our knowledge, the *longest single-driven* belt conveyor (about 2150 ft.) in operation.

4 The writer has since learned of a slightly longer belt carrying grain. Its speed is 1800 ft. per min., and hence a much greater power is necessary. As to the power required to operate it, it is reasonable to assume that if it takes about 6 h.p. to operate the belt when empty, and 3 h.p. when loaded by 20 men, the belt will practically run itself when loaded by 40 men, and will require a band-brake when loading is increased above this number.

[1] Abstracted.

No. 1235

AUTOMATIC FEEDERS FOR HANDLING MATERIAL IN BULK

By C. Kemble Baldwin, Chicago, Ill.

Member of the Society

In the writer's paper on the Belt Conveyor, read before the Society in June 1908, mention was made of the advisability of using some type of automatic feeder when feeding a conveyor from bulk, for example, from a storage bin. This brief mention of the automatic feeder brought so many inquiries for information regarding feeders for various materials that this paper has been prepared in order to present a brief description of the various types now in use. The cuts are not intended to show the construction, but to illustrate the principle involved, so that they may be compared.

2 Careful study of this subject reveals the fact that a particular type of feeder has been developed in a certain industry, or locality, and is little used elsewhere. The types illustrated and described below are only those which have come under the writer's personal observation in many processes and locations within the past fifteen years. There may, therefore, be many other types.

3 When handling dry, free-flowing material such as grain, from a storage bin to a conveyor, a feeder is not required, as a simple gate may be set to give the desired opening, thus allowing the proper quantity to flow from the bin. Should the material be of varying size, such as mine-run coal, a simple gate is not satisfactory unless constantly attended; even then it is impossible to get the same constant, regular feed that a properly designed feeder gives. If the gate is raised high enough to allow a large lump to pass, there usually results a rush of fine material, which floods the conveyor before the gate can be closed. The automatic feeder, therefore, not only saves the expense of an attendant, but insures a constant and regular feed, irrespective of the size of the material.

Presented at the Spring Meeting, Washington, May 1909, of The American Society of Mechanical Engineers.

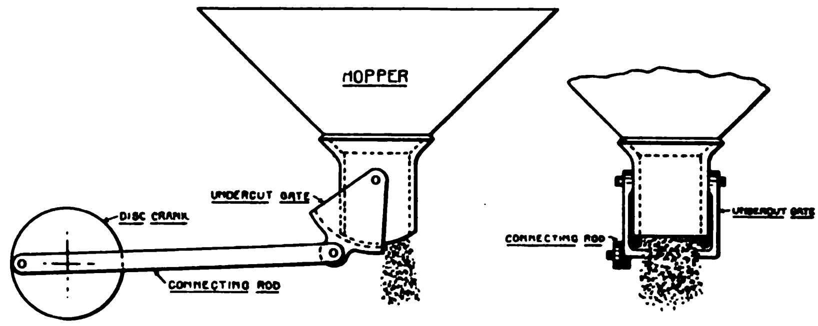

FIG. 1 UNDERCUT-GATE FEEDER

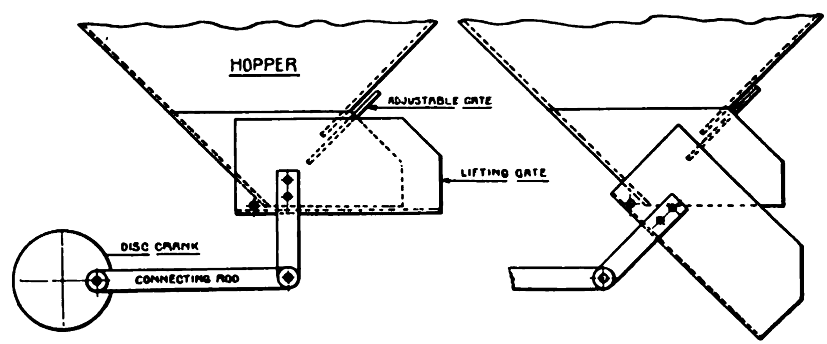

FIG. 2 LIFTING-GATE FEEDER

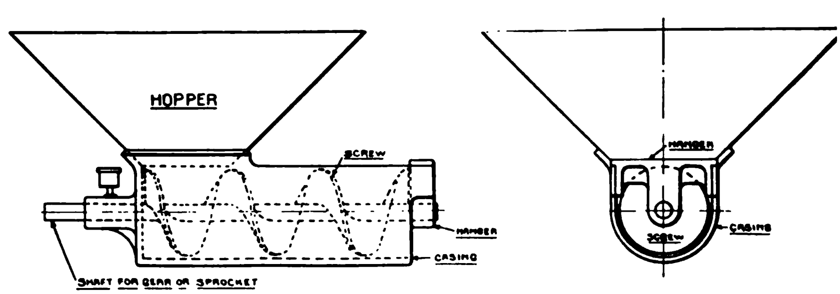

FIG. 3 SCREW-CONVEYOR FEEDER

4 Fig. 1 shows the undercut-gate feeder, with a body either of cast-iron or steel plate. Pivoted near the top is the undercut gate—which is swung back and forth by a connecting rod from crank or eccentric. This type of feeder is best adapted to fine-sized, free-flowing material. Material containing lumps is likely to bridge. As the feed is intermittent, the feeder is generally used in connection with chain or bucket conveyors, the strokes being timed to feed material between the flights, or into the buckets. The capacity may be changed only by changing the length or the number of strokes. As the length of stroke is more easily changed, it is preferable to use a crank with an adjustable throw rather than an eccentric. Elliptic gears are sometimes used to give a quick return, but in practice this quick return has not been found of sufficient value to justify the special and more expensive gears.

5 The lifting-gate feeder, shown in Fig. 2, also gives an intermittent feed and is therefore used only with a chain or bucket conveyor. The chute is hinged, so that when down, the material will flow out of the hopper, but when raised above the angle of flow of the material, the discharge is stopped. The moving of the chute may be accomplished by a connecting rod receiving motion from either crank or eccentric. This feeder will handle material regardless of size, but it must be free-flowing material, so that it will move by gravity when the chute is lowered to the angle of flow. The capacity may be adjusted by varying the number of strokes, also, in a measure, by increasing the length of the stroke, thus increasing the maximum angle of the chute and causing the material to flow more quickly.

6 The screw-conveyor feeder, illustrated in Fig. 3, will deliver a constant stream of material, but in this case also it must be of such a nature that it will flow by gravity to the screw. The capacity can be changed only by altering the speed of the screw shaft. This type of feeder has a large field in the handling of pulverized material, such as coal, cement, etc.

7 The roll feeder, shown in Fig. 4, is extensively used in the mineral industries for handling both large and small materials. The roll is so located under the hopper that the material will not flow when the roll is stationary, but when rotated it will carry the material forward. The capacity is determined by the speed and width of the roll, and the thickness of the stream, as fixed by the adjustable gate.

8 The roll feeder has been successfully used in handling iron-ore, coke and stone from the bins to the weigh cars for furnace changing. Edison used this type for feeding ore and stone from bins to crushing-rolls. The disadvantage is the head-room required, owing to

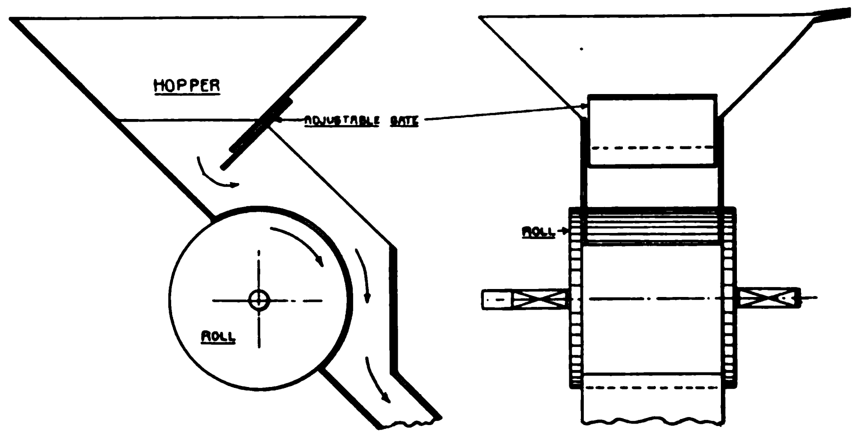

FIG. 4 ROLL FEEDER

the large roll necessary to satisfactory operation. For handling mine-run material, the roll should be 6 ft. to 8 ft. in diameter and in many cases it is not possible to obtain this space.

9 The rotary-paddle feeder, Fig. 5, acts not only as a feeder, but as a measuring device. It is used for fine material which flows readily from the blades. The capacity is fixed by the speed of the paddle shaft.

10 The revolving-plate feeder, shown in Fig. 6, is used mostly for feeding stamp-mills. The inclined plate driven by gears, placed either above (as shown) or below, moves the material out of the hopper

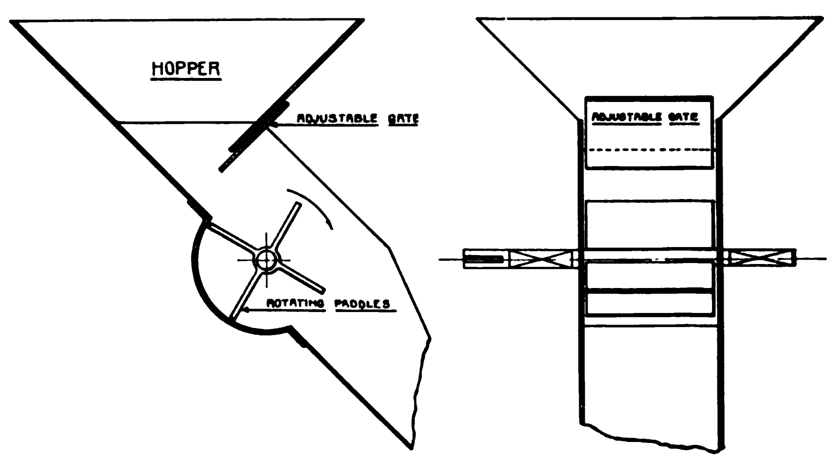

FIG. 5 ROTATING-PADDLE FEEDER

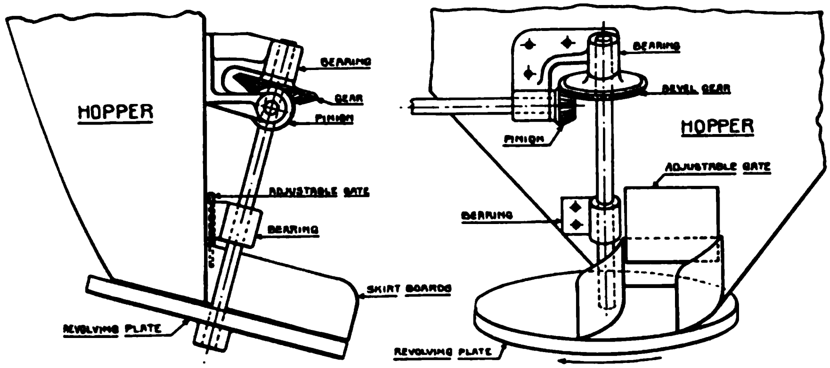

FIG. 6 REVOLVING-PLATE FEEDER

where it is scraped off by the skirt-board. When the skirt-board is made adjustable, sticky material may be handled by this feeder because the curved plate will scrape the material off the revolving disc and into the chute. The capacity is fixed by the speed of the plate and the location of the adjustable gate.

11 Fig. 7 illustrates the apron-conveyor feeder used for handling material of all sizes. The conveyor may be of any of the various types of apron flights, depending on the nature of the material handled. The chain should be provided with rollers or wheels traveling on

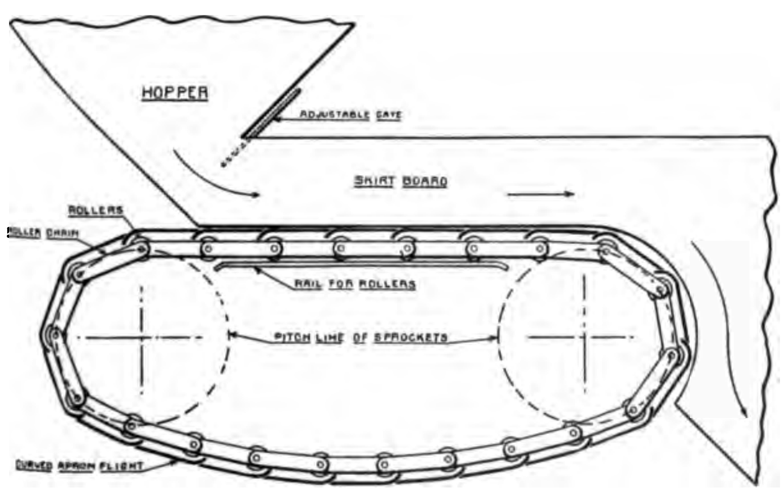

FIG. 7 APRON-CONVEYOR FEEDER

track to prevent the apron from sagging. The capacity is fixed by the speed of the apron and the position of the adjustable gate.

12 The disadvantage of this type is the inherent disadvantage of the apron conveyor. If the flights become bent or buckled, the material leaks through or catches between them. It has an advantage over other feeders in that it may be used to carry the material a greater distance.

13 A rubber or canvas belt may be used in place of the apron, in which case the belt is supported by idlers placed close together.

14 The swinging-plate feeder, shown in Fig. 8, is used for handling coal and such material of all sizes. It consists of two castings pivoted at their tops and swung alternately so as to move the material forward on the bottom plate. The plates are moved by connecting-rods

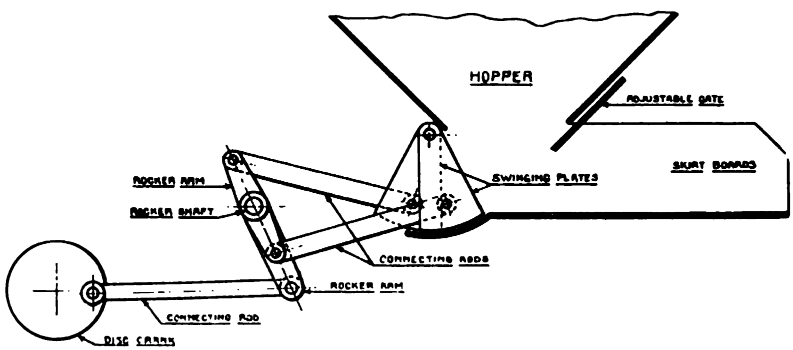

FIG. 8 SWINGING-PLATE FEEDER

from a crank or eccentric through a rocker shaft. The capacity is fixed by the length and the number of strokes, but as it is limited to the amount of material displaced by the plates, a wide range is not possible.

15 The disadvantages are the lack of adjustability and the tendency of the material to pack. It will also be noted that the feeder is not self-cleaning, so that the bottom plate always contains material which is very liable to freeze in winter.

16 The plunger feeder, illustrated in Fig. 9, is similar in operation to the swinging-plate feeder in pushing the material along the bottom plate. The plunger may be built either in one or two parts, moving ahead alternately and driven through a rocker shaft, as in the case of the one previously described. The capacity is fixed by the number and length of the strokes and the location of the adjusting gate.

This type has the same disadvantages as the swinging plate feeder, the most serious being that it is not self-cleaning.

17 Fig. 10 shows the reciprocating-plate feeder, consisting of a plate mounted on four wheels forming the bottom of the hopper. When the plate is moved forward, it carries the material with it, and when it is moved back the plate is withdrawn from under the material, allowing it to fall into the chute. The plate is moved by a connecting rod from crank or eccentric. The capacity is determined by the length and number of strokes and the location of the gate. The disadvantages are the lack of adjustment and the inability to clear the feeder of material.

18 The shaking feeder, Fig. 11, consists of the shaker-pan located under the opening in the bottom of the hopper at such an angle that the material will not flow when the pan is stationary. When given

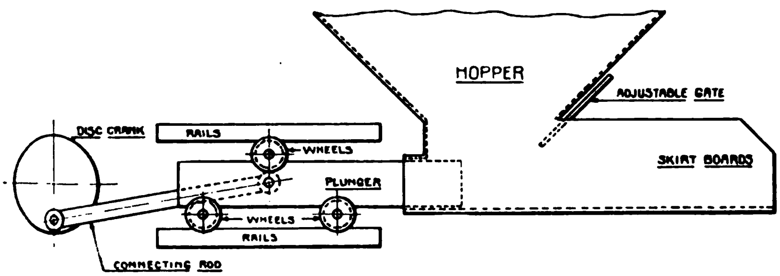

Fig. 9 Plunger Feeder

a reciprocating motion by the crank and connecting-rod, the material is moved forward on the pan. The front end of the pan is carried by a pair of flanged wheels; the back end is suspended by two hanger-rods, each being provided with a turn-buckle so that the angle of the pan may be varied. The crank having an adjustable length of stroke, there are three variables, viz: number of strokes, length of stroke, and inclination of the pan. As the number of strokes is difficult to change, and the others easily changed, the feeders are usually designed for about 75 strokes per min., a number determined by experiment. The angle of the pan is fixed by the capacity desired and the nature of the material handled. For coal, stone, ore, etc., 8 deg. to 10 deg. is sufficient, while clay and other sticky substances require from 15 deg. to 20 deg. The length of stroke varies from 4 in. to 12 in., so that a large range is possible.

19 A feeder designed to handle 400 tons per hr. of mine-run coal

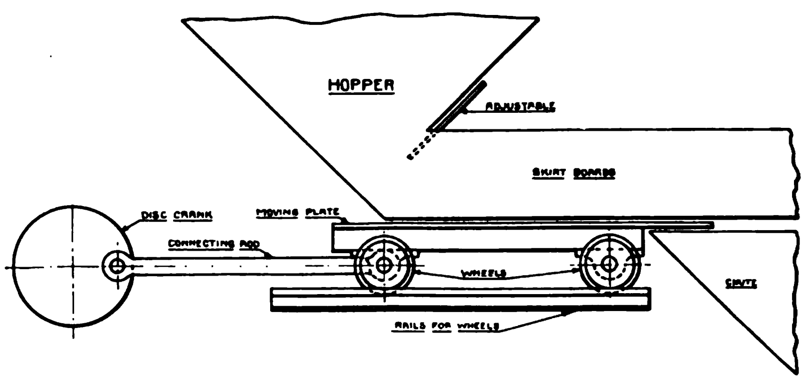

FIG. 10 RECIPROCATING-PLATE FEEDER

was changed in five minutes to deliver 30 tons per hr., by shortening the length of stroke and lowering the pan until nearly horizontal.

20 Not only has this feeder the widest possible range in capacity, but it is self-cleaning, a very important feature. From the cut it will be noted that the pan is placed under the opening and the material rests directly on the pan, so that when the pan is moved the material in the hopper is moved, which prevents the material from bridging.

21 The shaking feeder has none of the disadvantages of the other types for general use, and possesses many advantages which the others lack. Owing to its great flexibility it is more easily standardized

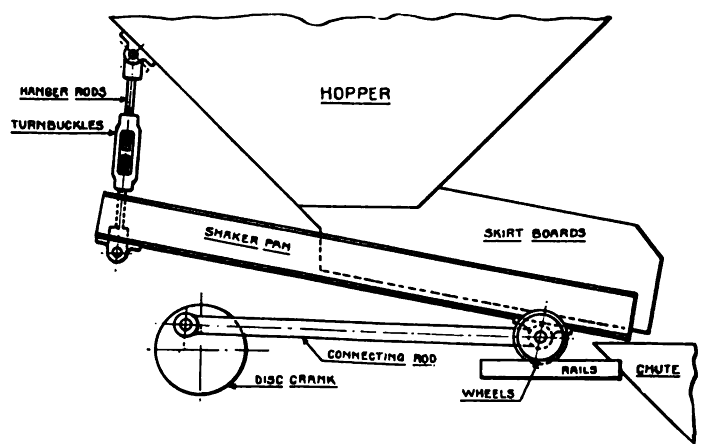

FIG. 11 SHAKING FEEDER

and will successfully handle practically any material, regardless of size or condition. If desired the bottom plate may be perforated to screen out the fine material, thus acting as both screen and feeder. This is not possible with any of the other types.

22 The power required by all of the types is so small that it is not an important consideration. The shaking feeder mentioned above, which handled 400 tons of coal per hr., required but 3.5 h.p.

23 The preceding cuts and descriptions will give a general idea of the different types and their possible uses, so that an engineer may readily choose the best type for the work to be done. The point that should be kept in mind is, that it is always advisable to gear the feeder to the conveyor, crusher, or other machine which it feeds so that they will both start and stop simultaneously.

DISCUSSION

T. A. Bennett. An automatic feeder is absolutely necessary in some installations; in others it is demanded for economic reasons. For example, run-of-mine coal, on account of constriction in the chute, requires properly a 36-in. belt, although it can be handled on a 30-in. belt. With an automatic feeder it is possible to use a 24-in. conveyor provided the capacity will permit. In handling damp sand, a large chute opening is necessary and this usually requires at least a 16-in. conveyor belt. With a feeder, however, this width can easily be reduced to 12 in. A 12-in. conveyor has capacity to take care of nearly every problem in handling damp sand.

2 The feeder is also economical in filling in the blank spaces on the belt. The loading is usually intermittent, the belt being either over-loaded or under-loaded intermittently; the feeder can be regulated to give a uniform maximum loading, greatly increasing the capacity. Intermittent loading also increases the wear on the belt. As the only wear worth considering is that done by the material coming in contact with the surface of the belt in the delivery of material to it, the larger the load the less is the proportionate wear on the belt.

3 A type of feeder has been developed similar to that in Fig. 8, but doubled; that is, a hopper and either one or two swinging plates at each end and pushing to an outlet at the center part of the skirt boards. This type is now working very satisfactorily in two large plants, the Hudson Company's power house at Jersey City, N. J., and the Illinois Steel Company at Joliet, Ill. The chief advantage over any other is the saving of headroom. Where there is sufficient headroom, the shaker feeder is correct practice and is in general use.

The Author. Mr. Bennett's discussion emphasizes three important points with reference to the use of automatic feeders in connection with belt conveyors:

a The installation of the feeder frequently permits the use of narrower belts.

b Delivery of an even and continuous stream of material enables the conveyor to operate at its maximum capacity.

c By loading the conveyor to its full capacity, a smaller proportion of the load comes in contact with the belt, thereby reducing the wear per ton carried on the belt at the loading point.

2 The type of feeder mentioned by Mr. Bennett as similar to that illustrated in Fig. 8 is a variation of that type, used where material is taken from two hoppers; instead of two swinging plates placed side by side, there is a single plate under each hopper. These two plates are connected by rods, so that when one plate is in the forward stroke the other will be in the back stroke. It is therefore nothing more than two single-plate feeders so connected that they operate together.

3 Mr. Bennett is mistaken regarding this type saving headroom over any other type. It requires about the same headroom as the plunger feeder (Fig. 9) and the reciprocating plate feeder (Fig. 10). The great drawback of the swinging-plate feeder is that it is not self-cleaning, so that if exposed in winter the material will freeze to the bottom plate.

4 This type of feeder was originated by Mr. Lincoln Moss of New York, and the two installations mentioned by Mr. Bennett were designed by Mr. Moss under the writer's direction.

No. 1236

A NEW TRANSMISSION DYNAMOMETER

By Prof. Wm. H. Kenerson, Providence, R. I.

Member of the Society

The author has received from time to time many requests for a simple transmission dynamometer, and has himself often felt the need of one which would be more generally applicable than those now in use. These continued requests, together with the requirements of a definite problem whose solution demanded a rigid transmission dynamometer in the form of a coupling, led to the design and construction of the instrument described below. The accompanying illustrations show the construction of the dynamometer and its method of application and use. In Fig. 2 and Fig. 4 the corresponding parts of the dynamometer are given the same letters and are referred to in the text.

2 The couplings *A* and *B*, each keyed to its respective shaft, are held together loosely by the stud bolts *C*. The holes in the flange *A* are larger than the studs *C*, so that these studs have no part in transmitting power from one shaft to the other. The power is transmitted from *A* to *B* through the agency of the latches *L*, four of which are arranged around the circumference of the flange *B*. These latches are mounted and are free to turn on the studs *E*. The two fingers of the latches engage the studs *F* on the flange *A*. On the ends of each latch are knife-edges parallel to the stud about which the latch turns. For either direction of rotation of the flange *A* the latches *L*, which are in effect double bell-crank levers, will exert a pressure on the disc *G*, tending to force it axially along the hub of the coupling *B*, and this pressure, it will be seen, is proportional to the torque.

3 Between the end-thrust ball, or roller, bearings *M M*, is held the stationary ring *S*, which is the weighing member. *O* is a thrust-collar screwed on the hub of *B*, and *P* is its check nut, which is ordi-

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narily pinned to the hub when in position. The stationary me *S*, in the form of a ring surrounding the shaft, is prevented rotating by fastening to some fixed object the attached arm s in the view (Fig. 1) of the assembled instrument. In this ring annular cavity covered by a thin, flexible copper diaphragı against which the ball-race of one of the thrust bearings pr The edge of this ball-race is slightly chamfered to allow some m

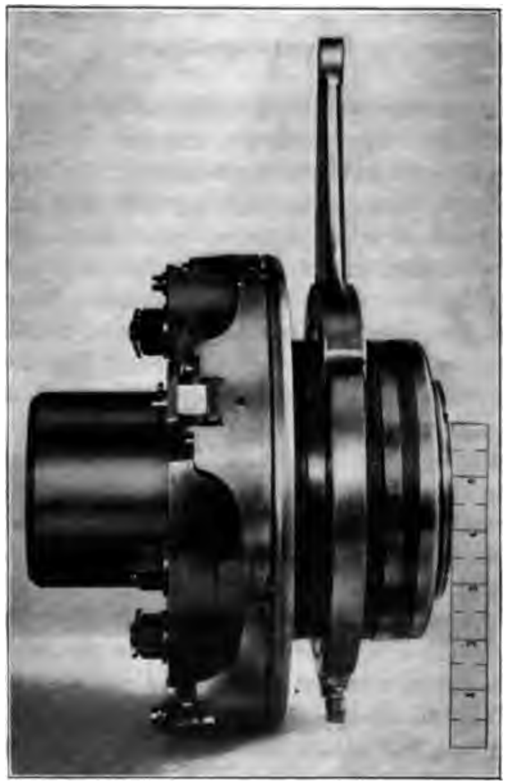


FIG. 1 DYNAMOMETER FOR 2-IN. SHAFT, WEIGHT 60 LB.

to the diaphragm. The cavity is filled with a fluid, such as oil connected by means of a tube to a gage. The oil pressure meas by the gage is proportional to the pressure between the thrust l ings, which in turn is proportional to the torque.

4 The instrument may be calibrated in the torsion-testing ma or by means of a sensitive friction brake. Fig. 6 is an actual bration curve for a small instrument, obtained by hanging stan

weights at proper distances from the shaft on a horizontal lever attached to the shaft, and reading the pressures indicated by the gage for the various torques shown in the diagram. For ordinary purposes, however, it is not necessary to calibrate the instrument by actual trial, since computations of the oil pressures for the various torques from the lengths of the lever-arms and diaphragm area check very closely those thus obtained.

5 It will be seen that the weighing means is similar to that employed in the Emery testing machine, which is recognized as being extremely accurate. It will be possible to employ the Emery flexible

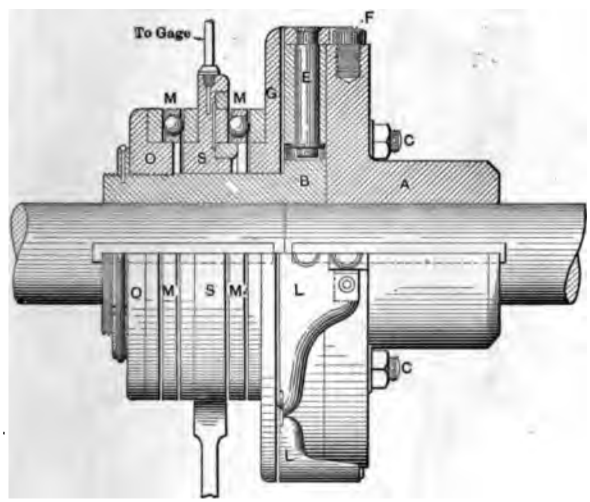


FIG. 2 DYNAMOMETER SHOWN IN SECTION

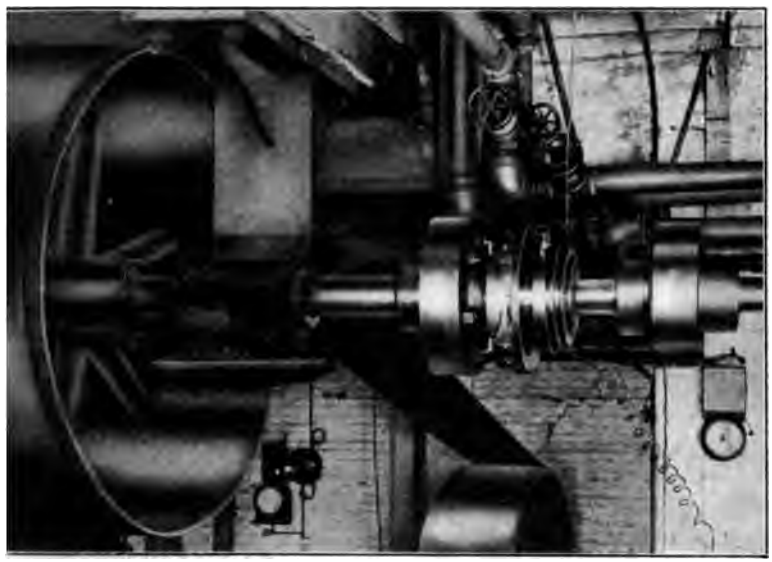
steel knife-edges on the levers, if desired, but this has been found in practice an unnecessary refinement.

6 The construction makes the coupling as nearly rigid as materials will permit, the movement of the diaphragm being extremely small. The only flow of oil through the copper connecting pipe is that sufficient to alter the shape of the Bourdon tube, if that be the form of gage employed. As soon as the normal position of the gage is reached this flow ceases, hence there can be no fluid friction. It is possible therefore, to use as long and as small a tube as desired, without introducing error. Where the gage is placed at a distance above or below the coupling, correction should of course be made for the static head.

FIG. 3 TRANSMISSION DYNAMOMETER IN AUTOMOBILE PROPELLER SHAFT, 30 H.P. AT 500 R.P.M., WEIGHT 25 LB.

7 Other means than the gage shown may be employed to measure he fluid pressure. Where extreme accuracy is desired it will be well employ the weighing device used with the Emery testing machine. ıe manograph has been used in this connection to measure variaıns in torque too rapid for indication by the ordinary gage. For ample, the variations in torque in a single revolution of the shaft a 3-cylinder gasolene engine have been recorded with its aid.

; Where the rate of rotation of the shaft is variable and it is



G. 5 DYNAMOMETER PLACED BETWEEN FLANGES IN MACHINE-SHOP DRIVE 3-INCH SHAFT. SPIRAL RUNNING TO THE WALL IS OIL PIPE TO GAGE

esired to indicate the horsepower direct, the combination of gage nd tachometer shown in Fig. 7 is employed. The hydraulic gage connected to the coupling described, its pointer therefore indicatıg torque. The pointer of the tachometer shows the number of evolutions per minute. Being a function of the revolutions per inute and the torque, the horsepower will be indicated by the intersection of the two pointers and suitable curves on the dial as shown. rrangements for recording or integrating the work done may also be ttached to the coupling.

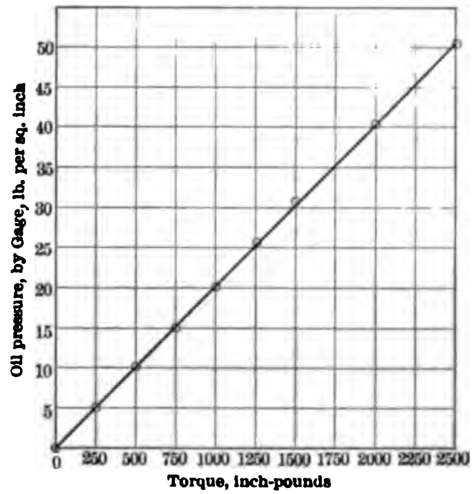


FIG. 6 CALIBRATION CURVE FOR TRANSMISSION DYNAMOMETER.

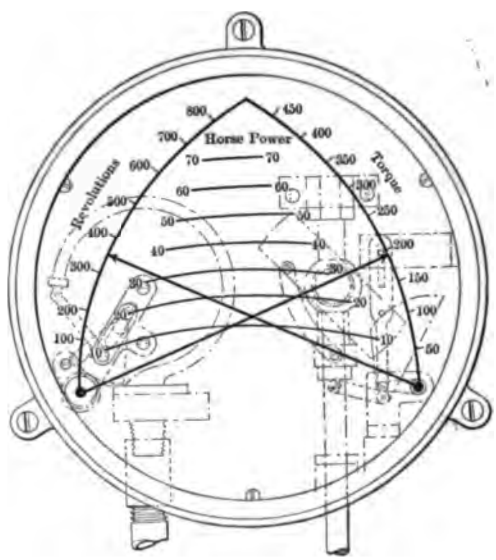


FIG. 7 COMBINATION PRESSURE GAGE AND TACHOMETER INDICATING T
REVOLUTIONS PER MINUTE AND HORSE POWER

9 A summary of some of the more important characteristics of the instrument follows:

a The instrument is compact. The example shown in Fig. 3 and Fig. 4, which is designed to transmit 30 h.p. at 500 r.p.m., is about 5¾ in. in diameter and weighs about 25 lb. That shown in Fig. 5 driving a 3-in. shaft is about 13 in. in diameter and weighs about 160 lb.

b It is as rigid as an ordinary flange coupling.

c It may be made in the form of a coupling, and will then occupy about the same space as the usual flange coupling, or it may be made in the form of a quill on which a pulley is mounted. This form may be made in halves for application to a continuous shaft.

d It will indicate for either direction of rotation of the shaft.

e The torque may be read and recorded or the work integrated at a considerable distance from the coupling.

f The readings do not require correction for different speeds of rotation. All parts containing oil are stationary, hence are unaffected by variation in speed. Other parts are likewise unaffected by centrifugal action.

g It may be made very sensitive and accurate. The construction lends itself very easily to variation of range of application and to varying degrees of sensitiveness, since the oil pressure, and hence the sensitiveness of the instrument, depend upon the area of the diaphragm, the relative lengths of the arms of the latches L, and the diameter of flanges. Its accuracy is dependent mainly on the degree of accuracy of the means employed to measure the fluid pressure, of which a number of forms, other than the usual pressure gage, are available.

h The only power absorbed is the small amount due to the friction of the ball, or roller, bearings, and this can be determined from the pull of the retaining arm. It is unnecessary to make correction for this, however, since the amount is so small as to be negligible.

i Since the only wearing parts are the ball, or roller, bearings, which may be lightly loaded, the instrument should not be deranged easily. Because of the very small volume of oil contained in the weighing chamber, ordinary temperature changes do not affect the calibration. All parts containing oil are stationary, hence all joints may be soldered and leakage entirely prevented.

j With suitable material and ordinary workmanship, it is believed that there is little likelihood of failure of any part of the instrument. It is conceivable, however, that the balls or rollers, although lightly loaded, might crush; the diaphragm might shear; or the stationary member, although bearing only its own weight and lubricated, might seize to the hub. Remote as are any of these possibilities, should any or all of them occur, the worst that could happen would be the tearing-off of the oil pipe and retaining arm, when the whole would revolve as a solid coupling. In no case can the coupling fail to drive the shaft because of its variation from the standard form, since, in addition to the driving latches employed to carry the load normally, the same number of connecting bolts may be employed as in the ordinary coupling, which will still hold the coupling together should the latches fail. Since, however, these latches are farther from the shaft, they should, if properly constructed, be less likely to fail than the connecting bolts usually employed.

10 It is believed that uses for the instrument here described will suggest themselves, and it is with the hope that the device will prove of some interest to those who deal with the use and transmission of power that the matter is presented to the Society.

DISCUSSION

A. F. Masury. I want to say a few words as to how the Kenerson dynamometer may be applied to the betterment of design in motor vehicles. In the first place, we must have exact data regarding the effect of road irregularities, wind pressures, and the resistances set up by grades and speed. These figures are absolutely necessary in order to determine the best torque to apply on motor gearing and equipment, such as tires, etc., to overcome the existing conditions in each particular car.

2 At present we have two recourses: first, the figures procured by Mr. S. F. Edge at the Brooklands track in England with his Napier car. He first calibrated his motor and then made the test on the track. These figures must necessarily include many errors. Second, the dynamometer at the Automobile Club of America in New York. Here again satisfaction is not entirely procurable as all the road conditions are obtained artificially by attachments on the machine.

3 With the Kenerson dynamometer we can certainly get exact readings while the machine is working on the road. These can even be made graphic if desired. Manufacturers will thus have available means of getting information which should result in more perfect design.

4 There is one thing more, the dynamometer of the Automobile Club of America cost, I believe, in the vicinity of $15,000 by the time it was completely installed, while the price of the Kenerson machine, around $500, will make it possible for even a small manufacturer to get his own reading.

The author desired to present no closure.—Editor.

No. 1237

POLISHING METALS FOR EXAMINATION WITH THE MICROSCOPE

By Albert Kingsbury, Pittsburg, Pa.

Member of the Society

In 1902 the writer made experiments to find the most suitable method of polishing samples of metals for microscopic examination The polishing of the surface is one of the most important as well as most troublesome details of metallography, particularly when high magnification is required.

2 At the outset, trials were made of all the methods of which descriptions have been published. Some of those methods have been successfully employed by various metallographists, as shown by numerous reproductions of excellent micro-photographs in different publications. Nevertheless the writer did not find any of these free from objectionable features. The ideal method should produce a fairly flat surface, free from excessive relief of the harder constituents, rounded edges at flaws, or scratches and smearing of the metal. The method should be simple, the materials employed readily available, and the process as rapid as consistent with the first-named requisites. None of the published methods embodied all these requisites, nor is a perfect method likely to be found. However, the method finally developed by the writer appears to him superior.

3 The preliminary trials were made with rotating discs covered with various materials, including canvas, felt, silk, leather, chamois, parchment, paper, wood, pitch, asphalt, resin, shellac, beeswax, etc. The polishing powders included commercial abrasives, such as emery, carborundum, tripoli, crocus and jewelers' rouge; also precipitates, such as carbonates and sulphates of the alkaline earths. Attempts were made to obtain fine finishing powders by the levigation process from commercial abrasives. These abrasives were tried both wet and dry and with various speeds of the discs. Hand polishing was also tried.

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It is needless to detail the objectionable features encountered, which are probably familiar to all metallographists.

4 The method finally adopted was the result of two distinct discoveries: (*a*), that ordinary paraffin wax makes a good polishing bed; (*b*), that excellent polishing powders of certain grades are commercially available.

5 The paraffin is used as a facing for rotating discs of metal, preferably brass, about 8 in in. diameter. The discs are grooved on the flat face for anchoring the wax. To prepare the discs, they are warmed to about 100 deg. cent., and laid flat, and the melted paraffin is poured on them to a depth of about ½ in., a removable ring or band retaining the melted wax. The whole is then covered to exclude dust and allowed to cool. After the wax has solidified it may be dipped in water to hasten the hardening. Since the wax has very little viscosity when melted, all hard foreign particles, which might produce scratches in the samples, settle out before the wax hardens, the elimination being practically complete. No advantage in this respect was gained by keeping the wax in a fluid condition on the disc for several hours in an oven. After the hardening of the wax. the discs are placed on the spindle of the polishing machine and the face of the wax is turned true and flat by a hand-tool.

6 In the writer's machine the spindle was horizontal and four discs were used for abrasives of progressive fineness, two discs being placed back to back at each end of the spindle. The disc used for the final polishing should not be perforated and the wax should be continuous to the center of the disc, as that part is best for the finishing touches to the sample. This latter disc should be at the right-hand end of the spindle. The speed of rotation should be about 200 r.p.m.; a higher speed throws off the polishing powder with the water used, and a lower speed makes the work too slow. A stationary sheet-metal strip about 3 in. wide bent over the discs serves as a screen.

7 The polishing powders, in the order used, were as follows: (*a*) commercial flour of emery; (*b*) washed Naxos emery, 3/0 grade; (*c*) washed Naxos emery, 7/0 grade; (*d*) soft optical rouge, light grade. These were obtained from the George Zucker Co., New York, except the first, which is available everywhere.

8 The emery powders were mixed to a paste with water in tall glass jars provided with covers; the paste was applied to the rotating discs with small brushes as required, the brushes being kept in the jars when not in use. The rouge was in cake form, best applied by holding a small piece in the hand, wetting both the rouge and the wax, and pressing the rouge lightly against the rotating surface.

9 A small quantity of water is required throughout the polishing process, but water cannot be used very freely without wasting the powders. The water is best applied as required, from an ordinary chemist's wash-bottle, held in the left hand while the right hand manipulates the sample. No water pipes or drains are required for the polishing machine. Distilled water may be used if available. If tap water is used, it should be drawn into large jars provided with covers and siphons, and allowed to stand a day or more before use, in order that all gritty particles in suspension may be deposited. The inner ends of the siphon tubes should be at least 3 in. above the bottom of the jars.

10 The treatment of the samples is as follows: the samples are first dressed to shape and size by any convenient method, the surface to be polished made flat by an emery wheel or file, and the sharp edges rounded to prevent cutting into the wax. The dimensions of the samples should depend to some extent upon the coarseness of structure. For normal iron and steels, and for much other work, a ½-in. cube is a convenient sample. Massive castings sometimes have grains an inch or more in diameter, and correspondingly large samples are required. The samples are held flat against the waxed discs, which are kept well covered by the polishing paste, using successively the flour of emery, the 3/0 emery, the 7/0 emery, and the rouge, on the several discs. At each grinding with emery the sample should be held without rotation and with a slow transverse motion across the face of the disc until the grinding marks show over the entire surface. The sample may then be given a quarter turn, so that the new marks cross the old ones, and so on. The discs must be kept wet continually while grinding. With each grade of powder the grinding should continue for some time after the marks of the last previous grade have disappeared, especially with soft metals, since the scratches cause a flow or disturbance of the metal to a minute depth below the surface, and if this disturbed metal is not ground off, the deep effect of the scratches becomes apparent on etching. In the final polishing on the rouge disc, the sample should be continuously rotated; this is most readily done by moving the sample nearly in a circle about the center of the disc in an opposite direction from the rotation of the disc. This keeps the direction of the grinding marks constantly changing, and avoids grooving. The finishing should be done near the center of the disc, the slower motion being most effective for very fine polishing. After grinding with one grade of powder and before proceeding to the next, the samples and the operator's hands should be thor-

oughly washed; and the hands and the apparatus should be kept free of dust or dirt, to secure a polish free from scratches.

11 The most important item to be noted by the beginner is the liability of the paraffin to adhere to the samples when the grinding is begun, particularly in the case of the rouge disc. When the sample is first brought into contact with the disc, especially if the latter has been freshly prepared, the paraffin nearly always smears over the surface of the sample in a second or two, and if the sample is not removed and cleaned at once the result is a roughened disc, requiring re-turning with the hand-tool and re-application of the paste. Therefore the sample should at first be touched very lightly to the disc, and at once removed and wiped with the finger, or with a cloth. If this is repeated several times, the surface of the sample will no longer become coated with paraffin but can be ground continuously, except when a fresh coating of paste is required by the disc. One great advantage of the paraffin disc over discs covered with cloth or felt, is that if the disc becomes roughened or cut, it can readily be turned smooth and true again.

12 For cleaning the samples after polishing, the best material is a stock of old linen or cotton cloth well-laundered and cut to 3-in. squares. These small pieces are preferable to larger ones, since they can be discarded for fresh ones after once using. The old cotton or linen is also the best material for cleaning the lenses and mirrors of the optical apparatus, being superior to chamois for this purpose.

13 The time required for polishing a sample varies somewhat with the hardness. A single sample of normal steel, cast iron, or wrought iron, may be finished in fifteen minutes; a set of five or six such samples may be finished in an hour. Hardened steels require a slightly longer time. The method has not thus far proved serviceable for very soft metals and alloys, particularly lead, owing to the persistent adhesion of the paraffin to the surface of the sample. The harder alloys polish well by this process. The finished surface presents a minute relief of the harder constituents, but much less than is produced by the use of felt or other very soft materials.

14 The paraffin beds are more durable than might be supposed; on long standing at summer temperatures the surfaces become distorted by the flow of the wax, but they can always readily be made true by the turning tool. The harder paraffin (ceresin) offers no advantages over ordinary paraffin, except that it flows less at summer temperatures. It is serviceable for use with the emery powders but too hard for best results with the rouge.

No. 1238

MARINE PRODUCER GAS POWER

A COMPARISON OF PRODUCER-GAS AND STEAM EQUIPMENTS

By C. L. Straub,[1] New York

Non-Member

So much interest is exhibited both by the engineering profession and the general public in the application of producer gas power to marine, commercial and naval service, that a brief summary of recent progress in this field appears timely.

2 Any innovation which makes for improvement in present practices, surely, though sometimes slowly, achieves its end. Producer gas power, on impartial analysis, offers so many benefits to marine service that it appears strange indeed that more rapid progress has not been made in its adoption. The delay appears to be due to several causes.

3 The marine public, which since the days of the Clermont has exclusively associated the term "motive power" with steam, has every reason for demanding exact and conclusive evidence of the superiority of gas power or any other power, before adopting it in lieu of present methods. This evidence is only now slowly coming forth. Many who have been credited with authority by the engineering profession and others, either through ignorance or through misinformation, have beset the way of marine gas power with numberless imaginary obstacles, ridiculous in proportion to the real difficulties, but sufficient nevertheless to instill some doubt of the possibilities of the system into the minds of the waiting public.

4 Only recently has such progress been made in the development of gas power for marine work, as to warrant its early adoption in commercial service. Two years ago, less than 300 h.p. in the aggregate was being developed by marine producer gas power installations; these were experimental in nature and were of the German Capitaine

[1]With the Loomis-Pettibone Co., New York.

Presented at the Spring Meeting, Washington, May 1909, of The American Society of Mechanical Engineers.

type. There are now installed and accepted 23 Capitaine marine plants, aggregating 2035 h.p., a partial list of which follows:

a Emil Capitaine: Launch, 60 b.h.p.; 4-cylinder single-acting, 4-cycle engine; boat 60 ft. long, 10 ft. beam, 4 ft. draft; ran an average speed of 10 miles for 10 hr. on 412 lb. of anthracite coal.

b Rex: Sea-going Swedish boat; 102 ft. long; 22 ft. beam, carries 350 tons on 9-ft. draft; fitted with a 3-cylinder single-acting, 45-h.p. engine at 300 r.p.m.

c Capitaine: Tow boat at Genoa; length 47 ft., beam 12 ft., draft 7 ft.; fitted with a 3-cylinder, single-acting, 4-cycle engine, 105 b.h.p. at 240 r.p.m.

d Duchess: Canal barge; length 71 ft., beam 7 ft. 1 in.; carries 20 tons cargo on 42-in. draft; fitted with double-cylinder, single-acting, 4-cycle engine of 25 b.h.p.

e Dusseldorf: Tug at Hamburg; fitted with a 4-cylinder, single-acting. 4-cycle engine, 60 b.h.p. at 240 r.p.m.

f Isee: Tug, fitted with a 3-cylinder, single-acting, 4-cycle engine, 45 b.h.p., 300 r.p.m.

g Wilhelm: Combination freight and passenger Rhine boat, fitted with a 5-cylinder, single-acting engine, 175 b.h.p. at 240 r.p.m.

h Badenia: Rhine freight boat, fitted with a 2-cylinder, single-acting, 4-cycle engine of 30 b.h.p.

i Katrina: Canal freight boat, fitted with a 3-cylinder, single-acting, 4-cycle engine, 45 b.h.p.

j Marie: Canal freight boat; fitted with a 3-cylinder, single-acting, 4-cycle engine, 45 b.h.p.

k Hoffnung: Combination freight and passenger Rhine boat, fitted with a 5-cylinder, single-acting, 4-cycle engine of 210 b.h.p.

l Amersie: Volga freight boat, fitted with a 4-cylinder, single-acting, 4-cycle engine of 60 b.h.p.

m No. 58: Canal freight boat, fitted with a 4-cylinder, single-acting, 4-cycle engine of 60 b.h.p.

5 In addition to the above there were a number of freight boats, the dimensions and names of which we were unable to obtain, but whose power plants varied in capacity from 30 to 175 h.p. each.

n H. M. S. Rattler: An old gun boat, 165 ft. long, 29 ft. beam, originally fitted with a triple expansion engine. The gas engine is 5-cylinder, single-acting, 4-cycle. Cylinders 20 in. diameter by 24 in. stroke, developing 500 b.h.p. at 120 r.p.m. This engine is started by means of a mixture of gas and air which is pumped into the cylinders at a pressure of about 95 lb. per sq. in. This complete plant was designed entirely in the Capitaine Works at Düsseldorf. The total weight of the entire plant, including the donkey boiler for working the pumps and auxiliaries, is 94 tons, as compared with 150 tons in the case of the displaced steam engine. A consumption of 1525 lb. of coal was made for a measured distance of 45 knots on an average speed of

10½ knots per hr. The cost per mile for fuel with coal at 15*s.* 6*d.* per ton is $0.064 U. S. currency. This boat made a maximum speed of 11.3 knots per hr. against a 1½ knot current at 110 r.p.m. of the engine shaft.

6 All of the above plants by their design and construction are restricted to operation on anthracite coal, coke or hard-burned charcoal, and any plant so restricted by its design to one class of fuel is seriously limited in its scope of application. The development of a simple marine gas-producer for use with any class of solid fuel is a necessity, if the system is to be considered seriously by the marine profession.

7 The writer is fortunate in having been associated with some recent American developments both in stationary and marine gas-power plants, a brief survey of a portion of which will enable us to draw more clearly the comparison between a typical steam and a possible gas installation.

8 There are in commercial operation in this country today, two distinct types of stationary power gas-producers which are suited by their design for operation on almost any class of solid fuel. They may, by their systems of operation, be qualified as up-draft and down-draft producers.

9 In the up-draft producer, the fuel is charged into the generator through an air-tight mechanism at the top, while air and steam, or air and products of combustion are admitted at the bottom of the fuel bed, and passing upward, leave the generator at the top in contact with the fresh fuel. Almost all of the hydrocarbons are unfixed when leaving the generator with the hot gas, and are condensed later in the gas coolers or scrubbers and gas mains, forming large amounts of tar, which, if not removed to a minute degree, will positively prevent the operation of the engine. The removal of this tar is troublesome and is accomplished at a loss of power and efficiency. The fuel in the upper zone of the bed in the up-draft producers cokes and cakes so seriously as to require continuous poking of the fuel bed, either mechanically or by hand. These features and others in this type of apparatus contribute to limit the rates of combustion per sq. ft. of grate to a relatively low quantity. All things considered, therefore, this type of apparatus has not lent itself agreeably to modification for marine service.

10 In the down-draft type of apparatus, the fuel is charged by hand through a large door at the top of the producer, which is normally in an open position, allowing the operator unrestricted inspection of the whole upper zone of the fuel bed. The hydrocarbons con-

tained in the fuel are driven off in the upper zone, mixed with air and almost completely burned, and the burnt products, passing downward through the relatively deep bed of fuel, are decomposed and regenerated into carbon monoxid and hydrogen gases. All of the tar and the lighter hydrocarbons are completely fixed in this process, and no tar is found in condensation in any portion of the plant after cooling. Coking or caking of the fuel bed is not detrimental, but on the other hand assists in keeping the fire in the open porous condition, which is desirable and necessary where high rates of combustion obtain. This feature eliminates the poking necessary in the up-draft apparatus. The gas leaves the bottom of the producer through brick-lined connections, and a portion of the sensible heat is extracted in passing through an economizer. The gas is then cooled and washed and passed through an exhausting mechanism, whence it is delivered under pressure to the engine.

11 This type of apparatus lends itself admirably to the high rate of fuel combustion, which for the sake of economy in space and weight is desirable in marine service. There are in actual commercial operation today, a number of plants of this type having an average fuel consumption of over 40 lb. of good bituminous coal per sq. ft. of grate per hr. These producers are sold on a rating of from 18 lb. to 20 lb. of fuel per sq. ft. of grate per hr., which is almost 100 per cent greater than the average rating of the up-draft type of producers.

12 Undoubtedly a better method of measuring the ability or success of these two systems, is to make note of the number and capacity of plants of each type in actual operation on engine service. A report of the committee on gas engines of the National Electric Light Association, spring of 1908, showed that in gas-engine power plants, of capacities of over 300 h.p. each, there were in operation 32 plants of both types having a total capacity of 57,225 h.p. Of these, 4 plants were of the up-draft type, having an aggregate capacity of 4050 h.p., and 28 plants were of the down-draft type, with an aggregate capacity of 53,175 h.p. The latter contain the Loomis-Pettibone gas-generating apparatus, some of which has been in operation on engine service for 13 years.

13 Three years have been devoted to the modification of these stationary plants for marine service. The work involved a reduction in the size and weight of the generators; complete revision of the scrubbing, gas cleansing and exhausting mechanism; elimination of all gas holders, storage receptacles, mixing chambers, etc.

14 The plant as modified to date has a light compact producer,

which while retaining the same rate of combustion as the stationary apparatus, has materially reduced dimensions and weight of the shells, brick lining, fittings, etc. The economizer boilers which were used on stationary work have been abandoned, and replaced with light air-heating economizers. The gas coolers no longer contain any coke or broken material, or wooden trays, and are built of very light, non-corrosive sheet metal, and arranged for either vertical or horizontal positions, the latter arrangement being convenient for space which would be otherwise wasted in the vessel. The cooled and partially cleansed gas is drawn through the above portion of the plant by a centrifugal gas-cleaning exhauster, driven by direct-connected motor. The gas passes directly from the exhauster under pressure, through an automatic pressure-regulating valve, to the engine manifold.

15 That the plant is adaptable for marine service with regard to space occupied and weight, may be seen from the following conservative estimate:

> Plants of from 100 to 500 h.p. each occupy from 0.4 to 0.5 sq. ft. per h.p., and weigh from 70 lb. to 90 lb. per h.p., including all auxiliaries, piping, etc.; plants of from 500 h.p. to 1000 h.p. occupy from 0.30 sq. ft. to 0.45 sq. ft. per h.p., and weigh from 40 lb. to 70 lb. per h.p., including all auxiliaries, piping, etc.

16 Undoubtedly the rational opportunity at the present time for marine gas power lies in commercial service, in which regard the most rapid advancement in America has been made in the freight, ore and fuel carriers of the Great Lakes.

17 We have therefore taken for our example a ship built from the designs of Messrs. Babcock & Penton within the last year. For the sake of clearness, the views show only the machinery space; all of the ladders, stairways and grates have been omitted from the plans, and the piping is shown only on the gas installation. The machinery installation proper is all there, however, and while the parts eliminated are merely accessory, the contrast between the two plants would be all the more striking were they included.

18 The boat is a modern lake freighter and represents the best standard practice in this service. She is 306 ft. long over all, 45 ft. beam and 24 ft. deep. Her present power equipment consists of a single-screw, triple-expansion, three-crank condensing engine, 18–30–50 by 36-in. stroke, which indicates 1050 h.p. at 90 to 95 r.p.m. The vessel is fitted with a jet condenser and has independent

steam-driven reciprocating. bilge, sanitary and feed pumps. The complete engine room weight, including piping and all auxiliaries, is, in round figures, 182,000 lb.

19 The boiler room equipment consists of two single-ended Scotch boilers 11 ft. 10 in. mean diameter, 11 ft. length over heads, operating on a working pressure of 180 lb. per sq. in. Each boiler is fitted with two 42-in. corrugated furnaces and has two hundred and forty-four $2\frac{3}{4}$-in. tubes. The grate surface is $36\frac{3}{4}$ sq. ft. and the heating surface 1642 sq. ft. in each boiler.

20 The boilers are fitted with forced draft from a 66-in. steam-driven fan. The air for the draft is taken from the stoke hole and the fan is located in the engine room. The fan discharge passes through air heaters in the up-take and thence through ducts to the under side of the grates. The complete boiler-room weight, including water in the boilers, but not fuel, is 170,000 lbs. These weights are actual rather than mere estimates.

21 The coal bunker extends from the main deck to the tank top and is arranged athwartship. It has a capacity of 170 tons. The bunker doors face the stokers on the stoke hole floor. The bunker is 6 ft. fore and aft at the stoke hole. The distance from the forward to the after bulkhead in the boiler room is 24 ft. 0 in. The distance from the forward to the after bulkhead in the engine room is 22 ft. 0 in., making a total over-all length for the plant, including bunkers, of 52 ft. 0 in.

22 The coal consumption on this vessel is from 1.80 lb. to 2 lb. per i.h.p. hr. This coal is of approximately 13,500 B.t.u. per lb.

23 The problem of substitution of gas for steam, aside from the design of the construction of the gas producers or cylinders of the gas engines, has been thoroughly worked out by Messrs. Babcock & Penton, of Cleveland. The illustrations show two different arrangements of gas producers with the same engine. The proposed gas engine is a four-cylinder, double-acting, reversing type, having cylinders 24 in. bore by 36 in. stroke, delivering 1000 b.h.p. at 100 r.p.m. The reversing is accomplished by means of compressed air, which is used to shift the cams from the head to the stern position. Compressed air is admitted to the cylinders by timed cams in proper cycle. The crank shaft of the engine is rigidly coupled to the tail shaft of the screw.

24 The illustrations show a column-framed engine. Since making this layout, the design of the engine has been modified to meet all of the present marine conditions now found in marine engine design on the Lakes. In fact, with the exception of the condenser shown on the

steam drawings, the gas-engine frame will be very similar to that of the steam engine.

25 For the generation of current to drive the auxiliaries, there will be installed a double-cylinder, double-acting gas engine, direct-connected to a 50-kw. direct-current generator. All of the pumps and auxiliaries will be motor-driven. A smaller direct-connected unit operating on oil will be used for pumping, blowing fires, or other service, when the gas plant is down. Allowing a distance of 4 ft. 3 in. between the forward bulkhead in the engine room and the forward side of the flywheel, which distance is one foot greater than that in the steam installation, we have an over-all distance between forward and after bulkheads in the engine room of 19 ft. 6 in.

26 As previously stated, two arrangements of the producer room are shown. The first, the four-generator plant, consists of four 6 ft. by 9 ft. generators, each fitted with independent economizers. The forward pair and the aft pair are connected independently to two horizontal gas scrubbers which are shown slung under the main deck beams. The gas passes from these scrubbers to independent motor-driven centrifugal gas-cleaning fans, whence it is delivered, either through common connection to a purge or blow-off pipe which also acts as a by-pass, or through two gas pressure regulator valves to the air and gas mixing valve at the engine manifold. The 6 ft. generators require only one cleaning door each. As a result a single cleaning space suffices for the four machines, allowing them to be grouped with reference to athwartship space, so as to give ample room on each side of the vessel for coal bunkers. The total space occupied by the producer plant is 21 ft. 10 in. athwartship, and 15 ft. between forward and after bulkheads. The producer room weight, including generators, economizers, piping, and scrubbers, complete, of the four-generator set, is 110,000 lb. This weight is estimated, but has been carefully checked and completely covers all the mechanism. In addition to the above mechanism, there will be a heating boiler which is shown on the main deck. This boiler will serve to furnish low-pressure steam for heating the vessel and supplying hot water for washing down decks, etc. This boiler, with water, will weigh about 8000 lb.

27 The two-generator producer plant, which will undoubtedly be the one installed, will consist of two 8 ft. diameter by 9 ft. 6 in. generators, connected to independent air economizers and each fitted with an independent horizontal scrubber, located athwartship under the main deck beams. The gas outlet at the scrubbers will be connected with a cross-over, so that either exhauster may operate either or both

producer plants. The exhausters are installed in duplicate and are connected with common purge or blow-off and common gas outlets leading either through one pressure-regulator valve, or through a by-pass direct to the air and gas mixing valves at the engine manifold.

28 On account of the fact that the 8-ft. generators require two cleaning doors set at 120 deg. the double generator unit plant will require the full athwartship space in the producer room. The approximate floor space occupied, therefore, will be 30 ft. athwartship and 15 ft. between forward and aft bulkheads. The producer-room weight, including generators, economizers, piping and scrubbers complete for the two-generator set, is 82,000 lb. This weight is estimated, but has been carefully checked and completely covers all of the mechanism. As in the case of the four-generator plant, a low-pressure boiler for heating service will be installed. In the two-generator plant, however, this boiler will be located on the producer-operating floor, so that one set of firemen may suffice for both.

29 The only guide we have for estimating the probable fuel consumption for this service is found in the large number of stationary producer gas power plants now in operation. Fortunately, in marine service, the load factor will be uniformly much higher than that found in any stationary service to which gas power is applied at the present time. The builders of this apparatus are prepared to guarantee one brake horse power per hr. on *one* lb. of good bituminous coal, averaging 13,500 B.t.u. per lb.

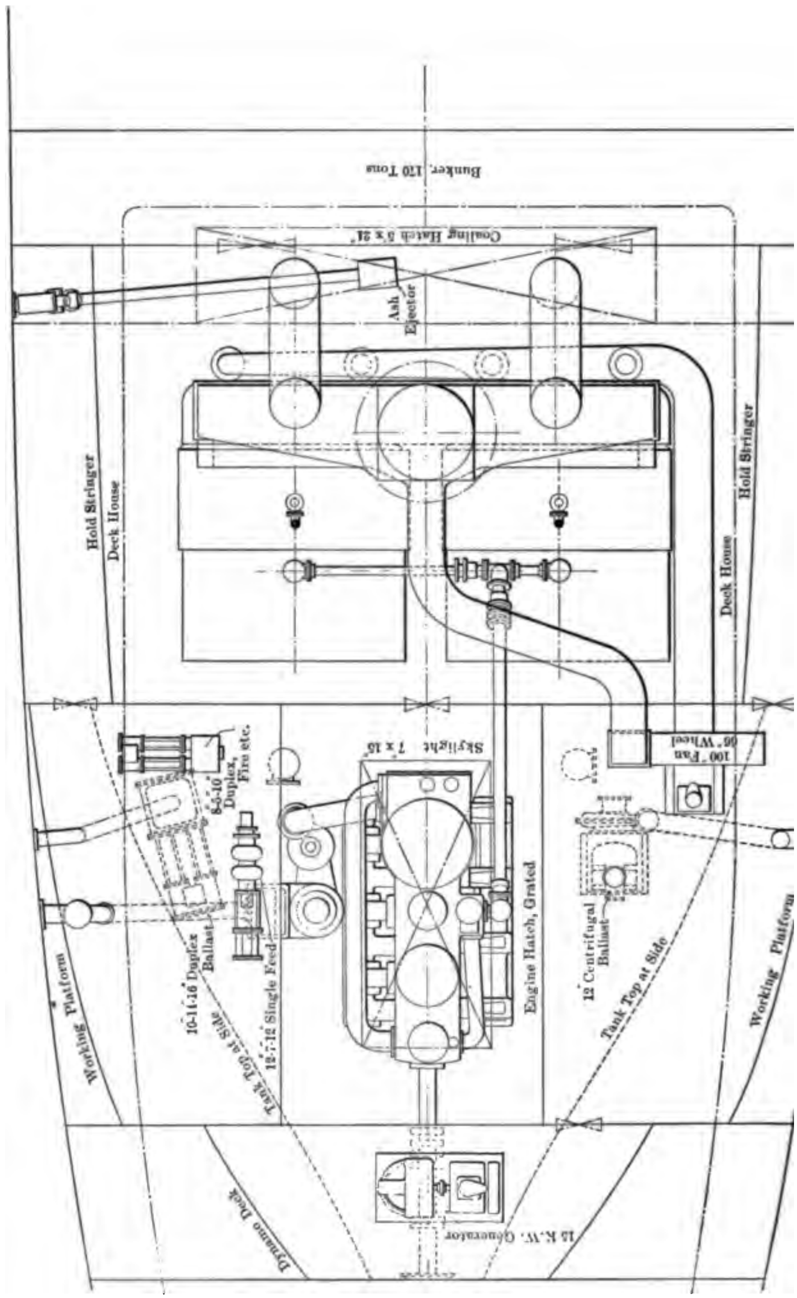
30 Messrs. Babcock & Penton, the engineers who designed and built the steam plant, and who have spent years on the problem of the substitution of gas for steam, have suggested that the coal bunker, which will be placed above the charging deck of the producer, should have a capacity of about 80 tons of coal. These bunkers will run from the charging deck to the deck-house and will have doors opening closely adjacent to the charging doors of the generators, so that little or no coal passing on the operating deck will be required.

31 In making the comparison shown in the table, it is unnecessary to go into the cost of fuel, labor, hours of service, etc., as these elements vary with every class of service. In this particular proposition, it will suffice to state that the engineers who have been working on this substitution problem have conservatively figured that with the saving in fuel and the increased cargo carried, the cost of the complete plant will be saved in two years of operation.

32 While the gas plant here described has neither been constructed nor ordered at this writing, its forthcoming will not be long delayed,

TABLE I COMPARISON OF POWER PLANTS FOR GREAT LAKES FREIGHT-CARRIER

Length over all .... 306 ft. 0 in. Beam .... 45 ft. 0 in. Depth .... 24 ft. 0. in.	Displacement .. 6000 tons gross, 6600 tons net. Cargo .... 4200 tons net, 18 ft. draft Speed, 12 statute miles per hr. on 900 i.h.p.
STEAM	**GAS**
ENGINE ROOM	ENGINE ROOM
3-cylinder triple-expansion, condensing, 18–30–50 by 36 in., 1050 i.h.p. at 90 to 95 r.p.m.	4-cylinder, 4-cycle, double-acting, gas engine, 24 in. diam., by 36 in. stroke 1000 b.h.p. at 100 r.p.m.
Auxiliaries steam-driven	Auxiliaries motor-driven
Length between bulkheads, 22 ft. 0 in.	Length between bulkheads, 19 ft. 6 in.
Engine room weights, including auxiliaries and piping, 182,000 lb.	Engine room weights, 105,000 lb.
BOILER ROOM	PRODUCER ROOM
2 single-ended Scotch boilers fitted with economizers, forced draught. Length each boiler, overheads 11 ft. 0 in.	Two down-draft gas producers and auxiliaries
Mean diameter, each, 11 ft. 10 in.	Diameter shell, each generator, 8 ft. 0 in.
Two 42-in. furnaces each	Inside diameter, lining generator, 6 ft. 3 in.
244 2¼-in. tubes, each	
	Height shell, each generator, 9 ft. 6 in.
Grate surface, each, 36.75 sq. ft.	Grate surface, each generator, 30.67 sq. ft.
Heating surface, each, 1642 sq. ft.	
Boiler room weight, water in boilers, no fuel, 170,000 lb.	Producer room weights, no water, no fuel, 82,000 lb.
Length boiler room 24 ft. 0 in.	
Length boiler room, includes bunkers, 30 ft. 0 in.	Length producer room, includes bunkers, 15 ft. 0 in.
	Square feet producer room, 450
	Square feet per h.p., 0.45
Square feet boiler room, including bunkers, 900	Square feet producer room with four smaller generators, 330
Square feet per h.p., 0.9	Square feet per h.p., four generators, 0.33
Bunker capacity, 340,000 lb.	Bunker capacity, 160,000 lb.
Total weight, machinery and fuel, 692,000 lb.	Total weight, machinery and fuel, 347,000 lb.
Total length of machinery space including bunkers, 52 ft. 0 in.	Total length of machinery space, 34 ft. 6 in.
	Saving in weight, 355,000 lb.
	Saving in fore-and-aft length, 17 ft. 6 in.
	Saving in cubic space 17 ft. 6 in. by 32 ft. beam by 20 ft. high, 11,200 cu.ft.


Bunker, 120 Tons
Coaling Hatch 5 x 21'
Ash Ejector
Hold Stringer
Deck House
Skylight 7 x 15
Engine Hatch, Grated
Tank Top at Side
Working Platform
Dynamo Deck
15 K.W. Generator

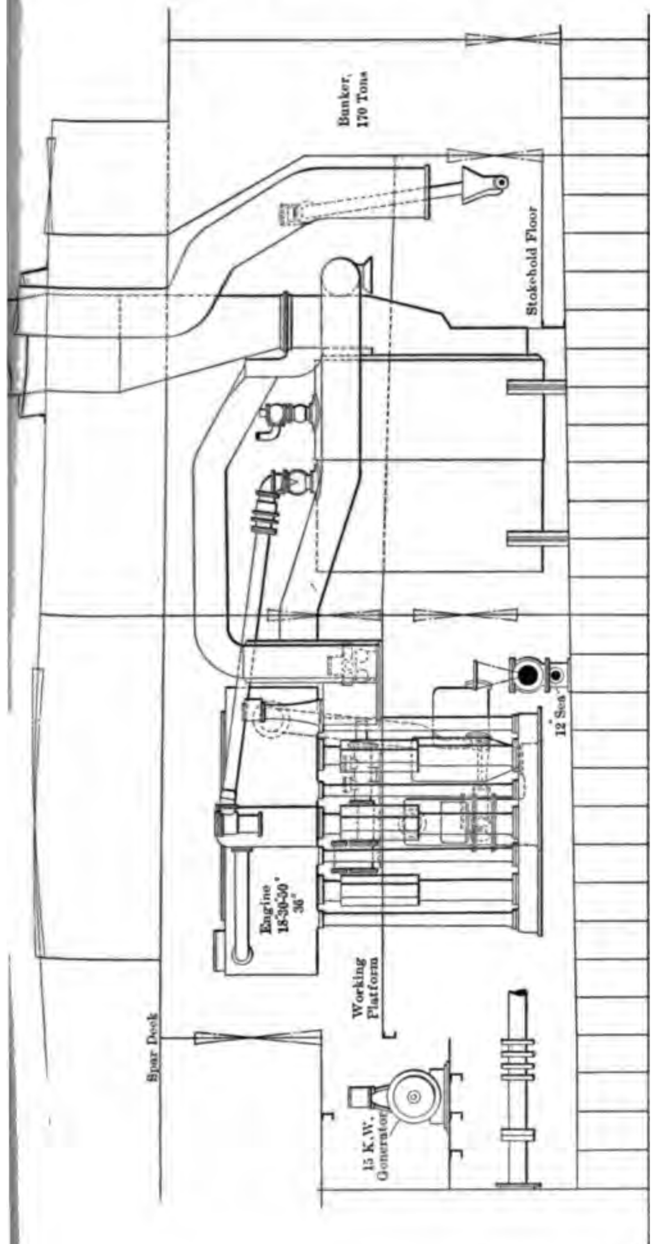
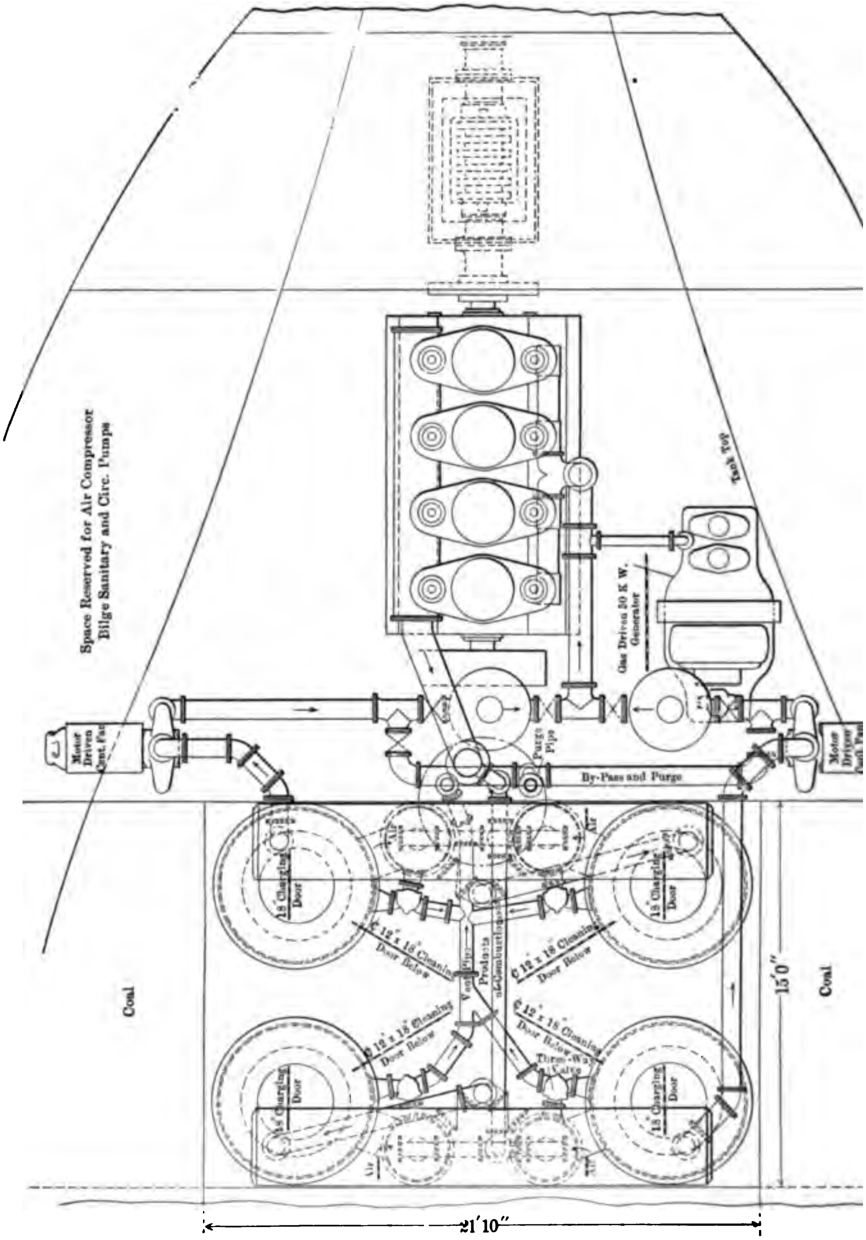


Fig. 1 Plan and Elevation of 1000 h. p. Steam Power Equipment now Installed in Lake Freighter. Triple-Expansion Engines; Two Scotch Boilers


Space Reserved for Air Compressor
Bilge Sanitary and Circ. Pumps
Tank Top
Gas Driven 30 K.W. Generator
Motor Driven Cent. Fan
Purge Pipe
By-Pass and Purge
18" Charging Door
12 x 18 Cleaning Door Below
Products of Combustion
Vent Pipe
Three-Way Valve
Coal
15'0"
21'10"

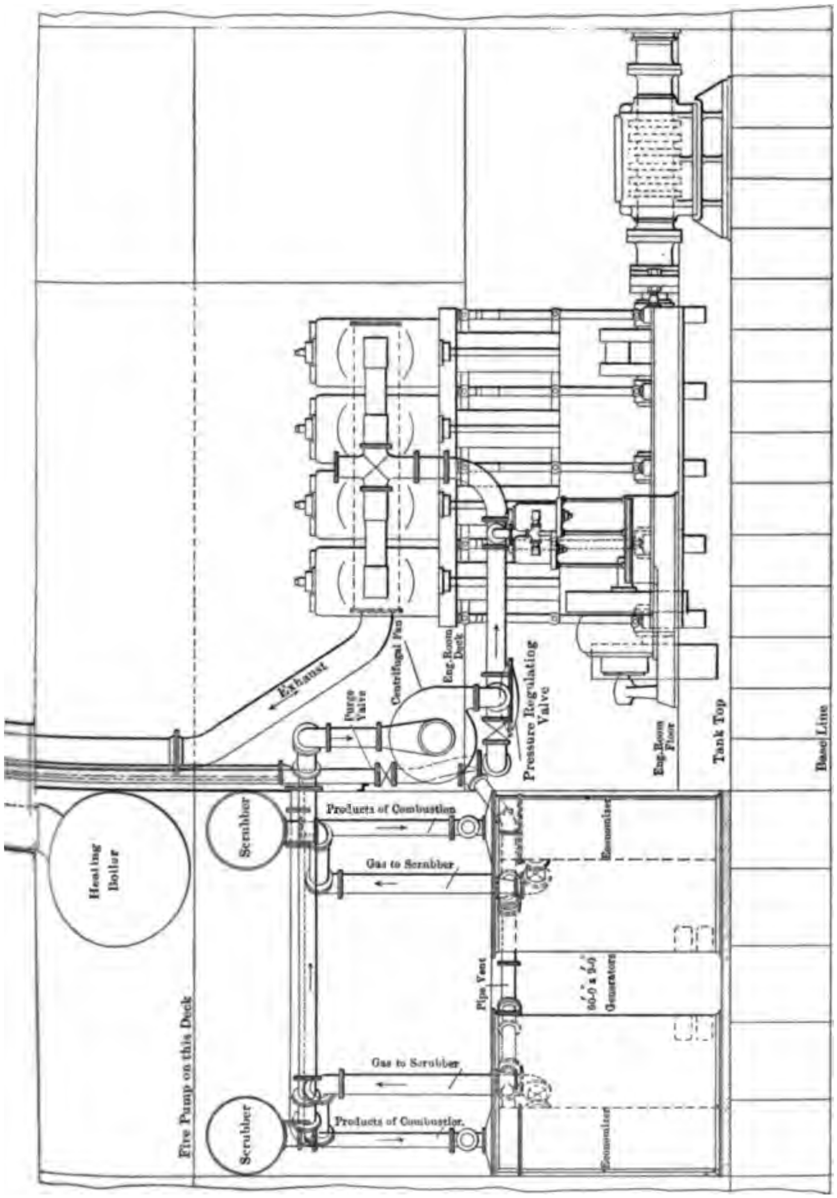
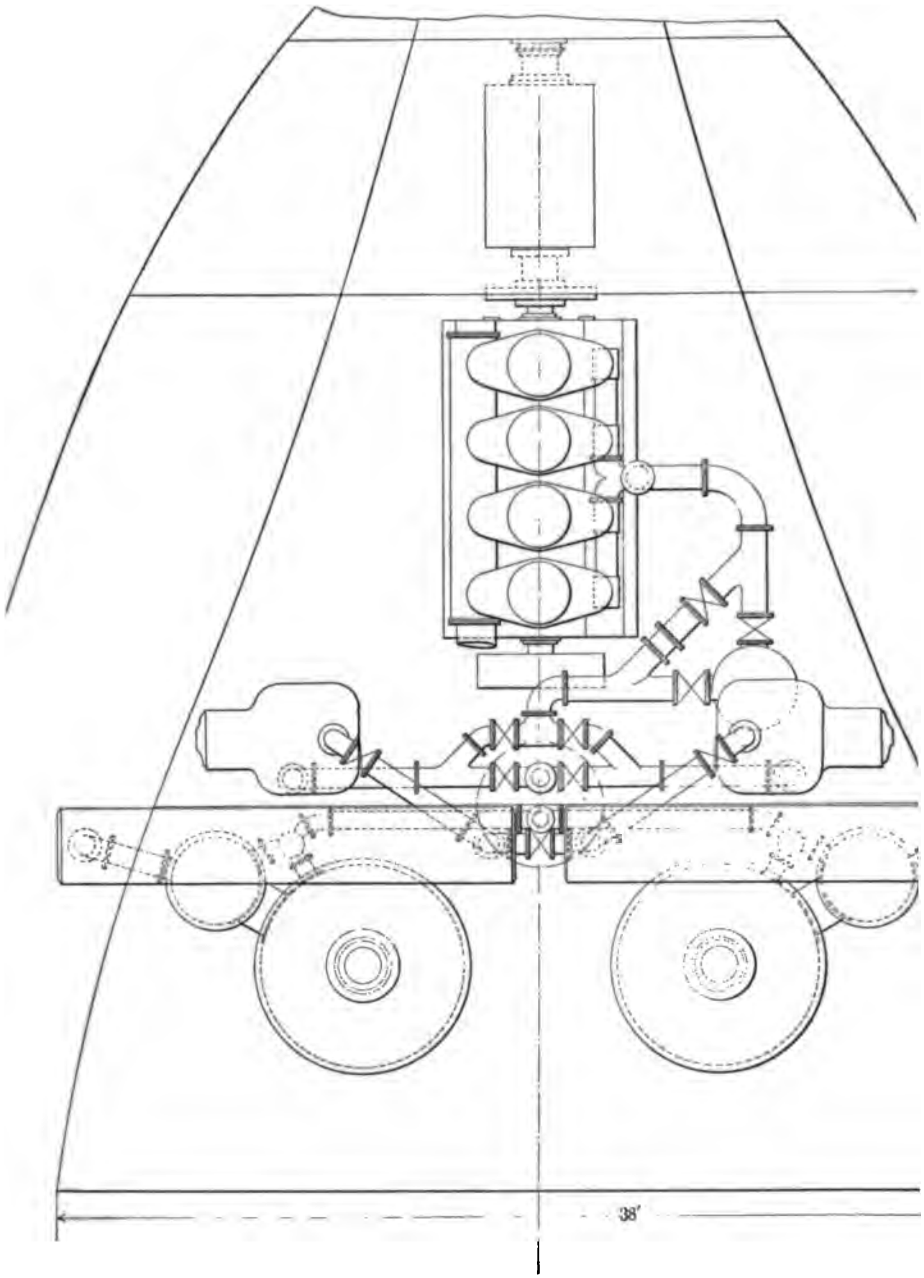


FIG. 2 PLAN AND ELEVATION OF PROPOSED FOUR-GENERATOR MARINE PRODUCER PLANT; 1000 H.P.


38'

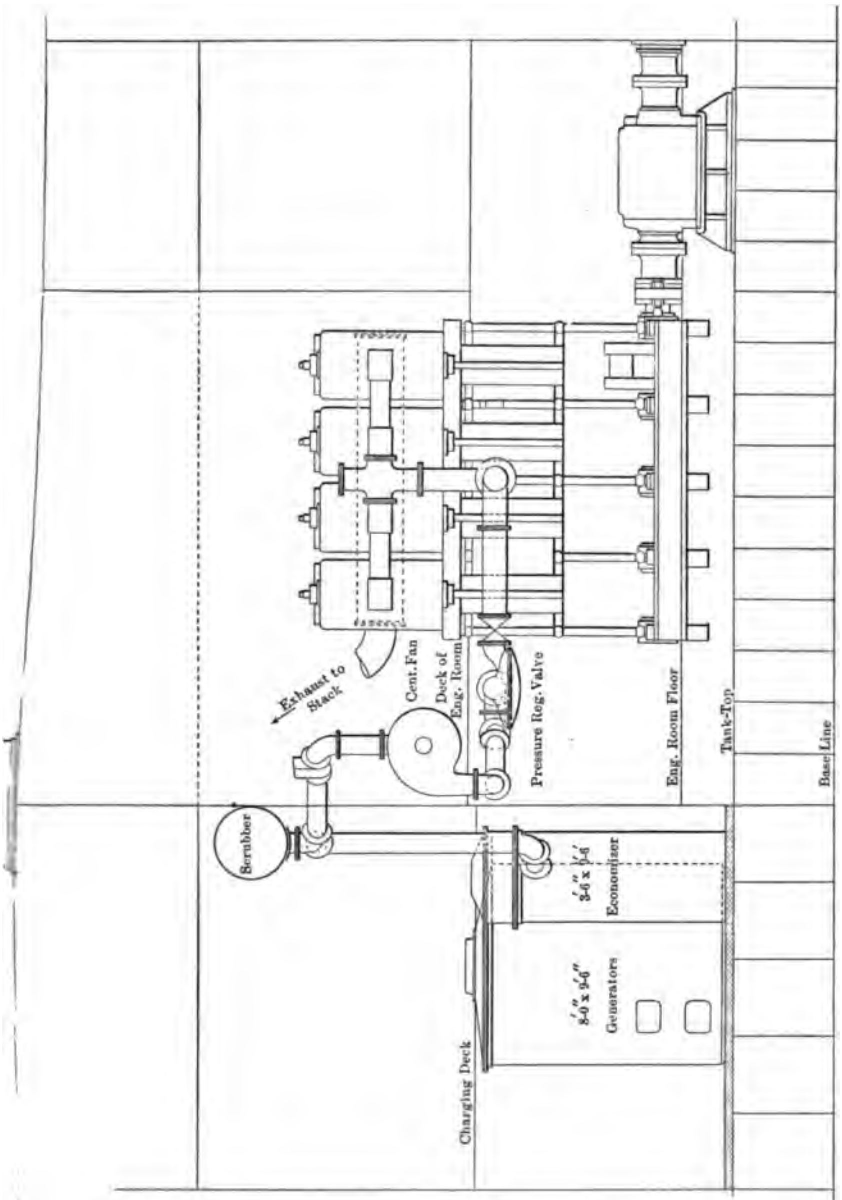


FIG. 3 PLAN AND ELEVATION OF PROPOSED TWO-GENERATOR MARINE PRODUCER PLANT; 1000 H.P.

and this comparison, while somewhat premature, is made to present the possibilities of marine producer gas power to those interested in its future.

33 A marine bituminous gas plant, similar in construction and operation to the one described, but of 300-h.p. capacity, has been in commercial operation driving a six-cylinder, single-acting, reversing marine gas engine for over a year. The results obtained give ample security for the statements made in this paper, and point to the early adoption of this type of prime mover for our marine commercial service.

DISCUSSION

C. L. Straub. We have received reports from abroad of progress in the marine gas-producer field, a summary of which I hope will prove of interest.

2 In Holland teaming is practically unknown and local freight trains are never run, the canals being used for moving freight from one city to another or between different parts of the same city. The canal barges range from 40 ft. in length, with an 8-ft. beam, drawing 3 ft. of water, to 150 ft. in length, with a 20-ft. beam, drawing 6 ft. of water. The majority of the barges are hand-propelled, about 9 per cent have steam equipment, while about 6 per cent are propelled by gas engines. Of the last-named, a few—about $\frac{1}{2}$ of 1 per cent—use gas producers, the others using liquid fuel.

3 The gas-engine barges range in size from 40 ft. to 130 ft., the engines from 10 h.p to 300 h.p., and the engine speed from 250 r.p.m. for the larger to 400 r.p.m. for the smaller engines. The reason for the small number of power boats is the great scarcity of fuel. Holland is without coal mines or any natural growth of timber. Hence coal is expensive and difficult to obtain, though wood is more plentiful as large quantities of lumber are shipped in from Germany. Peat is used a great deal, while compressed peat and some domestic and imported briquettes are burned to some extent.

4 The gas-engine boats using liquid fuel are more handicapped by fuel conditions than the steamboats are. The Standard Oil Co. supplies fuel in a few large cities such as Amsterdam and boats of large power can work around these fuel depots only. This limit to the

Note.—At the time of the presentation of his paper Mr. Straub also presented a report on gas producer development abroad, with special reference to marine work. An abstract of this report is here given as a part of the discussion on his paper.—Editor.

radius of action prevents the extensive equipment of boats with gas engines operating on liquid fuel.

5 The producers installed on the comparatively small number of boats include principally modifications of standard German stationary types. They are not successful and apparently cannot be made to give continued good results with the fuels available. Producers that will operate economically on peat, wood, briquettes or poor coal, and require little attention, are in great demand. The engines are giving satisfaction. Boat owners want gas-engine equipments but not for use with liquid fuel. There is a great demand for gas-producer equipment of capacities up to 300 h.p. for inland waterway traffic, and for capacities up to 600 h.p. for general inland and foreign traffic. Holland thus offers an attractive field, because success with these small capacities will open the way for larger equipments.

6 A test run of two tug-boats was made from Hamburg to Kiel and return, one boat having a steam equipment, and the other a gas-producer equipment. The results are shown in Table 1. The weather was rough and the speed maintained 8½ knots an hour. The coal consumption for a period of 10 hours was as follows: Gastug, 530 lb. anthracite coal; Elfrieda, 1820 lb. bituminous coal; an economy in coal consumption of nearly 3.5 to 1 in favor of the gas-producer equipment.

TABLE 1 COMPARISON OF STEAM AND GAS-PRODUCER EQUIPMENTS

	Gas	Steam
Name	Gastug	Elfrieda
Length	44 ft. 3 in.	47 ft.
Beam	10 ft. 6 in.	12 ft.
Horsepower	4 cyl., 70 b.h.p.	Triple-Exp., 75 b.h.p.
Towing meter pull	2140 lb.	2020 lb.

7 Herr Körting has established a marine department at Kiel, and has practically a monopoly of the government submarine work. He is experimenting and preparing to take up larger work. The German government has decided to adopt gas-engine boats for all warship launches, pinnaces, videttes, portable torpedo boats, and the like.

8 The Nürnberg Company (Vereinigte Maschinenfabrik Augsburg und Maschinenburg Gesellschaft Nürnberg A.G.) have installed a number of small Diesel oil engines for marine use. Both suction and pressure producers are built by them. As to fuel, the company states that anthracite, coke, charcoal and lignite briquettes are the most

suitable for producers. Efforts to gasify raw pit coal have not been entirely successful, at least in medium and small-sized plants.

9 In Great Britain, Vicker Sons & Maxim, Ltd., have built 40,000 h.p. of marine high-speed gas engines. Plans have been drawn for an oil-engine torpedo-boat destroyer of about 30 knots speed. With the same dimensions and speed, the oil-engine destroyer saves enough weight and space so that the armament may be increased from the one 12-lb. and five 6-lb. rapid-fire guns of the steam destroyer to four 33-lb. and two 6-lb. rapid-fire guns. Moreover the quantity of ammunition (number of rounds per gun) is the same for both boats, although the guns and ammunition per round of the oil-engine destroyer are much heavier than those of the steam-engine destroyer. Sufficient fuel capacity is provided to allow a speed of 30 knots an hour with a radius of action 6½ times greater than that of the steam-driven destroyer. Also more space below decks allows superior accommodations for the crew.

10 A marine gas-producer equipment built by the Beardmore Company under the Capitaine patents was tried out with satisfactory results. The capacity of the plant was 70 h.p., the engine having four cylinders 8¼ in. in diameter and of 11-in. stroke. The weight of the whole was slightly over 13,440 lb. The equipment was installed in a launch 60 ft. long and of 10-ft. beam. A 10-mile run was made in one hour without recharging the producer.

George Dinkel. I would like to ask the author if there are any gas-producers working with the small grades of the steam sizes of anthracite coal, such as No. 3, 2 and 1 buckwheat, especially as he states in Par. 8 that there are in commercial operation in this country today two distinct types of stationary power gas-producers suited by their design for operation on almost any class of solid fuel. Where are those producers being used on the same grade of steam sizes, and what has been the result?

Henry Penton.[1] As the members probably know, freight is carried on the Great Lakes at a lower cost than anywhere else in the world, and over 75 per cent of the merchant steam tonnage of the United States is built on the Great Lakes. We are, of course, constantly seeking methods of reducing carrying costs, and so far as the ship is concerned the most important item of expense is that of power. We have for some time been firm believers in gas power: power for

[1] Henry Penton, Babcock & Penton, Cleveland, O.

propulsion, however, is only one of the problems to be met; the problem of the application of gas power to the auxiliary service has given us more concern. At present our ships are handled entirely by steam, both in port and out, and we must have power available at all times.

2 We believed we could depend upon the combustion engineer to perfect the producer-gas engine when the opportunity arrived; but it has arrived and while we believe the producer to be satisfactory we are not yet satisfied with the engine.

3 We first considered the installation of gas power in one of our modern ships where the average horsepower (indicated) is in the neighborhood of 2000, but subsequently decided to select a smaller type, believing that we should practice creeping before walking. While the ship to which the author has alluded represents the best standard practice in design, she is not a representative lake steamer in that she is relatively small. Her engines develop approximately 1000 i.h.p. Her carrying capacity is about 4080 tons on 18 ft., while our modern ships run to 12,000 and 13,000 tons on the same draft. The fuel is necessarily bituminous coal, fairly uniform in heat value and averaging about 13,500 B.t.u. Anthracite and coke are out of the question both as to delivery and cost.

4 It should be noted that the fuel consumption given by the author includes fuel used for all purposes aboard ship charged against the actual indicated horsepower. This is the customary method of stating the consumption and is used merely for purposes of comparison. The propelling engines actually do their work on an average of 1.5 lb. to 1.65 lb. per i.h.p. per hr., including the auxiliaries.

5 In Par. 25 mention is made of the installation of a 50-kw. direct-current generator. I think, however, it will be necessary to use at least two of this size, depending somewhat on the method adopted for handling the windlass, which calls for the largest individual motor of any of the apparatus. There must be no such thing as a generator shutdown. As just stated, every function of the ship, including propulsion, is now performed by steam, and power must be available every instant from the time the ship goes into commission in the spring until she is laid up the following winter. If we take out steam we must provide something just as available in its place. While a great part of the time the output will be small, we must be able to take care of the maximum requirements.

6 It may be wondered why, as stated in Par. 26, the auxiliary boiler is required for heat and for washing down decks. These ships run until well into the winter and when the weather becomes severe

they ice up badly, and it is not uncommon to make port with 200 or 300 tons of ice aboard. The quickest method of clearing away is with the hot-water hose. I hope we shall be able to use the exhaust gases for generating steam at sea and thus operate the boiler without the use of coal; but I do not know whether this is yet feasible.

7 With reference to figuring the elimination of cost in two years, the operation is brought about in this way: some of our ships get in more, some not so many, but the average is not far from 20 round trips per year; and taking into consideration the reduction in weights, which means additional revenue-producing cargo capacity; reduced space, which in some trades is also additional cargo capacity; and reduction in fuel, which is both reduced expense and additional cargo, I have succeeded in convincing myself that we can get even in about 1½ seasons; but two seasons is perfectly satisfactory, and you can readily see that the addition of one or more trips in the year will go a long way toward the extinction of that cost.

Irving E. Moultrop. Examination of Table 1 gives some very interesting information. It is rather surprising that the total weight of the complete gas-power plant is so much less than that of the steam plant. Of course, the steam plant has a number of auxiliaries which the gas plant does not require, and these auxiliaries are quite heavy, yet up to the present time, in stationary practice, the total weight of a gas engine has been very much in excess of that of a steam engine for the same power. The two prime movers discussed in this paper operate at the same speed; one would naturally assume, therefore, that the extra weight in the gas engine would go far toward making up for the saving in weight due to the omitting of a number of steam engine auxiliaries. As the total weight of the gas engine room machinery is only about 60 per cent of that of the steam engine room machinery, one naturally wonders if the factor of safety in the gas engine design has not been reduced to save weight, or if this is not the case, if some special weight-saving features have not been introduced in the gas engine design, which might have been used to equal advantage in the steam engine.

2 Comparing the producer room with the boiler room it is interesting to note that the total grate area of the producers is only about five-sixths of that of the steam boilers. Stationary practice has shown that the best producer results are obtained at a very much lower rate of combustion per square foot of grate than in good steam boiler practice. It would be interesting to know how the engineer of

the gas-power plant expects to obtain capacity out of its producers when the full consumption per square foot of grate area in the producer will exceed what is considered good practice on the grate of steam boilers.

3 In comparing the total weight, machinery and fuel, in the gas-power plant with the steam plant, and also the total space occupied, it should be noted that in the gas plant the bunker capacity is less than half that in the steam plant.

HERBERT M. WILSON.[1] Perhaps it would not be a breach of confidence for Mr. Straub to tell us something concerning the gas producer for the new non-magnetic vessel of the Carnegie Institute. This vessel is being constructed with as little iron as possible, for use in magnetic surveys; and I understand the gas producer and gas engine were selected for auxiliary power because of the small weight of metal required and the possibility of substituting bronze and other non-magnetic metal for iron and steel. Their vessel is about to be launched, and perhaps Mr. Straub could tell something of the gas producer engine plant which is under construction for actual operation.

E. T. ADAMS. Great changes in the weight required have come about in the past few years. The first designs in any line of manufacture uniformly carry unnecessary weight in the various parts and it is safe to say that the gas engine of today, built with the same factor of safety, would be 25 per cent lighter than any engine of the same horsepower built three years ago.

2 This applies to engines in use for ordinary commercial purposes, as electric lighting or power. In view of this fact the statements of the author on this point are not at all surprising. The increasing use of steel and the modification of design based on experience have led to great reduction in weight of engines for special purposes, such as are here specified.

THE AUTHOR. With regard to the question of the speed of vessels depending on the character of power equipment, taking a given hull and a given power equipment, the craft will go at a certain speed, the character of the equipment notwithstanding. The comparative space required by steam and by gas equipments can be best shown by an example given by Capt. A. B. Willits, U. S. N., in an article, Gas vs. Steam for Marine Motive Power, printed in the United States Naval Institute Proceedings, December 1908. Mr. Willits

[1]With U. S. Geological Survey, Washington, D. C.

states that the floor space occupied by the boilers in the New Hampshire equals 0.33 sq. ft. per h.p., and that the weight per b.h.p. is 110 lb. The power of these boilers is rated at their forced capacity. If we install a producer plant and rate it at its forced capacity at 40 lb. of fuel per square foot of grate, it will occupy 1/10 square foot per b.h.p., for such a plant as the New Hampshire would require, and weigh 30 lb. per b.h.p.

2 From these figures it is apparent that a marine producer-gas plant can be installed in at least the same space and of certainly not greater weight than a modern marine steam boiler plant. These figures refute Capt. Willits' figures of the Westinghouse gas plant, which occupied 1 sq. ft. per b.h.p., and weighed 28.5 lb. per h.p., and the R. D. Wood producer which occupied 1.84 sq. ft. per b.h p. and weighed 194 lb.

3 Mr. Dinkel asked regarding the small anthracite coals in gas producers. I can refer him to the generator of the R. D. Wood plant at Jersey City, which has been operating at the plant of the Erie Railroad for five or six years on a mixture of No. 1 and No. 2 buckwheat coal. The Lehigh Coal and Navigation Company has installed a gas producer which has operated on rice coal and has been running for almost two years. We have two plants in operation, one at Hartford, Conn., and one near Philadelphia, running on a fine grade of anthracite coal, mixtures of Nos. 1, 2 and 3 buckwheat.

4 Regarding the factor of safety in gas-engine parts, which Mr. Moultrop brought up, the six-cylinder single-acting engine referred to in Par. 33 weighs less than 30 lb. per b.h.p. Of course, that was a comparatively high-speed engine and delivered 300 b.h.p. at 320 r.p.m. The 1000-h.p. engine will be fitted with cast-steel parts, in almost every instance where cast iron was used on the steam plant, and this makes for a big reduction in weight at a very slightly increased cost.

5 The producer and equipment for the Carnegie, about which Mr. Wilson asked, is almost finished and will be in the boat when she is launched on May 10. The producer shell is about 6 ft. in diameter and is of copper. The pipe and scrubber are of composition metal, containing no iron or steel. The only steel or iron parts on the producer are manganese steel grates, doors and door frames, near the hot portion of the fire in the producer equipment, and this manganese steel is less than one per cent magnetic, when compared to mild steel, so that of it we have been allowed the use of 1500 lb. The engine will have bronze cylinders and will not be lined with cast iron, as the

published reports indicate. We are going to run cast-iron pistons in the bronze cylinders. The cylinders are comparatively so thin and so close to the water jackets, that we anticipate no trouble from deterioration. The only steel or iron parts about the engine will be the cams, rollers and valves. The valves will be of cast iron. The steel cams and rollers will be hardened and ground. On a commercial basis, using terms equivalent to mild steel, as far as magnetic force is concerned, we will have less than 200 lb. total of iron or steel in that vessel. The published reports make further detail unnecessary. The boat will be ready to sail July 1.

No. 1239

OPERATION OF A SMALL PRODUCER GAS-POWER PLANT

By C. W. Obert, New York

Associate Member of the Society

It has been the practice of the packing house of Swift & Company of Chicago, in the distribution of meats and provisions to retailers, to establish in different cities distributing depots with the necessary power equipment for the handling and refrigeration of the products. Some of these branches in the larger cities are establishments of considerable size, and with the extensive cold storage facilities required for the large stocks carried, require comparatively large power installations. The new Westchester market, which the company has recently built in New York at 152d Street and Brook Avenue in the Bronx, is a notable installation of this kind, involving a 400-h.p. producer gas-power plant for the operation of both refrigerating and electric generating machinery, which supplies similar service to a number of adjoining depots of other houses.

2 The refrigerating duty at present required embraces the operation of a total cooling system containing over 46,000 ft. of 2-in. pipe, which reaches a maximum of over 100 tons of refrigeration per 24 hours under the most severe summer weather conditions. Two 65-ton refrigerating machines were installed for this service, with equipments in duplicate, owing to the great importance of continuity of refrigeration, particularly in hot weather. A maximum of nearly 90 h.p. is required for compression machines of this size and engines of 100 h.p. were selected for driving them, to provide sufficient capacity for unfavorable or overload conditions.

3 The electrical load, which includes the operation of several electric elevators, fluctuates ordinarily between 30 kw. and 50 kw. but occasionally reaches a maximum of over 60 kw. For this service, duplicate 75-kw. generators were installed, with driving engines of 100

Presented at the Spring Meeting, Washington, May 1909, of The American Society of Mechanical Engineeys.

h.p. This was done to secure uniformity of size and detail in all four of the driving-engines.

4 For gas making, two producer equipments were installed, also in duplicate. One of these is a 200-h.p. producer, intended for the supply of one refrigerating machine and one generator engine when operating at maximum capacity. The other is of 150-h.p. capacity to permit of closer adjustment of the producer capacity to the load at other times.

5 The plant arrangement consists of an engine room in the easterly end of the sub-basement of the market building, and a producer room adjoining, the entire power equipment occupying a total space, including fuel storage, of 48 ft. by about 55 ft. Headroom for the machinery and piping is afforded by the depression of the sub-basement floor to a level 16 ft. below the street, and the omission of the basement floor in this section, giving thus a clear headroom of 18 ft. The machinery space was originally laid out as a single room, but as a result of the requirements of the underwriters, the producer space has been separated from the rest by a 6-in. hollow-tile fire wall, forming a producer room 20½ ft. by 24 ft. maximum dimensions. Under the 152d Street sidewalk, there is an 11 ft. by 29 ft. room containing pumps and auxiliaries for the power equipment and the building heater; and adjoining this, an 11 ft. by 30 ft. space for fuel storage. The latter has capacity for over 150 tons of coal, which is dumped into it through sidewalk coalholes from wagons in the street.

6 The engines are Rathbun vertical, three-cylinder units, of 100 h.p., rated at 280 r.p.m., built by the Rathbun-Jones Engineering Company, Toledo, Ohio. The two for the electrical service are direct-connected to 75-kw. generators and the other two through silent chain drives to the ammonia compressors of the refrigerating equipment. They are all of the four-stroke cycle, single-acting, enclosed type, and have 12¾ in. by 13-in. cylinders, designed for the above rating when operating on producer gas of not less than 125 B.t.u. per cu. ft. These engines are throttle-governed, a special form of centrifugal flyball governor being used, and have each a one-ton fly-wheel at both ends of the crankshaft.

7 The gas is generated for the engines in a duplicate equipment of Smith suction producers built by the Smith Gas Power Company, Lexington, Ohio. Each equipment consists of a simple shell producer, a wet scrubber and a dry purifier. While the producers differ in rated capacity to permit of more accurate adjustment of their capacity to the power requirements at different seasons of the year,

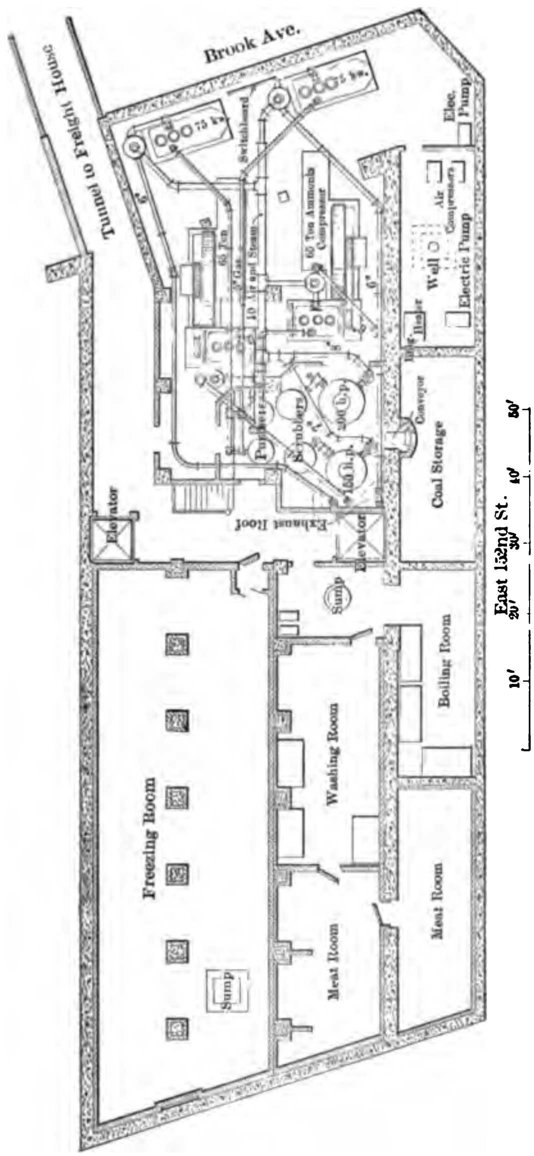


Fig. 1 Plan Showing Location of Machinery, Apparatus and Connections

the scrubbers and purifiers have a maximum capacity of 200 h.p., which permits the smaller producer to operate up to the maximum plant capacity of 200 h.p., if required to do so temporarily. The small and large producers have 6-ft. and 7-ft. shells respectively, both 12 ft. in height, their internal diameters being 4½ ft. and 5½ ft. respectively, and they are fitted with shaking grates on the up-draft principle for operation with anthracite coal. They are not fitted with attached vaporizers or air pre-heaters, but have an automatic control attachment for regulation of the amount of water vapor to conform to the power requirement and consequent rate of gasification. The scrubbers for cleansing the gas are vertical cylindrical tanks, each 4 ft. in diameter by 15 ft. high, and the dry purifiers have 4-ft. shells 6 ft. in height.

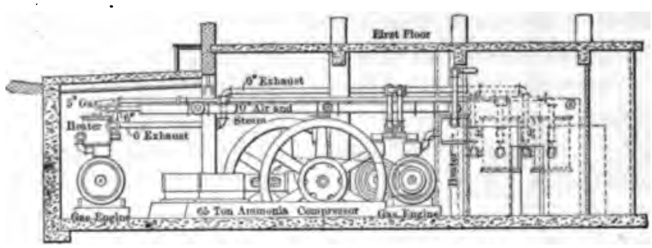


FIG. 2 ELEVATION OF MACHINERY ROOM IN CROSS-SECTION

8 The piping of the plant was somewhat involved by the arrangement of the engines relative to the producers, and, in the Smith producer system, by automatic vaporizers in the exhaust connections to utilize the waste heat of the engines for the vaporization of the water. The vaporizers are located close to the engines and attached to each vaporizer is an automatic device, through which air is admitted and superheated for the producer. The air is conducted to the producers from these devices by a 10-in. pipe main, extending through the engine room, and heavily covered with magnesia insulation.

9 The gas is delivered from the producers by 8-in. pipes connecting from the top of the producer to the bottom of the scrubber shell and each scrubber has a triplicate connection to its corresponding purifier, which is a three-part filter. From these the gas is conducted to the engines through a 5-in. line, with a 3½-in. branch to each. The exhaust connections from the engines to the vaporizers are 5-in. lines and from the latter. individual discharge pipes are carried up for each engine through a pipe shaft in the corner of the building to a roof outlet.

It is to be noted that this arrangement of exhaust connections is effective in so muffling the noise of the escaping gases that they cannot be heard from the adjoining street and are only barely noticeable when on the roof close to the outlets.

10 The electrical generators are 75-kw. General Electric direct-current machines, each rigidly coupled to the driving engine. They are wound to deliver current at 220 volts, the distribution for both lighting and power being on the two-wire system. The electrical circuits are controlled on a three-panel switchboard which contains the usual equipment of indicating and recording instruments, field-rheostat switches and generator and feeder switches. The building is wired separately for lighting and power circuits, and recording watt meters are connected into the feeder circuits for measurement of the power delivered. It is to be noted that separate bus bars are provided for both power and lighting feeders, as well as a switching arrangement by which the lighting service may be supplied from a generator other than that carrying the power load, in case the fluctuations of the latter should interfere with the voltage regulation. This provision has been found unnecessary, however, as the speed regulation of the engines and generators is satisfactory under all fluctuations of loading due to elevator operation.

11 The refrigerating equipment was installed on the direct ammonia expansion system, a feature of which is the connection of all coils in the coolers in series with those in the freezers, whereby all ammonia not thoroughly evaporated in the freezer coils will be in the cooler coils (temperature, 36 deg. fahr.), which permits carrying the freezer temperature at from 0 deg. to + 5 deg. without frosting the compressor. The compressors were built by the Hutteman & Cramer Company, Detroit, Mich., and are horizontal single-cylinder double-acting machines, with 14-in. by 30-in. cylinders, each driven at a speed of 60 r.p.m. by a Renold silent-chain connection from its driving engine, with a speed reduction of about five to one.

12 The ammonia condenser is located on the roof of the building and provided with the usual water-cooling sprays. The water supply for it is obtained from a well extending into water-bearing soil under the basement floor, and the drainage from the sprays is subsequently utilized in the scrubbers and in the engine cylinder jackets. One of the compressor units normally handles the load alone, which leaves one equipment always in reserve, to provide against the serious emergency of a complete stoppage of the refrigerating service during hot weather.

13 In operation this plant has proved particularly economical, largely due to the continuous character of the service resulting from the operation of the refrigeration plant 24 hours a day, seven days a week, thereby eliminating standby losses. The average load range of the plant is ordinarily from 50 per cent (100 h.p.) to full rated load (200 h.p.), the high and low load factors occurring during the summer and winter months respectively, when the refrigeration requirements are maximum and minimum. With the heavier load factor during the summer months, the fuel consumption has ranged between 3400 and 4800 lb. per 24 hours, the larger figure having been exceeded on only two days in 11 months, and the consumption per horsepower-hour as calculated from station fuel records and observed loads, ranged from 1.4 to 2.0 lb. of coal. The fuel rate has dropped during periods of continuous high loads, to about 1 lb. per horsepower-hour, as based on observed loadings, but the daily average under conditions of ordinary commercial operation is usually greater.

14 The operating conditions during the heavy-load season are indicated in the table at the end of the paper, in which the relation of fuel consumption to load carried is shown for two weeks of similar duty. The variations in the amount of fuel charged from day to day are due chiefly to the differing conditions of the fuel bed in the producer, the removal of a particularly large amount of ashes on any day necessitating a heavy fuel charge. No account is taken of cost of water used in the scrubbers and cooling jackets, as the supply is obtained from a well on the premises without cost other than that of pumping.

15 The fuel used is No. 1 buckwheat anthracite that has been passed over a $\frac{3}{8}$-in. mesh and through a $\frac{9}{16}$-in. mesh screen, with 5 per cent fineness, and costs $3.50 per gross ton delivered in cargo lots. It is charged only at the regular cleaning periods, at each of which from 400 to 900 lb. of coal are fed, after the fire has been cleaned down and the ashes removed from the grate. The fire is cleaned periodically twice every shift, or four times per 24 hr., and requires about an hour per cleaning on the average.

16 In this connection it is interesting to note the comparatively short time required to start a producer into service from the cold, which has been done repeatedly on short notice in about five hours; on December 12 when the 150-h.p. producer was placed in operation to relieve the larger unit, the kindling wood was lighted at 10 a.m. and the gas supply turned onto the engine at 2 p.m., with only about 12-in. of fire zone in the fuel bed. The reliability of a suction producer

operating under a continuous and exacting service of this character is well shown by the duty of the 200-h.p. producer during the summer season of 1908, which when taken out of service on December 12, had been continuously in service 24 hours per day and seven days per week since April 22, a continuous run of 235 days. During that time, it had received no more attention than the four cleanings and chargings per 24 hours.

17 The operating force for the power plant consists of an engineer and an assistant engineer and two producer tenders, who work in two shifts. This force is able to maintain the plant equipment in satisfactory operating condition, as well as the refrigerating and electrical equipment of the depot, and it is worthy of note that the plant has not been shut down for any reason since it was started on February 1, 1908, a period of 15 months. Experience during this period indicates that, contrary to the general opinion, no more attention is required than for a first-class steam plant, the necessary attendance comparing very favorably with that of a high-grade steam plant of the same capacity. Cleanliness of all parts of both producer and engine equipments, and careful adjustments, especially of the latter, are imperative and are the keynotes of successful operation. In order to maintain the equipment in such condition, a thorough and comprehensive operating system has been developed which may be of interest.

18 The operating system involves a detailed and thorough inspection routine that keeps the force well informed as to the condition of the entire equipment, and a division of duties tending to favor the maintenance work. To the day operating force is assigned the inspection and adjustments of the engines and repairs to igniters, batteries, etc., while the night force has the work of cleaning all machinery.

19 The regular routine of the day force is in detail as follows: First upon coming on duty at 7 a.m., an examination is made of all moving parts of the two engines in operation, and also of oil levels in lubricators and conditions of water jackets and ignition systems. There are always two engines in operation, one being a generator engine and the other a refrigerating engine, which in the periods of heavier loadings in summer time have a combined load of about 140 h.p. of which fully 75 h.p. is taken by the refrigerating system. Next the water regulation for the steam supply is noted and then the condition of the suction draft on the producer and also on the scrubber and purifier, there being three U-shaped draft gages provided for this purpose, one connected to the gas suction line to the engines, the

second to the gas connection from the scrubber to the purifier and the third to the connection between the producer and scrubber. A uniformity of suction of from 2 in. to 3 in. of water in these three gages indicates a proper condition of the three units, while any unusual suction in any of the connections would indicate an obstruction needing immediate attention. The latter is always clearly indicated, as an obstructed condition in the producer, for instance, will raise the suction to as high as 9 in. or 10 in. of water.

20 Next an inspection is made of the producer, the temperatures of different portions of the fire being determined to ascertain the condition of the fuel bed, the existence of cracks or fissures or pockets of unburned coal. To do this, a $\frac{5}{16}$-in. iron rod is pushed into the fire through the side peep holes in the producer shell, held there exactly one minute and then withdrawn, the temperature within being noted from the color of the rod. If the latter is at a uniform cherry red temperature throughout its length, this is taken as an indication of an even fire; but if at a brighter heat or dull in some portions of the rod, there is evidence of unnecessarily high local temperatures due to rapid combustion in fissures in the fuel bed, or of a stagnant condition in dirty or unburnt portions of the fire. The rod is first inserted in the lowest hole and then successively into the upper holes, in order to explore the fire in zones. On withdrawing the rod the operator notes graphically the condition of the fire by marking a line with chalk on the shell of the producer even with the hole, a straight line indicating an even temperature, and a broken line showing the dirty condition, etc. This operation is continued for the four holes and a fuel curve drawn from it which gives a practical idea of how the dirt lies in the producer and shows what quality of gas can be expected. Finding the producer in good order, the scrubber, purifier and connections are examined for unusual temperatures, condition of water flow, etc.

21 In the maintenance work, each engine is shut down after every seven days work of 160 hours for general inspection and cleaning, and thus on Monday mornings it is necessary to start up the two reserve units and transfer the respective loads to them. Before starting up either reserve engine, its igniters are cleaned, which takes about one hour. With the igniters clear and everything in good order, the attendant looks at the draft gage, which is equal in importance to the gage of a steam boiler, to see what gas the engines in operation are drawing and whether the start can be made without interfering with their suction. If there are any doubts the gas is

enriched temporarily by putting about four pails of water in the ash pit of the producer and slicing the fire to work down some hot coals, which, by turning the water into vapor, increase the hydrogen content of the gas and enable the third engine to be started without interfering with the others. After getting the engine warmed up, the load is thrown on and the other engine is shut down. The extra pull on the producer, due to overload from running the three engines and the hydrogen added, has usually so enriched the gas that on cutting out a unit the quality of gas is too rich for the two units operating alone. To counteract this, it is necessary to give additional air to each of the units that remain and then, as in starting, there will be no variation in speed of operation.

22 After the engines have been shut down their inspection is begun by the removal of the back crank case covers and examination of the bearings, crank pins, wrist pins, etc., for necessary adjustments. Besides this the exhaust valves are cleaned and the ignition system checked. This requires about two days, as but one thing is done at a time and then only at times when the load on the plant is not heavy. While the engineer is performing this work, the producer tender prepares to clean and coal the producer, as follows:

23 The method of cleaning is to rake off the ash from the grate table and then poke down around the shell from the top poke holes. Having before him the fuel chart which was noted graphically on the producer shell on coming on watch, the attendant knows what part of the bed requires most poking. Before opening the ash pit doors about the shell, water is placed in the ash pit as before and the hot ashes, dropping down, form sufficient steam to mix with the air coming through the ash pit door and offset any bad effect therefrom. This enables the cleaning to be done without affecting the engines. Having cleaned and poked the fire thoroughly and worked down all the ash so as to leave it as clean as possible, the coaling is then begun, count being taken of each hopper of coal charged. The coal is cleaned by screening if very fine or dirty. Having coaled, the operator slices across the grate so as to relieve the center of the fire and again puts water in the ash pit, this time to cool off the grate after cleaning and to offset the effect of any air that may have gotten in during the operation. The cleaning usually occupies one hour, the amount of coal put in ranging up to 900 lb. After giving the producer time to settle down the ashes are withdrawn from the ash pit, an average of 1½ ash cans (about 3 bushels) being removed after each cleaning. During the cleaning operation the operator is always on the lookout for any

change in the engine speed due to weak gas on account of opening the ash doors. Should this occur he immediately cuts the air supply to the engine, resulting in a combustible mixture without noticeably reducing the speed. The producer is now good for 6 hours' operation, after which the cleaning is repeated.

24 The refrigerating engines are operated for periods of 84 hours and then gone over. One exhaust valve is taken out of an engine each week, thoroughly cleaned, and reground if necessary, thus insuring attention to each valve once in every three months. Igniters are cleaned weekly and the batteries and ignition system checked. The temperature of the fuel bed of the producer is taken twice a day and a gas analysis is made once a week or oftener if necessary. The average calorific value per cubic foot of gas is 134 B.t.u., based on analysis: CO_2, 8.6 per cent; O, 0.6 per cent; CO, 20.2 per cent; H, 18.5 per cent and N, 52.1 per cent.

TABLE 1 RECORD OF LOAD AND FUEL FOR TWO HEAVY WEEKS

	ELECTRICAL LOAD		Refrigerating Load	Total Load	Coal Charged Pounds
	Kw. hours[1]	B.h.p. hours[2]	B.h.p. hours	B.h.p. hours	
Sunday, July 25, 1908	332	556	2010	2566	3600
Monday, July 26	400	600	2030	2630	3900
Tuesday, July 27	404	606	2020	2626	3540
Wednesday, July 28	390	585	2020	2605	3180
Thursday, July 29	410	615	2030	2645	4020
Friday, July 30	415	622	2040	2662	4320
Saturday, July 31	403	605	2020	2625	4080
Sunday, August 22, 1908	328	549	2030	2579	3660
Monday, August 23	386	579	2020	2599	3420
Tuesday, August 24	393	589	2020	2609	3600
Wednesday, August 25	392	588	2010	2598	3540
Thursday, August 26	397	596	2010	2606	3840
Friday, August 27	391	586	2020	2606	3540
Saturday, August 28	393	590	2020	2610	3720
Totals	5434	8266	28300	36566	51960

[1] Recorded by watt-hour meters.

[2] Deduced from kilowatt-hours by assuming 80 per cent efficiency for the generator during light-load periods and 90 per cent for the remaining time.

DISCUSSION

J. A. Holmes. The success of the small producer plant using anthracite coal is practically assured. Not long since (1905), in visiting a number of these plants in Cologne, Germany, I found a newspaper press that had been operated entirely for more than a year by a small gas-producer plant burning small-sized anthracite coal; one of the larger hotels there had been using such a plant for a longer period with entire satisfaction to supply all its electric light and power; in a large commercial house, electric lamps, elevators and all other machinery connected with the establishment were operated by one of these plants. In each of these cases the producer, engine-driven generators and other equipment in the power room, were all operated by one man, and the plant was regarded as a success in efficiency and economy of labor and fuel. In the United States, also, many producer plants have been using anthracite coal for some years. In our own investigations at the Government testing station, anthracite coal has been regarded as a fuel so simple and so easily regulated that we have done little work on it, turning our attention mainly to the bituminous coal producer problems.

2 In regard to producer work with bituminous coal, we have investigated fuels rather than different types of producers. Using every imaginable grade of bituminous coal and lignite in making short-time tests, we have encountered many difficulties due to a lack of familiarity with the special manipulations required by certain fuels. These difficulties, one being to secure a uniform quality of producer gas, would not be met in using the same fuel year after year. In early work, with the Taylor producer, we could get gas of absolutely uniform character not more than an hour at a time, and the variation in 24 hours was at times from 125 B.t.u. to more than 200 B.t.u. per cu. ft. of gas, these variations being largely due, no doubt, to inexperience in the handling of any special fuel. During the past three years, however, with more experience, the regularity and efficiency of this gas have been greatly increased.

3 Another difficulty, and one not entirely separable as yet, is the slagging or clinkering of the ash in the producer. The ash in certain coals slags more readily than in others; and different ashes slag more readily at different temperatures. One of the greatest needs in producer development at the present time is that of a regular mechanical feed of coal and removal of the ashes which now accumulate in some producers, to be cleaned out after the producer has cooled down. We

have sometimes found the slag from certain coals, burned at high temperatures, accumulating irregularly on the brick walls lining the producer, at the rate of 6 in. to 10 in. during a week's run. If mechanical arrangements can be devised, by which the ash may be removed from the base of the producer as regularly as from the base of a boiler, then the use of a double producer will be largely avoided. Decided progress is being made in overcoming this difficulty.

4 Still another line of progress is in the reduction of weight and bulk of the producer making its use possible instead of that of steam boilers for propelling ships. Mr. Straub's paper indicates what is being accomplished along this line. Already the anthracite producer and gas engine have been reduced in size and weight to less than those of the steam boiler and reciprocating engine; and the outlook is hopeful for the producer burning bituminous coal.

John H. Norris. I have been connected with the manufacture of gas engines for a number of years, and the principal trouble we have had in the operation of gas engines of any size is to overcome the notion that a gas engine needs no care. At the present time, however, gas engines are running successfully because in most installations they receive proper attention. I am glad to see put on record the statement that a gas engine installation needs as close attention as a steam engine installation.

William A. Bole. The Westinghouse Machine Company has been working on the gas-producer problem as well as on the gas-engine problem for some time, and now believes itself ready to offer gas producers that will be as practical and as easily manipulated and capable of as long-continued runs as any boiler plant. A producer plant of 175-b.h.p. capacity has been in operation at our works for practically a year, without pulling down the fires. During that period all sorts of runs have been made, continuous runs at full capacity for ten days or two weeks, and the more ordinary runs in which producer and engine are shut down at night; and the producers have burned not only the comparatively good coals of the Pittsburg district and the better coals of the Pocahontas region, but several of the Western and Southwestern lignites and even peats from New England. The latter have not been so successfully burned, but the success in burning Colorado lignites has been very decided. This producer was shut down and cleaned out, simply by shoveling the ashes out of the water seal, and observations of the condition of the interior walls showed that it might just as well have been operated continuously for five years instead of one, or as many years as the firebrick lining would last. The requirements for continuous performance seem to have been admirably met in this design.

2 This producer is designed for the burning of bituminous coal alone, and resembles a small producer inverted and placed on top of a large one, making a double-zone producer especially adapted for the gasification of bituminous coal without passing tar of any description out of the producer shaft. Apparently the only solid material emitted from the gas is a small amount of lamp-black which is successfully removed by the use of a static or stationary scrubber, and the cleanliness of the gas is proved by the fact that practically all the gas was converted into brake horsepower by being employed in the actual operation of a gas engine, without troublesome deposit of any kind upon the ports or other parts of the gas engine.

3 Whether such a producer would be available for marine purposes I do not know; the only question seems to be whether the motion of the ship would interfere seriously with the descent of the fuel from top to bottom. The producer has been subjected to practically every test, and we believe we are about ready to offer it for both large and small plants.

The author desired to present no closure.—Editor.

No. 1240

OFFSETTING CYLINDERS IN SINGLE-ACTING ENGINES

By Prof. Thurston M. Phetteplace, Providence, R. I.
Member of the Society

A great deal has been said recently about the offsetting of cylinders in single-acting engines and many claims of superiority are made by those who employ this form of construction.

2 About twenty-five manufacturing establishments in the United States are building engines in which the cylinders are offset, chiefly those of the automobile type, and one company is formed for the purpose of making engines in which the offset is equal to the crank radius and the connecting rod length is about $3\frac{3}{8}$ times the crank radius.

3 Among the claims made by manufacturers for offset engines are: greater power, less side-pressure of the piston on the walls of the cylinder, better turning effort, less vibration, smoother running qualities, and when one cam shaft is used, a more convenient mechanical arrangement.

4 On account of the importance of this subject and the lack of information concerning it, a complete discussion is desirable and is here presented.

5 The cylinder of an engine is said to be offset when its center-line is not in a plane through the center of the crank shaft. The practice is not new and is applied to both steam and gas engines having one or any number of cylinders.

6 In the diagram, Fig. 1, AB represents the stroke, OE the crank radius, DE the connecting rod, θ the crank angle, and OC the offset. It should be noticed that θ is the angle the crank makes with a line through the center of the crank shaft parallel to the center-line of the

The full development of the mathematical formulae of this paper, with some other related matter, is given in an unpublished Appendix, which is on file in the Library of the Society.

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cylinder, and not the actual angle passed over from the inner dead point. The length of the stroke is

$$AB = R\left(\sqrt{(a+1)^2 - k^2} - \sqrt{(a-1)^2 - k^2}\right)$$

which is greater than $2R$.

R = crank radius.
a = L/R.
L = connecting rod length.
k = offset divided by R.

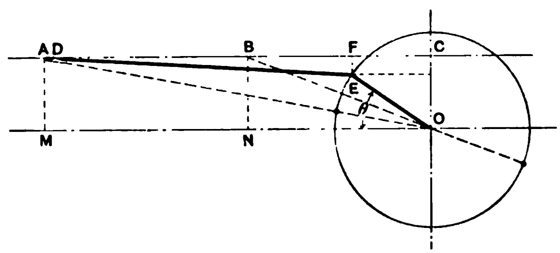


Fig. 1 Diagram of Crank and Connecting-Rod Train

Thus for 3-in. crank radius the strokes would be as shown in Table 1. The distance to the end of the stroke farther away from the center of the crank shaft is shortened, thus OM is less than $DE + EO$ or $L + R$ which affects the height of the engine.

TABLE 1 LENGTHS OF STROKES FOR DIFFERENT OFFSETS, 3-IN. CRANK ENGINE

Ratio L/R	Offset R	Stroke	Increase Per cent
Any	zero	6.00000	0.00
3	0.10	6.00375	0.06
3	1.00	6.42279	7.04
4	1.00	6.21180	3.50
5	1.00	6.12930	2.15
6	1.00	6.08730	1.01

7 The dead points are not opposite each other, so that the crank angle swept over while the piston makes the out-stroke is less than that for the in-stroke, causing a quick return motion and an average velocity for the in-stroke or compression and exhaust strokes greater than for the out-stroke, or explosion and suction strokes.

8 An expression for the piston position in terms of the crank angle θ is developed in the usual way and is

$$X/R = \sqrt{(a+1)^2 - k^2} - \cos\theta - \sqrt{a^2 - (k - \sin\theta)^2}$$

in which X = the piston displacement from the end of the stroke farther from the crank shaft.

9 The force of inertia due to the reciprocating parts is equal to the weight multiplied by the acceleration divided by 32.2. The value for the acceleration is found by differentiating the expression for the piston displacement twice with respect to the time. This is done by expanding the radical $\sqrt{a^2 - (k - \sin\theta)^2}$ by the binomial theorem, into a convergent series and then dropping all terms containing a with a negative exponent of 3 or larger in order to get an expression that can be easily differentiated. This gives

$$[a^2 - (k - \sin)^2]^{\frac{1}{2}} = a - \tfrac{1}{2}a^{-1}k^2 + a^{-1}k\sin\theta - \tfrac{1}{2}a^{-1}\sin^2\theta$$

This approximate expression for the radical differs from the radical for different values of k, a and θ, as shown in Table 2.

TABLE 2 DIFFERENCE BETWEEN EXACT AND APPROXIMATE EXPRESSIONS

k	a	θ	$\sqrt{a^2-(k-\sin\theta)^2}$	$a-\frac{1}{2}a^{-1}k^2+a^{-1}k\sin\theta-\frac{1}{2}a^{-1}\sin^2\theta$	Difference
1	6	90	6.000000	6.000000	zero
1	6	0	5.916079	5.916666	+.000587
1	6	45	5.992762	5.992867	+.000105
.5	6	90	5.979130	5.979166	+.000036
.5	6	0	5.979130	5.979166	+.000036
.5	6	45	5.996428	5.996429	+.000001
.5	3	90	2.958039	2.958333	+.000294
.5	3	0	2.958039	2.958333	+.000294
.5	3	45	2.992849	2.992859	+.000010
.5	$4\frac{1}{2}$	90	4.472136	4.472222	+.000086
.5	$4\frac{1}{2}$	0	4.472136	4.472222	+.000086
.5	$4\frac{1}{2}$	45	4.495236	4.495239	+.000003

10 The greatest difference has no significant figure until the fourth decimal place is reached and this is when $k = 1$, which is an unusual value. Hence it is readily seen that the error introduced by this approximate form is slight.

11 Substituting this value of the radical in the expression for the piston displacement and differentiating twice with respect to the time gives

$$F/A = 0.00034\ W/A\ N^2R\ (a^{-1}k\sin\theta + \cos\theta + a^{-1}\cos 2\theta)$$

which is the expression for the inertia force per square inch of piston head area when there is an offset.

A = area of piston head.
W = weight of the reciprocating parts.
N = revolutions per minute.
R = crank radius in feet.

This differs from the similar expression when there is no offset by the addition of the term $a^{-1}k \sin \theta$, so that tables for inertia factors for no offset may be used by adding the value of this term.

12 The expression for the tangential pressure or the turning force for any offset is

$$T = P\left(\sin\theta + \cos\theta\,\frac{\sin\theta - k}{a - \frac{1}{2}a^{-1}k^2 + a^{-1}k\sin\theta - \frac{1}{2}a^{-1}\sin^2\theta}\right)$$

in which P is the pressure on the piston pin in the direction of the center of the cylinder. This is a long expression to solve and a graphical solution may be followed if preferred. The work of solving the expressions for inertia force and tangential pressure may be somewhat lessened by tabulating the quantity $a^{-1}k \sin \theta$ which appears in these expressions.

13 The derivation of the preceding formulae and tables is shown in the appendix.

SIDE PRESSURE OF PISTON ON CYLINDER WALLS

14 A reduction of the side pressure of the piston on the cylinder walls is one of the advantages claimed for offsetting.

15 There are two ways in which the side pressure may affect the single-acting engine: (*a*) The maximum value of the side pressure determines the length of piston to keep the maximum pressure per square inch of projected area below a value which is assumed as not too great to destroy the oil film between the rubbing surfaces; (*b*) The average value of the side pressure produces the friction between the sliding surfaces causing a mechanical loss and some wear of the parts. The loss in mechanical efficiency is more important than the wear, especially in the small high-speed automobile engines.

16 The average side pressures may be found by adding all of the areas between the axis and the curve of side pressures and dividing by the total length, or the areas themselves may be taken for comparison, as they represent the work done on the side of the cylinder

by the piston, which is lost work and should be kept as low as possible. Curves of side pressures of the piston on the cylinder walls were constructed, it being necessary (*a*) to assume a gas card, (*b*) to assume engine dimensions, (*c*) to calculate inertia forces and plot curves,

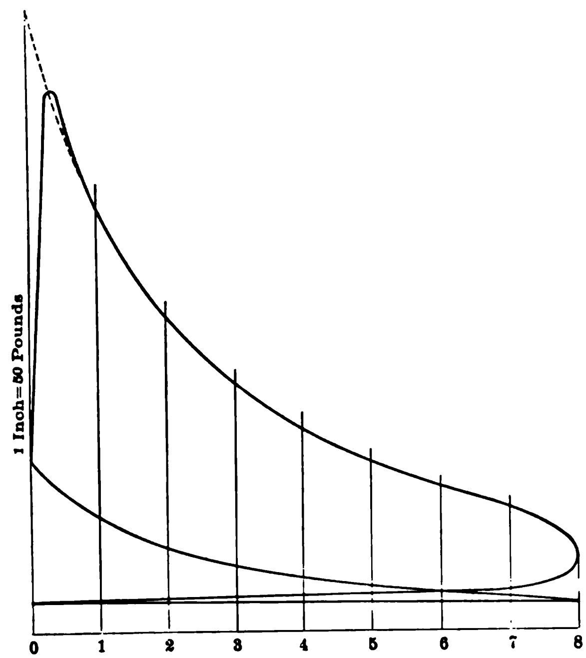


FIG. 2 OFFSETTING CYLINDERS IN SINGLE-ACTING ENGINES
Gasolene Card Compression, 70 lb.; Maximum Pressure, 259 lb.; Pressure ratio, 3.77

(*d*) to combine inertia forces with gas pressures, obtaining the force at the piston pin, and then (*e*) to determine the side pressure component perpendicular to the center-line of the cylinder for the changing angularity of the connecting rod. The gasolene card chosen is shown in Fig. 2. As it seemed desirable to investigate two similar cases, one

TABLE 3 CASES INVESTIGATED

Specifications	Slow	High
R.p.m.	450	1500
W/A	1 lb.	0.70 lb.
R	6 in. = 0.5 ft.	$2\frac{1}{2}$ in. = 0.208 ft.
0.00034 $W/A\ N^2R$	34.4	111.38

for high speed and the other for slow speed, the dimensions given Table 3 were chosen.

TABLE 4 PISTON POSITION FACTORS

CALCULATED BY THE FORMULA

$$X/R = \sqrt{(a+1)^2 - k^2} - \cos\theta - a + \tfrac{1}{2} a^{-1} k^2 - a^{-1} k \sin\theta + \tfrac{1}{2} a^{-1} \sin^2\theta$$

FROM BEGINNING OF STROKE TOWARDS THE CRANK SHAFT. MULTIPLY BY CRANK RAD TO FIND PISTON POSITION. (NOTE: CRANK RADIUS IS NOT ONE-HALF OF THE STROKE

Crank Angle	$L + R = 3$		$L + R = 4\frac{1}{2}$	
	Offset = 0.30 R	Offset = 0.50 R	Offset = 0.30 R	Offset = 0.5C
4°18′	0	.....	.....	.....
7°11′	.....	0	.....	.....
3°7′	.....	.....	0	.....
5°13′	.....	.....	.....	0
15	0.023	0.012	0.026	0.018
30	0.129	0.103	0.130	0.111
45	0.309	0.269	0.303	0.275
60	0.525	0.491	0.527	0.492
75	0.804	0.746	0.782	0.742
90	1.070	1.010	1.046	1.005
105	1.321	1.263	1.299	1.26
120	1.525	1.491	1.527	1.49
135	1.723	1.683	1.717	1.69
150	1.861	1.835	1.862	1.84
165	1.955	1.944	1.958	1.95
180	2.0037	2.0102	2.0018	2.0049
188°38′	2.0114	......	......	......
194°29′	......	2.0321	......	......
184°55′	......	......	2.0047	......
188°13′	......	......	......	2.0131
195	2.0065	2.0303	1.992	2.007
210	1.961	2.001	1.929	1.954
225	1.865	1.918	1.810	1.846
240	1.699	1.779	1.643	1.684
255	1.515	1.585	1.429	1.475
270	1.270	1.343	1.179	1.227
285	0.997	1.068	0.911	0.957
300	0.699	0.779	0.643	0.684
315	0.471	0.504	0.407	0.432
330	0.229	0.269	0.197	0.222
345	0.075	0.098	0.061	0.075
360	0.0037	0.0102	0.0018	0.0049

17 The piston position factors and inertia factors are given Tables 4 and 5, and Fig. 3 to Fig. 8 give the curves of inertia forc

The full lines represent the slow-speed and the mixed lines the high-speed cases. Inertia curves and side-pressure curves were plotted for ratios of L/R of $4\frac{1}{2}$ and 3, and for offsets of zero, $0.30\,R$, and $0.50\,R$ for both high and slow speeds, making twelve cases in all. When there is an offset the inertia curve must be plotted for 360 deg. instead of 180, since for the return stroke it is not the reverse of that for the forward stroke, as is the case when there is no offset.

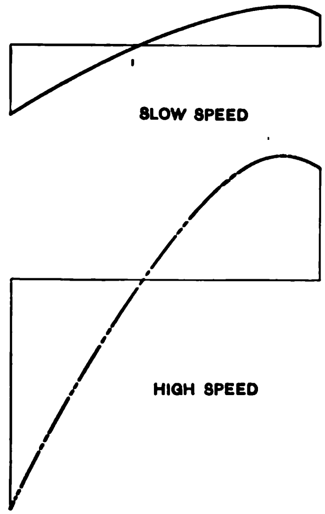
TABLE 5 INERTIA FACTORS

Calculated by Formula $(a^{-1}k \sin\theta + \cos\theta + a^{-1}\cos 2\theta)$

Angle	$L/R = 3$		$L/R = 3\frac{1}{2}$	$L/R = 4$	$L/R = 4\frac{1}{2}$		
	$K = 0.30$	$K = 0.50$	$K = 0.20$	$K = 0.30$	$K = 0.30$	$K = 0.40$	$K = 0.50$
15	1.280	1.297	1.229	1.200	1.175	1.181	1.187
30	1.083	1.116	1.037	1.028	1.010	1.021	1.033
45	.778	.825	.747	.760	.754	.769	.785
60	.419	.477	.406	.440	.447	.465	.485
75	.067	.131	.066	.104	.131	.152	.174
90	−.233	−.166	−.229	−.175	−.156	−.134	−.111
105	−.450	−.386	−.451	−.404	−.385	−.364	−.344
120	−.580	−.523	−.594	−.560	−.553	−.535	−.515
135	−.636	−.589	−.666	−.654	−.660	−.645	−.629
150	−.650	−.617	−.695	−.704	−.722	−.711	−.699
165	−.653	−.635	−.703	−.731	−.757	−.751	−.745
180	−.667	−.667	−.714	−.750	−.778	−.778	−.777
195	−.703	−.721	−.733	−.769	−.791	−.797	−.803
210	−.750	−.783	−.752	−.778	−.788	−.799	−.810
225	−.778	−.825	−.747	−.760	−.754	−.769	−.785
240	−.753	−.811	−.692	−.690	−.669	−.687	−.707
255	−.644	−.708	−.561	−.548	−.513	−.534	−.556
270	−.433	−.500	−.343	−.325	−.288	−.310	−.333
285	−.127	−.191	−.044	−.040	.003	.018	−.040
300	.246	.189	.308	.310	.331	.313	.293
315	.636	.589	.667	.654	.660	.545	.629
330	.983	.950	.981	.954	.944	.933	.917
345	1.228	1.211	1.199	1.164	1.141	1.135	1.129
360	1.333	1.333	1.286	1.250	1.222	1.222	1.222

18 Comparing Fig. 3 with Fig. 6 a slight hump is noticed at the right-hand side in the former but not in the latter. This is probably due to error in the formula, for the small value of L/R since the force could not be higher near the end of the stroke than at the end.

19 The general effect of offsetting on the inertia curve is shown in Fig. 9, where the curves for $L/R = 3$, offsets = zero and $0.50\,R$, are compared, the curve for no-offset being in full lines.



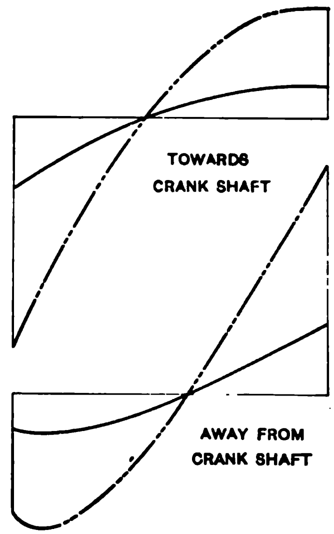


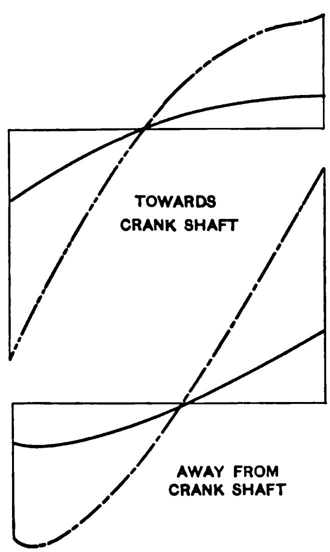
FIG. 3 OFFSET = ZERO. FIG. 4 OFFSET = 0.30 *R*.

CURVES OF INERTIA FORCES ON PISTON POSITION BASE

Slow Speed: r.p.m. = 450; R = 6; $\frac{W}{A}$ = 1 lb.; L + R = 3.

High Speed: r.p.m. = 1500; R = $2\frac{1}{2}$ in.; $\frac{W}{A}$ = 0.7 lb.; L + R = 3.

Full Lines, Slow Speed; Mixed Lines, High Speed.



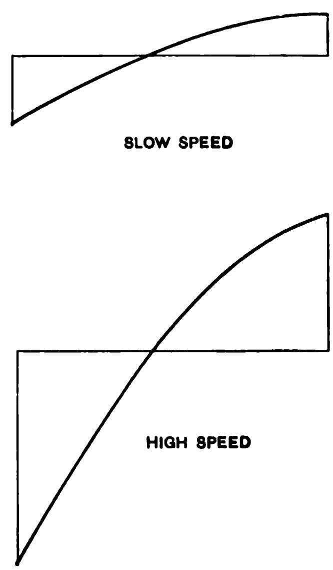
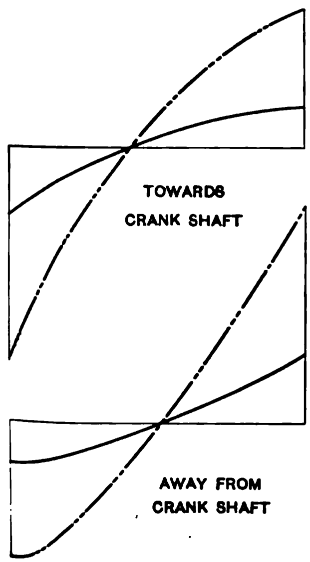


FIG. 5 L + R = 3. Offset = 0.50 R. FIG. 6 L + R = $4\frac{1}{2}$. Offset =

CURVES OF INERTIA FORCES ON PISTON POSITION BASE



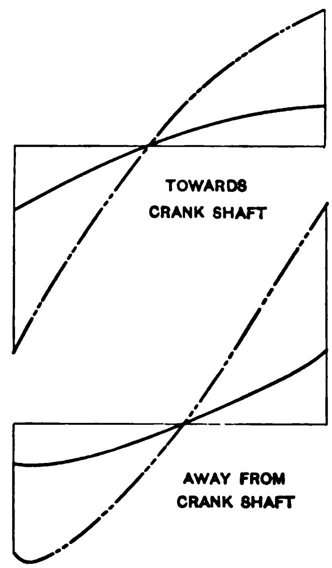


FIG. 7 L + R = 4½. Offset = 0.30 R. FIG. 8 L + R = 4½. Offset = 0.50 R

CURVES OF INERTIA FORCES ON PISTON POSITION BASE

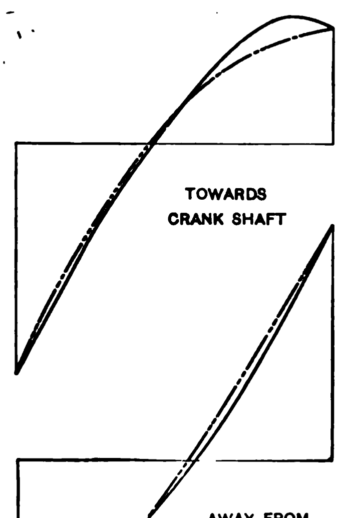


FIG. 9 INERTIA FORCE CURVES SHOWING EFFECT OF OFFSETTING

L+R=3, High Speed. Full Lines, Offset = Zero; Mixed Lines, Offset = 0.50 R

20 The curve for an offset is a little flatter near the end of the out stroke and the hump is increased near the beginning of the return stroke, which is probably due to inaccuracy in the formula.

21 The curves of side-pressures are shown in Fig. 10 to 15. The maximum side-pressure, its cause (whether combined gas and inertia pressure or inertia force alone) and its location are given in Table 6.

TABLE 6 MAXIMUM SIDE PRESSURES, CAUSE AND LOCATION

Ratio L/R	Offset	Max. Side-Pressure		Due to		Stroke	
		Slow	High	Slow	High	Slow	High
4½	0	25	26	Gas	Gas	1	1
4½	0.30 R	17	24	Gas	Inertia	1	2
4½	0.50 R	12	28	Gas	Inertia	1	2
3	0	35½	45	Gas	Gas	1	1
3	0.30 R	23	39	Gas	Inertia	1	2
3	0.50 R	20	51	Inertia	Inertia	2	2

22 From this for $L/R = 4\frac{1}{2}$, slow-speed, maximum pressure is lowest with 0.50 R offset, and if the offset were further increased the maximum side-pressure would probably not be reduced as the values at the beginning of the second and fourth strokes would increase, and now they are already 11 so that any increase would soon cause an increase in the maximum value instead of a decrease. In the case of $L/R = 3$ the lowest maximum value occurs when the offset is 0.50 R, but in this case it is possible that the offset is already a trifle large, as the maximum value occurs at the beginning of the second stroke, although it is not much greater than that in the first stroke, being 20 in the former case and 18 in the latter. Hence for the slow speed the best offset would seem to be about 0.50 R as far as the maximum value of side-pressure is concerned.

23 In the case of $L/R = 4\frac{1}{2}$, high speed, the maximum side-pressure due to inertia force at the beginning of the second stroke seems to increase with the amount of offset, while the maximum value due to the gas pressure in the first stroke seems to decrease with the increase in offset. These values are shown in Table 7. $L/R = 4\frac{1}{2}$.

TABLE 7 COMPARISON OF SIDE PRESSURES FOR $L/R = 4\frac{1}{2}$

Offset		Zero	0.30 R	0.50 R
Side pressures due to.........	Gas pressure, 1st stroke...	26	17	13
	Inertia, 2d stroke.........	15	24	28

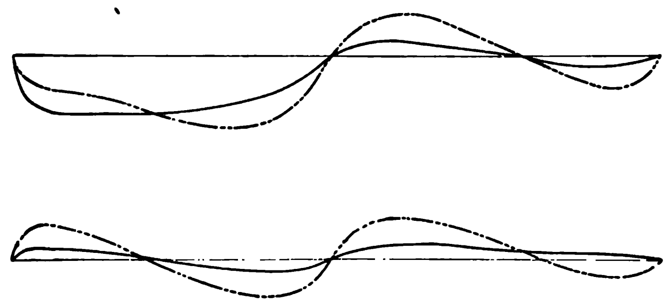


FIG. 10 CURVE OF SIDE PRESSURES ON PISTON POSITION BASE
C ÷ R = 3, No Offset. Full Lines, Slow Speed; Mixed Lines, High Speed

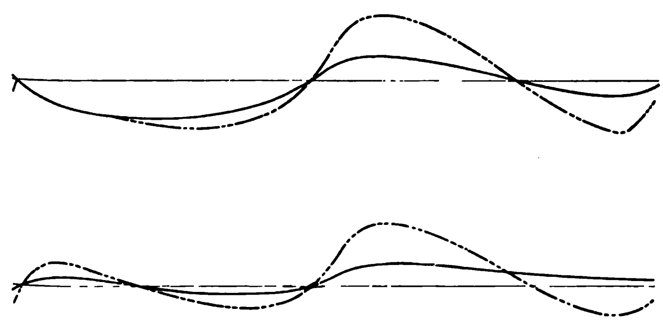


Fig. 11 CURVE OF SIDE PRESSURES ON PISTON POSITION BASE
L ÷ R = 3, Offset = 0.30 R. Full Lines, Slow Speed; Mixed Lines, High Speed

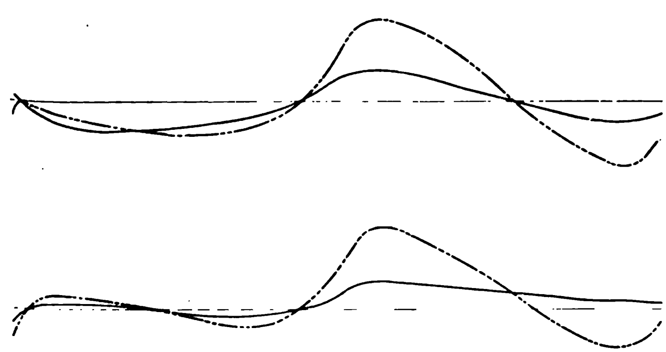


FIG. 12 CURVE OF SIDE PRESSURES ON PISTON POSITION BASE
L ÷ R = 3, Offset = 0.50 R. Full Lines, Slow Speed; Mixed Lines, High Speed

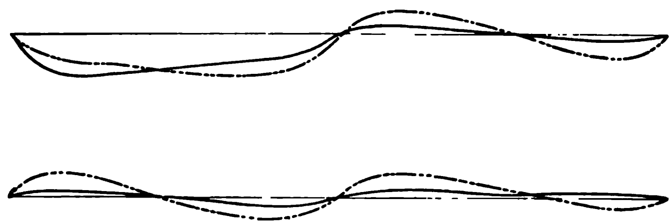


Fig. 13 Curve of Side Pressures on Piston Position Base

L + R = 4½, Offset = Zero. Full Lines, Slow Speed; Mixed Lines, High Speed

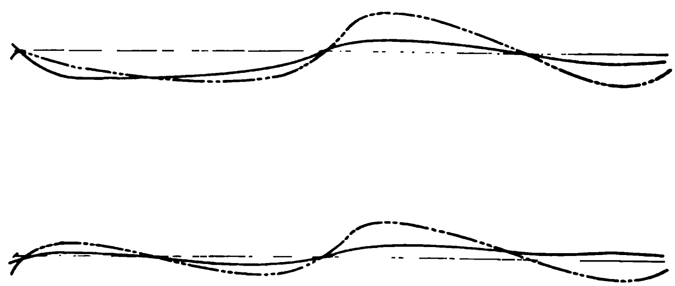


Fig. 14 Curve of Side Pressures on Piston Position Base

L + R = 4½, Offset = 0.30 R. Full Lines, Slow Speed; Mixed Lines, High Speed

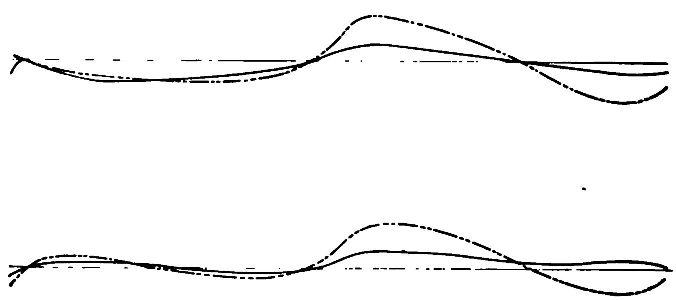


Fig. 15 Curve of Side Pressures on Piston Position Base

L + R = 4½, Offset = 0.50 R. Full Lines, Slow Speed; Mixed Lines, High Speed

24 Plotting curves of these values, the most favorable offset as far as maximum side-pressure is concerned is 0.16 R when $L/R = 4\frac{1}{2}$. This curve is shown in Fig. 16.

25 See Table 8 for values $L/R = 3$. This would place the best offset for $L/R = 3$, as far as maximum side-pressure is concerned,

TABLE 8 COMPARISON OF SIDE PRESSURES FOR $L/R = 3$

Offset		Zero	0.30 R	0.50 R
Side pressures due to.........	Gas pressure, 1st stroke...	45	30	20
	Inertia, 2d stroke.........	25	39	51

as 0.20 R, which would seem to indicate that a greater offset would be desirable as the ratio L/R decreased. It remains to determine if possible the best offset as far as the work done in side-pressure is concerned.

26 The work done is proportional to the areas included between the axis and the curve of side-pressures. It seems to make no difference whether a larger amount of work is done on one side than on the other, or in other words there seems to be no advantage in having the work done on each side the same, unless at some time it might be desired to rebore the cylinder, in which case wear occurring all on one side might have left the walls too thin or might necessitate the removal of much more metal.

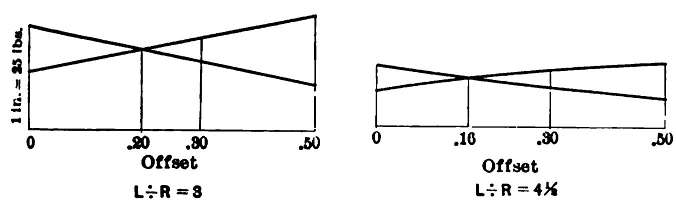


FIG. 16 CURVE SHOWING VARIATION OF SIDE PRESSURE WITH OFFSET

27 Table 9 shows the results of measuring the areas, namely the ratio of work done in one case to that in each of the other cases, and also the actual average side-pressures. For the *slow-speed* case there seems to be a decrease of work done on one side, and an increase of work done on the other side, resulting in a decrease in the total work done with the offset, which would indicate that the greater the offset, the less the loss in work. The average side-pressure also decreases with the offset, although it is less with no-offset when $L/R = 4\frac{1}{2}$ than with 0.50 R offset when $L/R = 3$.

28 In the case of the *high speed* the areas on one side decrease with the offset while those on the other side increase, but the totals for $L/R = 4\frac{1}{2}$ decrease and then increase, while for $L/R = 3$ they continue to increase, and of course the same is true for the mean side-pressure.

TABLE 9

		SLOW SPEED				HIGH SPEED			
Ratio L/R	Offset	+Area	−Area	Total Area	Average Side Pressure	+Area	−Area	Total Area	Average Side Pressure
$4\frac{1}{2}$	0	1.61	− .53	2.14	6.6	2.33	−1.11	3.44	10.7
$4\frac{1}{2}$	0.30 R	1.09	− .90	1.99	6.2	1.78	−1.58	3.36	10.5
$4\frac{1}{2}$	0.50 R	0.74	− .93	1.67	5.2	1.62	−1.90	3.52	11.
3	0	2.59	− .92	3.51	11.	4.11	−1.45	5.56	17.3
3	0.30 R	1.55	−1.15	2.70	8.4	3.01	−2.76	5.77	18.0
3	0.50 R	1.11	−1.39	2.50	7.8	2.47	−3.40	5.87	18.3

29 This would seem to indicate that there is little if anything to be gained by an offset in regard to work done by the piston on the walls of the cylinder when the inertia force is very high, since what is gained on one side is more than made up in loss on the other side.

30 If it is of sufficient importance to have the work done on each side of the cylinder the same, we may plot curves of the work done on each side and note where they intersect, as in Fig. 17. In the case of the slow speed we would have the work done on each side equal when the offset was about 0.40 R and in the high speed this point would be about 0.36 R.

31 *Thermal Cycle.* Offsetting increases the length of stroke, which gives increased expansion to the gas, and increases the piston velocity on the in-stroke, giving greater inertia to the gas on the exhaust and reducing the amount of leakage by the piston on the compression stroke. This refers to the 4-cycle gas engine.

32 *Lubrication.* The curves of side-pressure show the manner in which the side-pressure changes sides, which is a good thing for lubrication. This changing sides would be about the same for offset or no-offset except in the case when the offset is equal to the crank radius. Here the pressure is almost continually on one side of the cylinder so that oil would with difficulty be introduced between the surfaces. Other things being even, except for this extreme case, the reduction in amount of side-pressure should make lubrication more satisfactory.

33 *Vibration and Balance.* Revolving masses and reciprocating masses may cause vibration in gas engines. Offsetting the cylinders would not affect the revolving masses at all but does change the curves of inertia forces, as already shown in Fig. 3 to 8. These inertia-force diagrams are now combined in different ways according to different arrangements of cylinders, and are compared with similar curves when there is no offset.

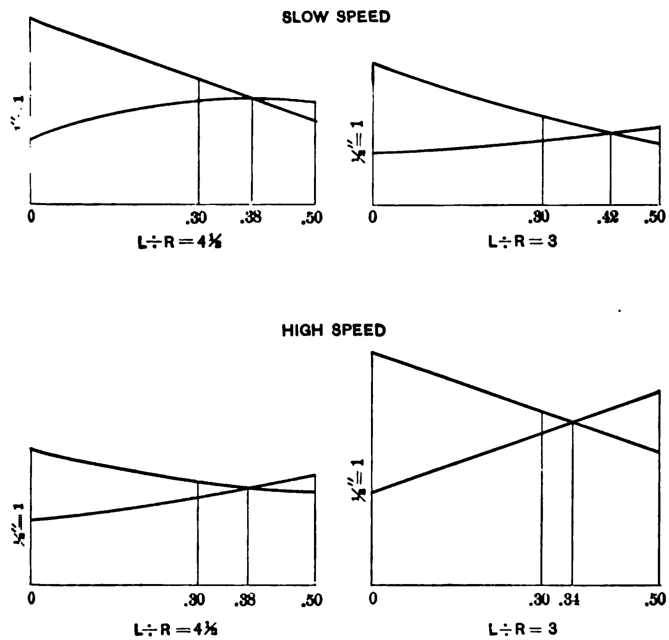


Fig. 17 Curves Showing Offset when Work is same on Each Side of Cylinder

34 The following discussion applies only to the 4-cycle type of gas engine, whose arrangements are:

a Single cylinder.
b Two-cylinder vertical.
c Two-cylinder opposed.
d Three-cylinder vertical.
e Four-cylinder vertical.
f Four-cylinder double-opposed.
g Six-cylinder vertical.

35 For this comparison the high speed case, when $L/R = 4\frac{1}{2}$, was chosen, the offset being equal to zero and one-half the crank radius.

36 Fig. 18 shows the inertia curves for a single-cylinder engine. These curves must be shown for 360 deg. of crank angle, for they differ on the return and forward strokes. The curves for no-offset are shown in full lines and for 0.50 R offset in dotted lines. The difference between the two curves is apparent.

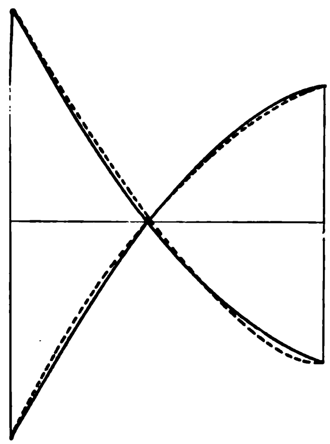


Fig. 18 Curves of Inertia Forces on Piston Position Base. Single-Cylinder Engine

L + R = 4½. Full Line, Offset = Zero; Dotted Line, Offset = 0.50 R. High Speed Case

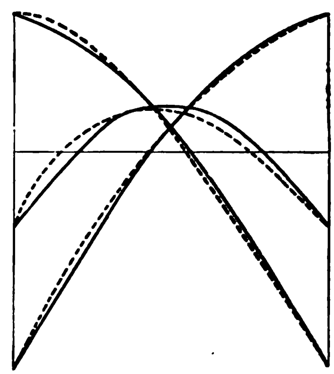


Fig. 19 Curves of Free Unbalanced Inertia Forces. Two-Cylinder Vertical Engine

L + R = 4½; Full Line, Offset = Zero; Dotted Line, Offset = 0.50 R. High Speed Case

37 Fig. 19 shows the curve of free inertia forces for a two-cylinder vertical arrangement. These curves are not so very different; the one for an offset being nearly the same as the other but moved along a little instead of being symmetrical with a center line perpendicular to the axis. The maximum values of the forces are about the same. The vibrations when there is an offset would have unequal periods but about the same amplitudes. For the four-cylinder vertical case the ordinates of these curves could be doubled and the same general difference would exist.

38 In the case of the two-cylinder opposed motor with cranks at 180 deg., the inertia forces would be balanced whether the cylinders were offset or not, but in the case of an offset a new couple in

a plane perpendicular to the axis would be introduced due wholly to the offsetting, which cannot be balanced. The couple in an axial plane due to the cylinders being not in line would be the same, offset or not, but with an offset there would be added another couple in this plane due to the offset, which would not be balanced.

39 In the case of a four-cylinder double-opposed motor the forces would be balanced and also the couples in an axial plane, but the couples in the plane perpendicular to the axis would be doubled while those in the axial plane due to the offset would be balanced.

40 The case of a three-cylinder vertical arrangement can be discussed by considering the formula for the inertia forces,

$$F/A = 0.00034\ W/A\ N^2R\ (a^{-1}k \sin\theta + \cos\theta + a^{-1}\cos 2\theta)$$

Let the cranks be at 120 deg.; then the crank angles will be θ, $\theta + 120$, and $\theta + 240$. Substituting these values in the formula, the part in brackets reduces to zero, showing that the inertia-forces are balanced. However, the moments resulting from these forces are not balanced. By placing two three-cylinder vertical engines together so that the two middle cranks are in the same plane the six-cylinder engine is obtained, in which the inertia forces and couples are both balanced.

41 From this discussion it follows that offsetting the cylinders has no effect on the vibration due to the reciprocating parts, except in the case of the 2-cylinder opposed and 4-cylinder double-opposed arrangements of cylinders. In these cases the offsetting increases the unbalanced inertia-force couples by adding new ones.

42 Vibration may be felt from the irregularity of the turning-effort curves, which for four different cases are shown in Fig. 22. There is such a slight difference here that it can be neglected, especially since the turning-effort curve depends so directly on the shape of the gas card, which may vary considerably. The conclusion in regard to vibration would be that offsetting does not affect the vibration appreciably except in the case of a two-cylinder opposed or a four-cylinder double-opposed motor.

GENERAL CONCLUSIONS

43 The following are perfectly general conclusions, to be followed by a more definite comparison of actual engines:

a The length of stroke for a given crank radius increases as the offset increases.

b The length of stroke for a given crank radius for any offset decreases as the ratio of L/R increases.

c The increase in length of stroke causes an increase in average piston speed.

d Offsetting the cylinders makes the crank and connecting rod train a quick return mechanism.

e When the cylinders are offset the crank passes over an angle greater than 180 deg. during the out-stroke of the piston, and less than 180 deg. during the in-stroke.

f The average velocity of the piston is greater on the exhaust and compression than on the explosion and suction strokes.

g Offsetting the cylinders reduces the angularity of the connecting rod on the out-stroke and increases it on the in-stroke.

h When there is an offset, the side-pressure of the piston on the cylinder walls does not change sides at the end of the stroke, but just after the beginning and just before the end of the out-stroke.

i The place where this change of side-pressure occurs approaches the middle of the stroke as the amount of offset approaches the crank radius.

j With no offset, high inertia forces do not greatly increase the maximum side-pressure during the explosion stroke, but do increase it considerably during all of the other strokes, and this effect is slightly greater as the ratio of L/R decreases.

k With no offset the work done increases with the inertia-force and as the ratio of L/R decreases.

l For low inertia forces, as far as the maximum value of side-pressure is concerned the best offset is one-half the crank radius.

m Considering the maximum value of the side-pressure only, the most favorable value for the offset decreases as the inertia-forces increase, for any ratio of L/R, but does not decrease as rapidly, for smaller values of the ratio L/R.

n For low inertia forces, the work done by the piston on the cylinder walls decreases as the offset increases, but of course is greater for smaller values of L/R.

o For very high inertia forces, the work done decreases slightly with the offset up to 0.40 of the crank radius for value of $L/R = 4\frac{1}{2}$.

p For very high inertia forces, and small values of L/R, there is no advantage in an offset, as far as the work done by the piston on the cylinder walls is concerned.

q The thermal cycle is slightly benefited by offsetting and the benefit increases with the amount of offset.

r Lubrication should be slightly improved by offsetting the cylinders.

s Vibration due to the free inertia forces is no different except in the case of a two-cylinder opposed or four-cylinder double-opposed motor.

44 Table 10 gives data of gas engines that have been constructed and put in operation. The average crank radius is about 2½ in., the

TABLE 10 DATA OF GAS ENGINES HAVING CYLINDERS OFFSET

No.	R Crank Radius	L Length of Connecting Rod	Ratio L/R	Offset Amount	Offset Per cent	Length of Piston	Diam. of Bore	Ratio Piston Length to Diameter
1	7	23⅝	3.37	7	100	14	9.47	1.47
2	2 9/16	8 15/16	3.48	⅝	24.	6	5½	1.09
3	2¼	8½	3.77	⅜	16.6	6	4	1.5
4	2½	9½	3.8	⅜	15	5⅝	5	1.125
5	2½	10	4.0	⅜	15	6½	4¾	1.37
6	3	12¼	4.08	9/16	18.75	5⅝	....	....
7	2⅜	9¾	4.1	½	21	5⅜	5	1.075
8	2½	10½	4.2	⅞	35	5 7/32	4¾	1.098
9	2¾	12	4.36	15/16	34	6¾	5¼	1.28
10	2¼	10½	4.66	⅞	38	5½	4¼	1.29
11	2½	12	4.8	1	40	6	4¾	1.26
12	2½	12	4.8	1	40	6¼	5	1.25
13	2½	12	4.8	1	40	6	4¾	1.26
14	2½	....	....	1¼	50	....	....	....
15	2¾	....	....	1⅛	40	....	....	....
16	3	....	....	1	33	....	....	...
17	2½	....	....	¾	30	....	....	....

Westinghouse standard engine has an offset of 50 per cent of crank radius.

ratio of L/R varies from 3.37 to 4.8 and the percentage of offset varies from 15 to 50. The average diameter of cylinder-bore is 4.81 in. and the average ratio of length of piston to diameter is 1.24.

45 For comparison of engines the following dimensions were taken:

Crank radius	=	2½ in.
Diameter of bore	=	4¾ in.
R.p.m.	=	1000

Weight of reciprocating parts per square inch of piston head area = 0.6 lb.

Ratios of $L/R = 3\frac{1}{2}$, 4, and $4\frac{1}{2}$ and an offset, for each of the values of L/R, the largest amount practicable. These offsets are:

$L/R = 4\frac{1}{2}$	Offset = zero
" $= 4\frac{1}{2}$	" $= 0.40\ R$
" $= 4$	" $= 0.30\ R$
" $= 3\frac{1}{2}$	" $= 0.20$ R

46 Tables were prepared for each of the above cases, and values calculated for crank angles varying by increments of 15 deg. each. Each of these tables contained values for the crank angle, piston position factor, the actual piston position, the gas pressure, inertia factor, inertia force, piston pin pressure, tangential factor, and the turning force from which the inertia curves and turning effort curves were plotted.

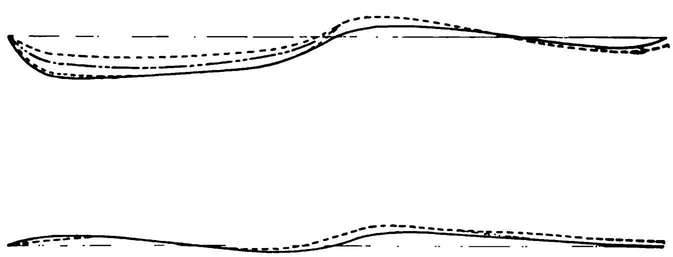


Fig. 20 Curve of Side Pressures on Piston Position Base

R.p.m. = 1000; $\frac{W}{A}$ = 0.60 lb.; R = 2½ in. Full Line, L + R = 4½; Offset = Zero Broken Line, L ÷ R = 4½; Offset = 0.40 R. Mixed Line, L + R = 4; Offset = 0.30 R. Dotted Line, L + R = 3½; Offset = 0.20 R.

47 Careful comparison of the curves in Fig. 20 will show a slight difference between them, but not enough to warrant the trouble of plotting them separately for use in connection with the gas pressures to find the piston-pin forces from which the side-pressures are determined.

48 The inertia forces shown in Fig. 21 were combined with the gas pressures and the curves of side-pressures plotted as before, with the results shown in Fig. 20.

49 The maximum values for the side-pressure were determined and the areas representing the work done by the piston on the cylinder

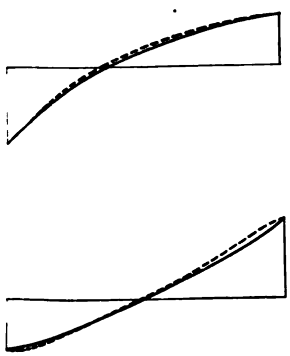


Fig. 21 Curves of Inertia Forces on Piston Position Base
R.p.m. = 1000;
R = 2½ in.; $\frac{W}{A}$ = 0.60 lb.
1 L + R = 4½ Offset = Zero
2 L + R = 4½ Offset = 0.40 R
3 L + R = 4 Offset = 0.30 R
4 L + R = 3½ Offset = 0.20 R

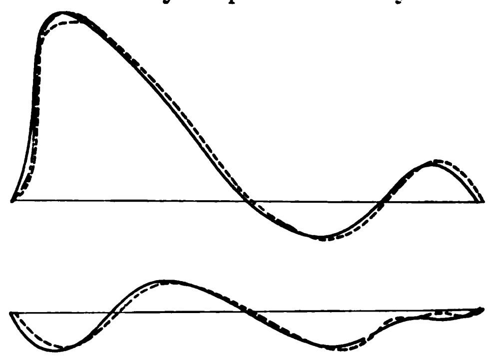


Fig 22 Turning Effort Curves on Crank Angle Base
Full Line, L + R = 4½ Offset = Zero
Mixed Line, L + R = 3½ Offset = 0.20 R
Dotted Line, L + R = 4½ Offset = 0.40 R
Broken Line, L + R = 4 Offset = 0.30 R

walls were carefully measured and recorded (see Table 11). As far as these quantities are concerned, the best value is L/R = 4½, offset = 0.40 R. The turning-effort curves (shown in Fig. 22) are so nearly alike that the difference is hardly worth mentioning.

TABLE 11 SIDE PRESSURES AND WORK DONE ON CYLINDER BY PISTON OF TYPICAL ENGINE

L/R	Offset	Max. Side Pressure		Work Done in Area Units		
		One Side	Other Side	One Side	Other Side	Total
4½	zero	24	7	1.55	0.60	2.15
4½	0.40	12	12	0.93	0.87	1.80
4	0.30	17	12	1.17	0.82	1.99
3½	0.20	23	12	1.53	0.81	2.54

50 Table 12 gives a comparison of the four cases chosen. This table explains itself, but a short discussion may bring out the important points more clearly.

51 There is a slight increase in the length of stroke, but less than one-half of one per cent, so that it amounts to very little. The angle passed over by the crank during the out-stroke is slightly greater than 180 deg. and the greatest gain is 1.7 per cent, which is small. The first great difference occurs in the length of connecting rod, No. 4 effecting a saving of 2.50 in. or 22.2 per cent.

52 Referring to the next line, the distance from the center of the crank-shaft to the position of the center of the piston pin at the end of the stroke, is a measure of the height of the engine and shows a decrease corresponding to the value of L/R.

53 The maximum side-pressure decreases with the offset and increases with the decrease in value of the ratio L/R, so the best case would be No. 2, where L/R is largest and the offset is also largest. Here a reduction of 50 per cent is gained, which reduced the necessary length of the piston 44 per cent. No. 4 is the worst case, L/R very small and the offset also small and then the side-pressure is a trifle less than it is with no offset. The maximum value of the side-pressure affects the length of the piston and consequently the height of the engine, and the length of the cylinder, and so the weight of the cylinder and engine, and the weight of the piston and the corresponding weight of the reciprocating parts, and so the inertia force. The length of the piston is reduced 43.7 per cent in No. 2, 24.4 per cent in No. 3, and 3.6 per cent in No. 4. The ratio of length of piston to diameter is rather small in No. 2 but is not unusual in the other cases.

54 If it is not desired to take advantage of the maximum value of the side-pressure by reducing the length of the piston, it can be made 1.20 times the diameter, a usual value as is seen in Table 10, which would reduce the pressure per square inch of projected area and so increase the chances of satisfactory lubrication. The reduction of this pressure per square inch of projected area is shown in the next row.

55 In order to find the exact resulting height of the cylinder up to the top of the piston at the end of the in-stroke it is necessary to calculate the position of the piston pin in the piston. This is done in the next row, by making the sums of the products of the areas with the distance from the piston pin center to their centers balance on each side of the piston pin. In case No. 2, with $L/R = 4\frac{1}{2}$, a 50 per cent offset might have been used without interference and this would give better results than 0.40 R offset, but in case No. 4, 0.20 R is undoubtedly about as much as could be used although it would be desirable to use more if a very low engine were wanted.

TABLE 12 COMPARISON OF ACTUAL ENGINES

Specifications	No. 1	No. 2	Gain	No. 3	Gain	No. 4	Gain
Crank Radius	$2\frac{1}{2}$	$2\frac{1}{2}$		$2\frac{1}{2}$		$2\frac{1}{2}$	
Diameter of Bore	$4\frac{3}{4}$	$4\frac{3}{4}$		$4\frac{3}{4}$		$4\frac{3}{4}$	
R.p.m.	1000	1000		1000		1000	
W/A	.60	.60		.60		.60	
Ratio L/R	$4\frac{1}{2}$	$4\frac{1}{2}$		4		$3\frac{1}{2}$	
Offset	zero	.40 R		.30 R		.20 R	
Length of Stroke	5.000	5.021	0.42%	5.015	0.30%	5.009	0.18%
Crank Angle passed over, Stroke toward Crank Shaft	180°	182.4°	1.30%	183.08°	1.7%	182.03°	1.1%
Crank Angle passed over, Stroke away from Crank Shaft	180°	177.6°		176.92		177.97°	
Length of Connecting Rod	11.25 in.	11.25 in.		10.0 in.	1.25	8.75 in.	2.50 in.
Distance. Center of Crank Shaft to end of Stroke	13.75 in.	13.693 in.	.057 in. 0.41%	12.477 in.	1.273 in. 9.2	11.239 in.	2.51 in. 18.2%
Maximum Side Pressure: One Side	24	12	50%	17	29%	23	4.19%
Maximum Side Pressure: Other Side	7	12		12		12	
Piston Length; 15 lb. per sq. in. proj. area; $\frac{7}{8}$ in. for rings, etc.	6.85 in.	3.86 in.	2.99 in. 43.7%	5.18 in.	1.67 in. 24.4%	6.60 in.	.25 in. 3.6%
Ratio: Piston Length to Diam.	1.44	.81		1.09		1.39	
Pressure per sq. in. proj. area Piston Length = 1.2 diam = 5.7 in	15.7	7.8	50%	11.3	28.1%	15.0	4.45%
Distance: Piston Pin to back end of piston	3.10 in.	1.68 in.		2.29 in.		2.98 in	
Distance: Crank Shaft to end of piston, end of in-stroke	17.5 in.	15.87 in.	1.63 in. 9.3%	15.36 in.	2.14 in. 12.2%	14.86 in.	2.64 in. 15.1%
Length of cylinder, end of piston in-stroke to other end of piston out-stroke	11.85	8.88 in.	2.97 in. 25.1%	10.19 in.	1.66 in. 14%	11.61 in.	.24 in. 2.0%
Work done by Piston on side of Cylinder, square inches	2.15	1.80	.35 16.6%	1.99	.16 7.4%	2.54	−.39 −18.1%
Average side-pressure	6.4	5.6		6.2		7.9	

56 The next two rows show the distance from the center of the crank-shaft to the end of the piston at the end of the in-stroke, which is a measure of the height and the length of the cylinder and also of the weight of the cylinder. As regards the height a gain of 15 per cent may be had in No. 4. No. 2 gives the shortest cylinder, 25 per cent shorter than No. 1, while No. 4 gives one only 2 per cent shorter. It must be borne in mind that these values are for 1000 r.p.m. and that the value of the maximum side-pressure will increase with the speed. However, the low value of the pressure per square inch of projected area, 15 lb., allows a considerable increase before a dangerous value is reached.

57 The total amount of lost work is shown in the next line. No. 2 gives the best value, a saving of 16 per cent, while No. 4 gives a loss of 18 per cent.

58 In working out a satisfactory solution it would seem that one of two predominating ideas should be followed. Either a very low engine should be aimed at in which everything is sacrificed to height, or else the important object is to reduce to a minimum the side-pressure and the work lost due to friction resulting from side-pressure.

59 In the first case, let $L/R = 3\frac{1}{2}$, offset as much as possible without interference, and a reduction in height of 13 to 15 per cent may be had. This means a reduction and a saving in weight of the connecting rod, cylinder, valve stems, exhaust pipes, inlet pipes, and piston. The actual saving in length in the case above is $2\frac{5}{8}$ in. There will be some increase in the work lost in friction due to the increased average pressure of the piston on the cylinder walls.

60 If a reduction in height is not of primary importance, then a ratio of $L/R = 4\frac{1}{2}$ and an offset of 0.40 R to 0.50 R would seem to give the best results. This gives a reduction in total height of 8 or 9 per cent, a reduction in piston length of 44 to 45 per cent, a reduction in cylinder length of about 20 per cent, and a saving in lost work of about 16 per cent. These reductions would cause a further reduction in weight of piston, weight of cylinder, weight of valve stems, weight of exhaust and inlet manifolds, and a reduction of inertia effects as well as an increased life to the piston, piston rings and cylinder. In this case it might not be desirable to take full advantage of the reduction in length of piston, making it less than the stroke because the oil hole in the side of the cylinder, if one were used, would be uncovered at one end of the stroke or the other.

61 In concluding this comparison, the most desirable offset seems to be as much as can be practically obtained with ratios of $L/R = 4$

and greater, with a decided gain over an engine with no offset for speeds less than 1400 to 1500 r.p.m. The subject may be summed up as follows:

62 Improvements due to offsetting, (1) in the thermal cycle, (2) in the mechanical arrangement, (3) in the turning effort curve, and (4) in lubrication, are very slight and may be neglected. The real advantages are:

a A reduction of the frictional losses due to the pressure of the piston on the walls of the cylinder, resulting in a slight increase in mechanical efficiency and less wear of the piston, piston rings, and cylinder, and consequently longer life.

b A reduction of the maximum value of the side-pressure of the piston on the walls of the cylinder, allowing the use of shorter connecting rods, shorter pistons, and shorter cylinders, resulting in a shorter and lighter engine and in lower inertia-forces due to the reciprocating parts.

The most important of these advantages would be a considerable saving in weight.

63 The disadvantage of offsetting lies in the fact that the reductions in average side-pressure and maximum side-pressure grow less as the speed and inertia-force increase, so that for a speed of 1400 to 1500 r.p.m. there is either no reduction at all or an increase.

Principal Conclusions

64 Offsetting increases slightly the length of stroke and the crank angle passed over during the stroke toward the crank shaft.

65 The maximum value for the side-pressure of the piston on the cylinder walls decreases as the offset increases up to a value of one-half the crank radius for any ratio of L/R.

66 The work lost in friction due to the side-pressure of the piston on the cylinder walls decreases as the offset increases up to a value of 50 per cent of the crank radius.

67 Both the maximum value of the side-pressure and the work lost in friction increase as the value of the ratio L/R decreases.

68 Offsetting decreases the height and weight of the engine.

69 Offsetting increases the life of the cylinder and piston.

70 Offsetting improves the thermal cycle.

71 The turning-effort curves when the cylinders are offset differ but slightly from those for no-offset.

72 The advantages of offsetting as regards the maximum side-pressure and work lost may be zero or negative for high inertia-forces resulting from speeds of 1500 r.p.m. or more.

DISCUSSION

WINSLOW H. HERSCHEL. In December 1907, I had occasion to investigate the question of offsetting cylinders for large-sized gas engines, and as the conditions are somewhat different from those of the automobile engines considered by Professor Phetteplace, the results may be of interest. I shall consider only the variations in maximum and average pressure on the cylinder walls, since, as indicated in Par. 62 of the paper, these are the main questions at issue.

2 For the sake of simplicity I used the graphical method mentioned in Tolle's Die Regelung der Kraftmaschinen, page 32. The computations were based on an actual card from a four-stroke cycle producer-gas engine, and upon the following data:

$$\text{R.p.m.} = N = 225$$
$$W/A = 4.18$$
$$R = 12 \text{ in.} = 1 \text{ ft.}$$
$$L/R = 5$$
$$0.00034\ W/AN^2R = 71.8$$

This last value, 71.8, happens to be very nearly the average of the two corresponding values, 34.4 and 111.38, given by the author in Table 3. Computations were also made using speeds of 450 and 1000, giving inertia constants of 289 and 1430 respectively, but it soon became evident that there could be no gain from offsetting under these conditions, and the investigation was restricted to the speed of 225.

3 As I understand the paper, the author has considered only vertical engines, or horizontal engines where the weight of the reciprocating parts is so small that its direct effect in increasing or decreasing the pressure on the cylinder walls need not be taken into account. In the present case, however, a distinction must be made between vertical and horizontal engines. For the latter, when the side pressure acts downward, due to gas pressure or inertia forces, the weight of the piston must be added, but when the side pressure acts upward,

the piston weight is subtracted. It should be noted that for a vertical engine, for a given value of 0.00034 W/AN^2R, that is, for a given inertia constant, as I have called it, it would make no difference whether this value were obtained with a large value of W/A and a small value of N, as in my case, or with a small value of W/A and a large value of N, as in the cases used by the author. But on the other hand it would make considerable difference for a horizontal engine where the value of W enters into the computation apart from the inertia constant.

4 By using the same indicator card as for the four-stroke cycle computations, and disregarding the second and third strokes, I

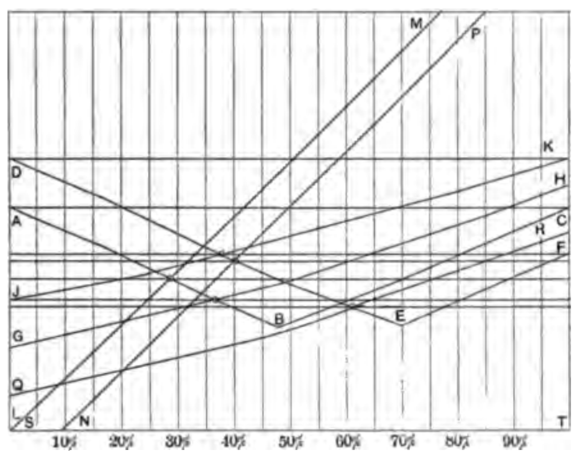


FIG. 1 DIAGRAM SHOWING VARIATION IN MAXIMUM PRESSURE ON CYLINDER WALLS, DUE TO OFFSETTING CYLINDER

obtained, somewhat approximately, the side pressures for a two-stroke cycle engine.

5 Fig. 1 shows the variation in maximum pressure on cylinder walls, or side pressure, due to variations in offset. The ordinates above the base line ST are proportional to the side pressures. The line AB shows the maximum side pressure for a vertical engine at about the middle of the first stroke. As the offset increases the angle of the connecting rod for this middle position decreases, while the angle at the end of the stroke increases, so that for large offsets the maximum side pressure is found at the end of the stroke, with

values as shown by line BC. Similarly for a horizontal engine we ge the lines DE and EF. For the maximum side pressure due to inerti forces we have line GH for a vertical, and line JK for a horizonta engine.

6 It will be noticed that line JK is not parallel to line GH. Th reason for this is that for the line JK we must use the inertia forc near the end of the second stroke, which gives a downward pressur on the cylinder walls to be added to the weight of the piston, and thi downward pressure is not as large as the upward pressure near th beginning of the second stroke, which was used for the line GH.

7 In the case of a two-stroke cycle engine, where we must us the fourth instead of the second stroke, our maximum inertia forc will be near the beginning of the fourth stroke acting upward, so tha the weight of the piston must be subtracted, giving line QR.

8 If we imagine a somewhat earlier ignition than that shown ir

TABLE 1 OFFSET AND PER CENT REDUCTION IN SIDE PRESSURE

Curves	Dominating Factors	Case	% Offset	C
AB and GH	Gas middle first, inertia beginning 2nd	Vertical late ignition, 2–4 Cycle	37.0	[illegible]
AB and LM	Gas middle first, gas beginning 1st	Vertical early ignition, 2–4 Cycle	28.7	[illegible]
DE and JK	Gas middle first, inertia end 2nd	Horizontal late ignition, 4 Cycle	37.7	[illegible]
DE and QR	Gas middle first, inertia beginning 4th	Horizontal late ignition, 2 Cycle	61.3	[illegible]
DE and JK	Gas middle first, inertia end 2nd	Horizontal early ignition, 4 Cycle	37.7	[illegible]
DE and NP	Gas middle first, gas beginning 1st	Horizontal early ignition, 2 Cycle	40.6	[illegible]

Fig. 2 of the paper, the maximum side pressure at or near the begin- ning of the first stroke will be increased. Whether for this reason o not, I found that with large offsets the maximum side pressure of the first stroke was at the beginning of the stroke, acting upward, witl values as shown by line LM for a vertical, and line NP for a horizon- tal engine.

9 Fig. 1 corresponds to Fig. 16 of the paper and may be used in the same way to determine the most favorable offset for the various con- ditions. Table 1 gives the offset and the per cent reduction in side pressure in each case.

10 The author (Par. 22) finds the most favorable offset to be 50 per cent of the crank length for slow speed, and in Par. 24 and Par. 25 he finds it to be about 20 per cent for high speed. These

values may be compared with the first line of Table 1; for the case of a vertical engine with late ignition, the offset is 37.0 per cent, which is nearly the average of 20 per cent and 50 per cent, as might have been expected from the inertia constants.

11 Fig. 2 shows the decrease or increase in average side pressure or total loss of work from side pressure, in per cent of lost work with zero offset. While the use of a different indicator card with a later ignition might have made some difference, it obviously could not have changed the results so materially as in the case of maximum side

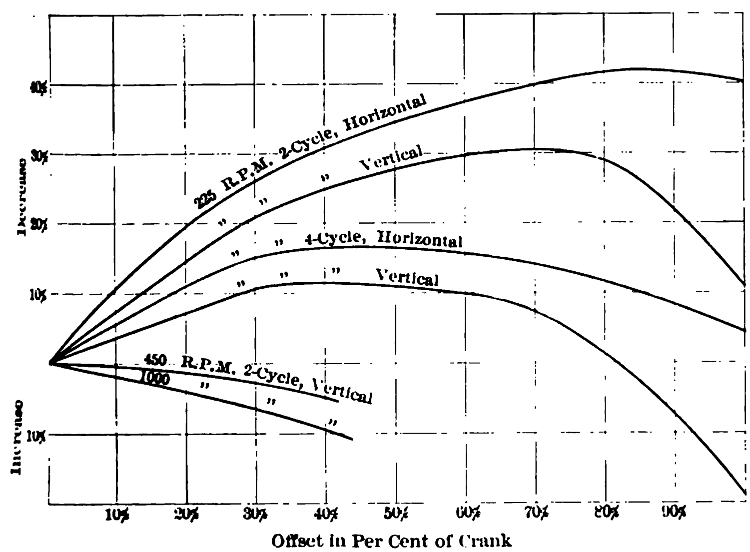


Fig. 2 Diagram Showing Change in Average Pressure on Cylinder Walls Due to Offsetting Cylinder

pressure. Thus the curve marked four-cycle vertical may be fairly compared with the result in Table 9 of the paper, that the most favorable offset lies between 30 and 50 per cent.

12 The curves marked 450 and 1000 r.p.m. show the results of the few computations concerning these speeds not considered in Fig 1.

13 Both Table 1 and Fig. 2 appear to indicate that more could be gained from offsetting with a two-cycle than with a four-cycle engine. But at present it is difficult to make general statements about this type of engine, and whether or not this advantage will be attained will depend upon the inertia constant and indicator card shown by these engines.

PRINCIPAL CONCLUSIONS

a An offset cylinder may be employed with least benefit on a high-speed four-stroke cycle vertical engine.

b It may be employed with most benefit on a slow-speed two-stroke cycle horizontal engine.

c The maximum advantageous offset is limited by the side pressure near the beginning of the first stroke.

John H. Norris. I have been designing and building both two-stroke and four-stroke cylinder engines with offset cranks for a number of years. Our concern was so impressed with the advantage that in 1886 they bought the patent right to apply the offset stroke to gas engines. We are still building a few small sizes with offset crank, but there is no practical gain and as fast as we can re-design the engines we find we can get better economy and a more convenient engine, by placing the cylinder directly over the crank shaft. If you want to reverse the direction of rotation of an engine with an offset crank, you are in trouble at once. We have built an offset engine as large as four-cylinder, 14 by 22, and scrapped it. I would like to say, in connection with the large engine above mentioned, that it was built in 1896 and was one of a pair that were to run a suburban electric railroad in the West, on a suction gas producer. We built at that time a suction producer that was reasonably satisfactory. We have had a great deal of success, however, with our small single and double-cylinder engines with offset cranks, of which we have built a large number, though we are just putting on the market engines to replace them, in which we have placed the cylinder directly over the center. We used an offset of from 20 to 25 per cent of the stroke. With this offset the side strain is quite sufficient on the upstroke. These were all vertical engines.

The Author would suggest that the fact that the cylinders were offset was not the real cause for scrapping the 4-cylinder 14 by 22 engine mentioned by Mr. Norris. Of course if stock engines were built, some to rotate in one direction and some in the other, or if it is desirable to build engines that may be reversed, offsetting may not practically be taken advantage of, as Mr. Norris points out. Furthermore, in spite of Mr. Norris' experience and his desire to eradicate offsetting from his product, this practice, in small vertical 4-cycle automobile engines, at least, seems to be increasing.

No. 1241

PRESENTATION OF PORTRAIT OF GEORGE W. MELVILLE

At the Spring Meeting, Washington, May 1909, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, a portrait of Rear-Admiral George W. Melville, U. S. N., Ret., painted by Sigismond de Ivanowski, was presented by friends to the National Gallery. Previous to the presentation of the portrait an address was made by Admiral Melville on The Engineer in the U. S. Navy. This is given in abstract, together with the addresses of presentation by Walter M. McFarland, Mem.Am.Soc.M.E., and of acceptance of the portrait for the Nation by Dr. C. D. Walcott, Secretary of the Smithsonian Institution.

THE ENGINEER IN THE U. S. NAVY

BY REAR-ADMIRAL GEO. W. MELVILLE, U.S.N., RET.

Past-President of the Society

Ten years ago my presidential address before the Society had almost the same theme as my remarks this evening. At that time the Personnel Law was passed, which amalgamated the engineer corps with the line or executive officers of the navy, with the understanding that thenceforth engineering was to be the function of these line-officers. In his report as Chairman of the Personnel Board, ex-President Roosevelt, then Assistant Secretary of the Navy, said, "On the modern war vessel every officer has to be an engineer whether he wants to or not." It is well that these lines should be constantly in mind, for they set forth the only justification for the Personnel Law.

2 Remarks have been made to the effect that a line-officer charged with all these duties would be a hybrid or Jack-of-all-trades. It is to be noted, however, that our naval officers have to perform definite duties. The curriculum of the naval school can be planned to give them a thorough and specific training for the work they have to do. In this respect these young men have a decided advantage over the students of any of our great technical schools, who can receive instruction only in general principles because they themselves rarely know the particular line of engineering work which they will follow. It is, in my judgment, just as ridiculous to speak of our modern line-officers, specially trained for the work

they have to do, as Jacks-of-all-trades, as it would be to apply this designation to a blacksmith, a lawyer or a doctor.

3 I have not forgotten that I am talking of men who go to sea and that the line-officers are responsible for the handling of the ships; in other words, that seamanship is an essential element of the training. It must be remembered that the modern navy has entirely dispensed with sails and that it is a misnomer to call the modern man-of-warsman, a sailor. He is not a sailor, because there are no sails for him to handle. He is a seaman, because he goes to sea. Seamanship is an art, proficiency in which comes almost entirely from practice, so that officers who are given the other portions of the training in the classroom, laboratory and workshop, get the requisite proficiency in seamanship from the practical exercises during their career as midshipmen at the naval school, and in the handling of vessels after they graduate.

4 It is natural to inquire how the amalgamation has worked out in practice. On January 21 of this year, the Chairman of the House Naval Committee quoted from the remarks of the officer who commanded the battleship fleet which cruised around the world, to the effect, "When I got to California, without any engineers, my fleet was in better condition than when it started." It would seem, however, to have agreed much better with the avowed intention of the Personnel Law if he had said, "Our cruise was a great success because every officer was an engineer."

5 The Chairman of the House Naval Committee further said, "It is the opinion of our naval officers, in command of our fleet and ships, that this consolidation has been a splendid thing for the navy, because it makes the man in command of the ship, the master of the ship, a man who understands all the workings of the ship. Before, the command of the ship was in the hands of the engineer. We had to make a change in the curriculum of the Naval Academy whereby the officer of midshipmen there must acquire a knowledge of engineering, further adding to that the experience which he must obtain in the engine room as a watch officer. By reason of these facts, the entire ship is today under the command of an engineer officer, a man who understands all the duties of engineering and who is complete master of the ship."

6 I have been told by officers who have recently served on board ship that one great benefit has resulted from the amalgamation: namely, that the idea just expressed in the above quotation is true; that the commanding officer is now the master of the entire ship. In my early days, few commanding officers felt any interest

in the machinery beyond their demand that it should always be ready for service. If anything went wrong, they washed their hands of the responsibility, which was naturally upon the special body of engineers. They now feel the same keen interest in the machinery that they do in the guns or any other part of the ship, and the chief engineer of the ship (still so-called) is generally looked upon as the officer next in importance after the captain.

7 The part of the new régime about which I have felt misgivings is that thus far there has been no systematic effort to assure training and experience for every line-officer in connection with the motive power. Every young officer should be required to serve an apprenticeship in the engine and fire-rooms, just as he does on deck, but so far as I have been able to learn there has never been careful attention given to this point.

8 Having touched upon the general conditions of the executive side of engineering as affecting the operation and integrity of the machinery at sea, it is now pertinent to consider the prospects with regard to present and future designs. Thus far, this work has remained in the hands of officers specially trained. Unfortunately, the same condition is found here as mentioned in the previous division of the subject. An effort has been made to arouse the interest of some of the younger line-officers by a course of special training for this most important work after the present highly competent and experienced men have retired. There has, however, been no settled policy for this, and the attempt that was started was interrupted by the cruise of the Atlantic Fleet.

9 I am very glad indeed to bear testimony to the fact that the recent designs of the Bureau of Steam Engineering have been highly creditable in every way. In saying this, I feel a touch of personal pride for the reason that the men who have been doing this work were formerly my assistants and received most of their experience during my term of office. I am naturally pleased that the record which was made during my own term is being maintained.

10 When such praise as this can be given in simple truth, what can be thought of the official who plans to discredit the men who have made such a record, and destroy the autonomy of the Bureau by subordinating it to the Bureau charged with the design of hulls? I believe you will agree that my service of a lifetime in the Navy and my record as the head of a great Bureau in the Department, the longest since the Civil War, entitles my opinion to some weight, and I want to register my earnest conviction that any such scheme of consolidation can only bring inefficiency, retrogression and waste.

11 There is still another side to engineering in the navy; namely, the work of the navy yards, which has been prominently before the public during the régime of the last Secretary of the Navy. Changes have been made abolishing the separate departments in the navy yards and consolidating their administration under one officer, whose work, while a vital element in the building of a ship, was certainly not the only important part, and moreover was so different in its nature from the other departments which were absorbed, that it is obvious he could not be an expert on these other lines. To me it was so marvelous as to be almost beyond belief that in this age of specialization a movement so absolutely counter to the spirit of the age should take place in the name of economy and reform. If the methods of great shipyards in civil life, or of the great manufacturing establishments, or the dockyard administration of other countries, had differed from the methods employed in our navy yards, a change would at least have been indicated; if, in these places a system somewhat like the one which it has been attempted to introduce in our navy yards, had been in vogue, there could be some understanding of the change; the facts are however, that in its essential features our navy yard administration was along the very lines which obtain in foreign dockyards, in the great shipyards at home and abroad, and in our great manufacturing establishments.

12 I am led to believe that the present Secretary is giving the matter very careful consideration with a view to undoing the tremendous harm brought about by his predecessor, and I trust he will be well-advised and will restore the yards to their former efficiency. It ought to be said, however, as a matter of record, that these changes were made without any consultation between the late Secretary and the officers most competent from long experience to know what was best. Indeed, by his own statement, the scheme was evolved from his own inner consciousness.

13 Our modern navy is essentially an engineering affair. The vessels themselves are the product of the engineer's brain, and their successful maintenance and utilization depend entirely on engineering skill. Ten years ago I said that the change which had been made, absorbing the engineer corps into the line of the navy and making every officer an engineer, was a tremendous step forward, *provided* a sincere and earnest effort was made to carry out the scheme which was thus outlined.

14 From what I have said this evening, it will be clear that

I am not as yet satisfied that this has been brought about. Undoubtedly the responsibility for the machinery of our vessels, guns, motive machinery, electrical machinery, torpedoes, etc., is upon the line-officer of the navy. He is charged with this duty by law. If the older officers of the navy had taken hold of this matter with enthusiasm, I believe that it would now have been settled and there would be no question as to the great success of the new officer, the line-officer of the twentieth century. I am not willing to believe (and indeed hardly willing to consider) the possibility that naval officers will neglect any duty with which they are charged, and I still hope that the scheme will be worked out to a great success.

15 Not much is ordinarily said about the machinists who are doing good work on board our vessels. They look after the routine work of repair and adjustment on board ship, but they are without the scientific training which is required for engineers who are really qualified for the duties comprehended by that title. If the line-officers of the navy do not maintain engineering efficiency, it will then, as the organization now stands, fall upon these machinists to perform the work of the engineers. In other words, in an organization whose efficiency is absolutely dependent upon the skill of engineers, the men relied upon for such work would be relegated to a position of inferiority so low that they are hardly counted. This is utterly un-American and can only be matched by absolute monarchies or countries as unprogressive in the mechanic arts as Spain. The speedy destruction of her navy in the war of 1898 was due to her utter incompetence even more in engineering than in gunnery. It is inconceivable that self-respecting men in a free country like ours will attempt to perform work of such vital importance without adequate recognition in the way of rank and position.

16 I will not permit myself to believe that we shall have to consider this as a practical question because I cannot conceive that naval officers would fail in their duty, but I feel that both sides of the question, so vital to our naval efficiency, should be presented.

17 During my entire career in the navy, it was my constant endeavor to show by my work the importance of the engineer and to encourage that spirit in my subordinates. I trust I will not be accused of vanity if I say that I believe my record as engineer-in-chief added a little to the reputation of engineering. My active work in the navy is done, but so long as I live my interest will never slacken and my voice will always be raised to encourage efficiency in every branch of the service.

PRESENTATION OF PORTRAIT

By Walter M. McFarland, Mem. Am. Soc. M. E.

The honor of being invited to pay a tribute to my dear old Chief, Admiral Melville, is one which I appreciate highly, as well as the allied one of acting as spokesman for the donors of the splendid portrait which is to be presented to the National Gallery this evening. I admire the Admiral as the fine flower of a splendid type of manhood, and his kindness to me for many years has been so like a father's that with a son's affection I rejoice at this splendid testimonial to his personality and his work.

2 Too often the pathway to greatness and fame is marked by the wreckage of competitors, and even friends, who have been ruthlessly thrust aside in the egoism of selfish ambition. Then, there may be a grudging admission of ability, but there is no love, no true admiration. When, on the other hand, the hero has always been the helper and friend of his companions, when he has cheerfully acknowledged their aid to his success, we have such greatness as we are celebrating tonight. Then, every member of the profession feels that the fame of the leader is reflected on the whole body, and they love the man while they rejoice in his reputation.

3 George Wallace Melville is such a man. He has been one of the famous men of engineering so long that we find it hard to remember a time when his name was not synonymous, as it is now, with all that represents progress and achievement in our profession.

4 It is a matter of delight to all of us who love him that the artist, in the picture which is to be presented to the National Gallery this evening, has faithfully depicted the chief characteristics which have made him great. These are, in my judgment, indomitable courage and unbending honesty. It is possible for a man to have great mental ability and yet fail of true greatness if he lack these essentials.

5 You all know Melville's Arctic record, which first brought him an international reputation; there he displayed a heroic courage which has never been surpassed, and for which Congress advanced him a grade in the Navy. This, however, was only one instance of the absolute fearlessness that began with his earliest days in the service. When he became Engineer-in-Chief, the same courage, but rather on the moral than the physical side, was shown. Beginning with his first annual report, he

spoke out fearlessly, setting forth the truth as he saw it and striving always for advancement and efficiency. Complaint was made to President Cleveland of the plain speech in this first report, but that strong man read it himself and said, "We want more such men."

6 His professional courage is also remarkable, and, moreover, a faculty, I believe which is characteristic of all great men, having once made his decision he does not worry about the result. Able men of minor rank are always fearful that something may go wrong and their reputation be injured. The really big man does not believe himself infallible. He knows that all men who *do* things will make some mistakes, and he is strong enough not to dread them. A notable instance of this kind in Melville's career was his use of triple screws for the Columbia and Minneapolis. I saw letters from some of his friends, for whose professional opinion he had the highest regard, urging him not to make the experiment. but he had studied the problem carefully, was satisfied with the correctness of the solution, and persevered. The result was perhaps the greatest triumph of his professional career.

7 His ability as an executive is of a very high order. The feature of deciding a case and then refraining from worry is an evidence. He had a rare talent for choosing able assistants, and having proved them he left in their hands all the detail work. thereby giving himself time for careful study of the larger problems, The effect of this was very marked in stimulating the entire staff to the highest efficiency and zeal. I have known them all personally, and every man counted it a pleasure to work, without regard to hours, for the credit of the "Chief" and the glory of the Service.

8 With respect to his professional work, it is notable that his career as Engineer-in-Chief of the Navy, from 1887 to 1903, is the longest on record. It covers the building of the "new navy," and the Spanish war. During this time he was responsible for new designs of machinery for about 120 vessels, among which were 24 battleships and 41 armored vessels. Best of all, there were no "lame ducks," and no failures.

9 I will mention briefly some details of his more important work. He was the first to use water-tube boilers in large war vessels and to determine the actual coal consumption by trials. He was also the first to use the method of determining trial-speeds, known as the "standardized screw," which is the simplest, most accurate and inexpensive, and fairest to the contractor as well as to the government.

10 It is to him also that we owe our first high-speed battleship. When in 1898 *the proposals for the* Maine, Missouri and Ohio

were being prepared, he stood alone in his demand for 18-knot ships. If he had not persisted, we should have been three years longer behind the other navies of the world in battleship speed.

11 It is very interesting to note also that only a little after this he proposed an "all-gun one-caliber ship," in other words, what is now called the "Dreadnought" type. Before I left the Service, I had often heard him talk of this big ship with ten twelve-inch or twelve ten-inch rifles and nineteen or twenty knots speed; and about 1899 he submitted a sketch plan of the battery of such a ship to the Board on Construction. Possibly the same influence which almost prevented the eighteen-knot battleships prevented consideration of this more advanced type. At all events Melville was in advance of the general naval mind, and our country lost the credit which it might have had for the introduction of this revolutionary improvement several years before the Dreadnought was produced.

12 During the war with Spain he brought out the repair ship and the distilling ship. The idea of the former was not new, but the Vulcan was by far the most complete vessel of the kind equipped up to that time. The latter furnished fresh feed-water to the boilers and enabled a vessel with a storage bunker capacity of 3000 tons to supply 60,000 tons of water.

13 A clever piece of work at this time was the fitting of new boilers to some of the old Civil War Monitors to enable them to be used for harbor defense. For years Melville had advised the Navy Department that new boilers must be supplied before these vessels could be used. When the destruction of the Maine made the outbreak of hostilities seem probable, the makers of water-tube boilers submitted estimates of time and cost for the work. Boilers were promised in 30 days, but it was necessary to use the standard land type. As these vessels were not to go to sea, however, this was satisfactory. The worn-out boilers were cut up and passed out through the smokepipe (because the armored deck could not be taken up; the new boilers were passed down the smokepipe in sections and erected on board; finally each of the boats was given a steam trial, which was entirely successful.

14 A great deal of experimental work was done under his direction, all of which is published in his annual reports. The last of such experiments was a series of tests of oil as fuel, probably the most comprehensive ever made.

15 My brief sketch of this famous man would be incomplete if I failed to speak of his personality. The lion-like head and the frank speech have led some to say that he is one of the old "Vikings,"

spared to us a thousand years after the others have gone; but if this leads any to think that he is harsh and cold, there could be no greater mistake. Like all strong natures, he is pronounced in his feelings, but he is a man of warm affection, and when he has once taken you into his heart, you are sure of an abiding-place there as long as you are worthy. It is often said that no man is great to his intimates, but I have been with him, day by day, for years; have seen him under all conditions; and my admiration and love for him have simply increased as the years go by. I have no ambition to be a Boswell and I have not kept notes of his doings; but I have seen the daily workings of a great, kind heart, tender for the humble yet fearless toward the great; and I can truly say that I count it a privilege and an inspiration to have been a trusted friend and helper of this noble man, who has exemplified the highest type of manhood and added new luster to the profession of engineering.

ACCEPTANCE OF PORTRAIT

By Dr. C. D. Walcott

It gives me pleasure, speaking for the Smithsonian Institution as the custodian of the National Collection of Art, to accept from you for the people of this country this fine portrait of Rear-Admiral George Wallace Melville, to be exhibited in the gallery of portraits of Americans who have achieved eminence in their life work.

2 Among the men who have rendered distinguished service to their country in literature, science, or art, in war or in peace, in professional or civil life—few have won such well-merited distinction in so many lines of duty as Admiral Melville. He stands high in the regard of the Nation as a naval hero, as an engineer of exceptional ability, and as a wise and resourceful administrator and advisor. It is only to be regretted that, under the operation of law governing retirement, Admiral Melville was obliged to retire from active duty in 1903, but it is to be hoped that the country which he has so efficiently and actively served may long be permitted to enjoy the benefits of his counsel.

3 The portrait of Admiral Melville is a most appropriate addition to this National collection and it is peculiarly fitting that his services should be emphasized in this happy manner by a Society which embraces so distinguished an array of men in the engineering profession, a Society that for nearly thirty years has exercised a powerful influence toward unity of interest and harmony of purpose in the broad field of American engineering.

No. 1242.

SMALL STEAM TURBINES

By George A. Orrok, New York.

Member of the Society

The papers upon steam turbines which have been presented before the Society have dealt mainly with the larger types of apparatus and have been written to show the reliability, efficiency and general desirability of this type of prime mover.

2 This paper treats of the smaller sizes of steam turbines from the standpoint of the designing and operating engineer, describing the commercial machines in sufficient detail, with reference to the service to which they have been applied, and giving certain facts concerning their operation which may be of advantage to the engineering profession. Curves of steam consumption are given which show in a general way what may be expected of these machines under certain conditions.

3 At the present time seven machines are on the market and can be obtained in various sizes from 10 h.p. to 300 h.p. with reasonable deliveries. These are the De Laval, Terry, Sturtevant, Bliss, Dake, Curtis and Kerr turbines. Three other machines are nearly at this stage of development and patents have been applied for on several others.

4 Many thousand horsepower of these turbines have been sold and are in successful commercial service. The following reports of total sales of sizes from 10 h.p. to 300 h.p. have been obtained from the manufacturers:

For further discussion of Steam Turbines, consult Transactions as follows: Vol. 10, p. 680, Notes on Steam Turbines, J. B. Webb; vol. 17, p. 81, Steam Turbines, W. F. M. Goss; vol. 22, p. 170, Steam Turbines, R. H. Thurston; vol. 24, p. 999, Steam Turbines from the Operating Standpoint, F. A. Waldron; vol. 25, p. 1056, The De Laval Steam Turbine, E. S. Lea and E. Meden; vol. 25, p. 1041, The Steam Turbine in Modern Engineering, W. L. R. Emmet; vol. 25, p. 782, Different Applications of Steam Turbines, A. Rateau; vol. 25., p. 716, Some Theoretical and Practical Considerations in Steam Turbine Work, Francis Hodgkinson.

Presented at the Spring Meeting, Washington, May 1909, of The American Society of Mechanical Engineers.

De Laval,	De Laval Steam Turbine Company................	70,000 h.p.
Curtis,	General Electric Company........................	70,000 h.p.
Terry,	Terry Steam Turbine Company..................	15,000 h.p.
Kerr,	Kerr Turbine Company.........................	10,000 h.p.
Sturtevant,	B. F. Sturtevant Company......................	
Bliss,	E. W. Bliss Company...........................	
Dake,	Dake-American Steam Turbine Co...............	

5 All of these machines are of the impulse type: that is to say, the steam is expanded in a nozzle and the kinetic energy of the jet is absorbed by passing one or more times through the buckets of the

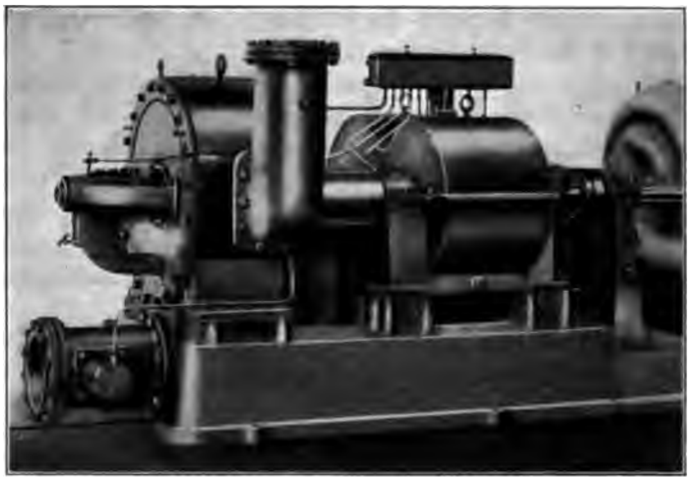


FIG. 1 HIGH AND LOW-PRESSURE DE LAVAL TURBINE

turbine rotor. In the De Laval turbine only one moving element and one steam pass are used, which necessitates a very high bucket velocity. In the Terry, Sturtevant. Bliss and Dake turbines a series of return passages is provided. The steam returns two or more times to the same rotor and the bucket speed is much lower. In the Kerr turbine the steam is used in stages with one bucket wheel in a stage; while in most of the Curtis machines two or three stages are used with two or three rows of moving buckets, separated by stationary guide blades, in each stage. Compound machines of the other types have been made but are not as yet produced commercially.

6 By far the larger number of these machines is used in connection with extra high-speed electric generators, the next larger application

being to centrifugal fans for high pressures. Centrifugal pumps adapted to high rotative speeds have been rather generally introduced in the last few years and it is becoming usual to connect small turbines direct to these machines. The small space required and the simplicity obtainable in a 100-h.p. turbine at speeds of from 800 to 1200 r.p.m. have been important factors in their introduction.

7 The first of the small turbines to be put on the market was the De Laval, made by the De Laval Steam Turbine Company of Trenton, N. J., and introduced in this country about 1896. This machine is of the

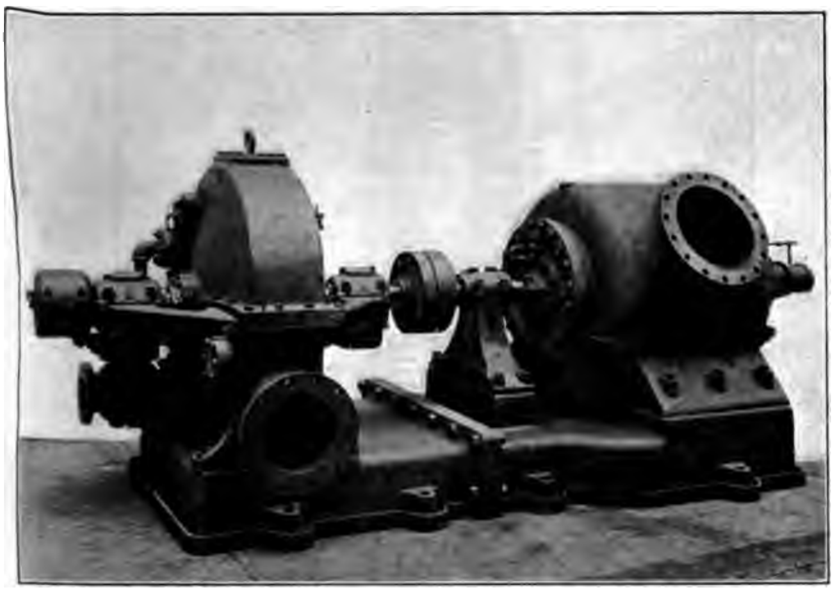


FIG. 2 TERRY STEAM TURBINE, 36-IN.

pure impulse type, the steam being expanded in the nozzle down to the exhaust pressure, and the resultant velocity transferred to the wheel in one steam pass. The bucket speed is high, ranging from 600 to 1300 ft. per sec. Eight sizes of wheels are made, generating from 10 h.p. to 500 h.p., with one nozzle in the smallest size and eight or more in the 500-h.p. size.

8 The high bucket speed necessitates the use of gears of special construction, which have been very successful. The design, construction and economy of this type have been discussed in vol. 25 of Transactions, p. 1056.

9 The Terry turbine, made by the Terry Steam Turbine Company of Hartford, Conn., has been manufactured for about ten years, although the commercial machine has been on the market only for about four years. This turbine is of the impulse type, but the steam passes through the buckets a number of times before its energy is absorbed. The case of the machine is parted on a horizontal plane through the shaft and at right angles to the wheel. The nozzles and

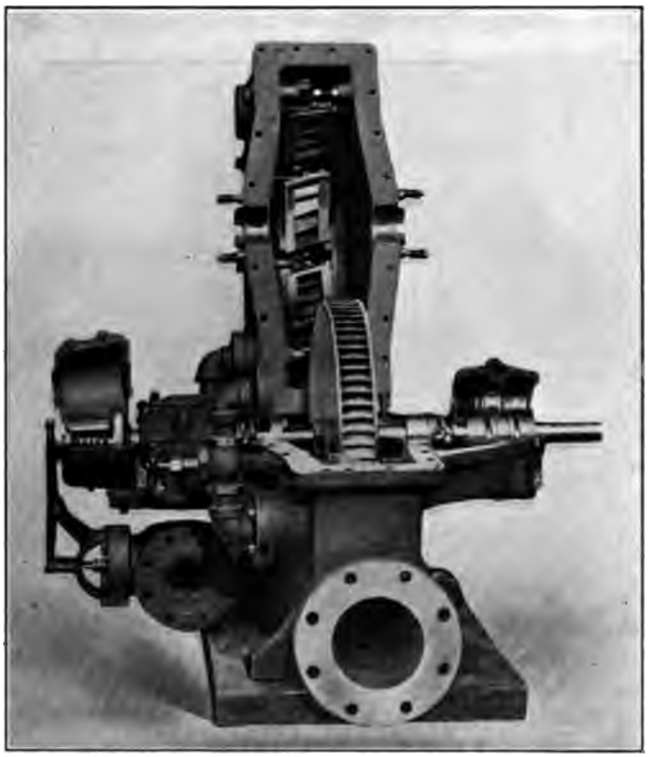


FIG. 3 TERRY TURBINE SHOWING CONSTRUCTION

return passes are bolted to the inside of both parts of the casing. The nozzles are in the plane of the side of the wheel. The return passages are of brass and are separated by partitions. The wheel itself is built up of two steel discs held together by bolts over a steel center. The buckets are built of steel punchings, fitting in grooves cut in the discs, as shown by the figures. The sizes of wheels manufactured at the present time are 12, 18, 24, 36 and 48 in., and the number of nozzles varies from two on the 12-in. wheel to eight or ten on the 48-in. wheel.

10 The Sturtevant turbine, made by the B. F. Sturtevant Company of Hyde Park, Mass., has been in the development stage for three or four years and quite a number of machines have been sold. The present type of turbine may be called "standard," however, and four sizes of wheel are built, 20, 25, 30 and 36-in., developing from

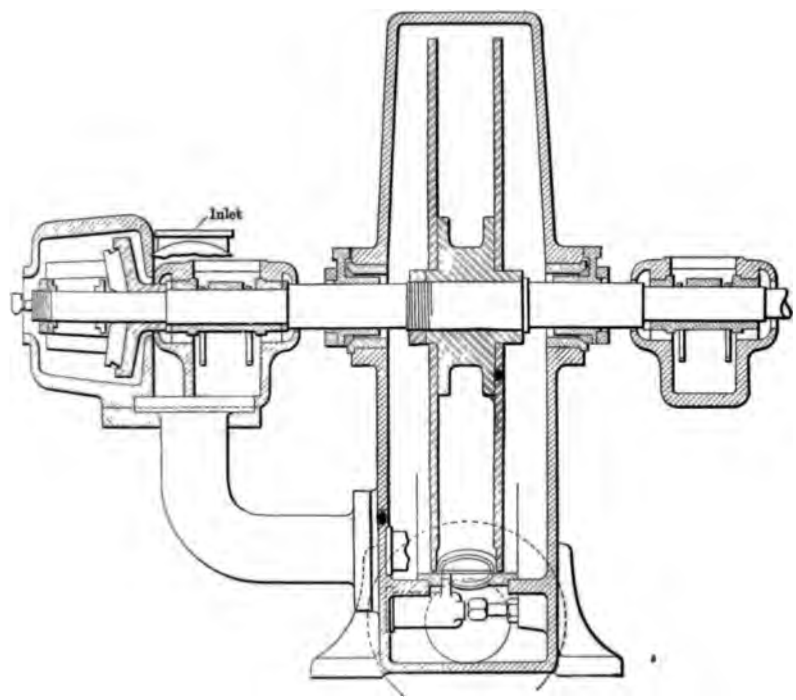


Fig. 4 Sectional View of Terry Turbine

3 h.p. to 300 h.p. The turbine is of the multiple-pass type similar to the Riedler-Stumpf. The casing is cast solid with one end. The nozzle and return chamber ring are inserted from one side and the wheel is milled from the solid. The return passages are from eight to twelve in number and are milled on the inside of the return chamber ring. They are partitioned and are similar in shape to the buckets. The nozzle lies in the plane of the side of the wheel.

11 The Bliss turbine, formerly known as the American, made by the E. W. Bliss Company of Brooklyn, N. Y., is of the same type as

FIG. 5 WHEEL AND CASING OF STURTEVANT TURBINE

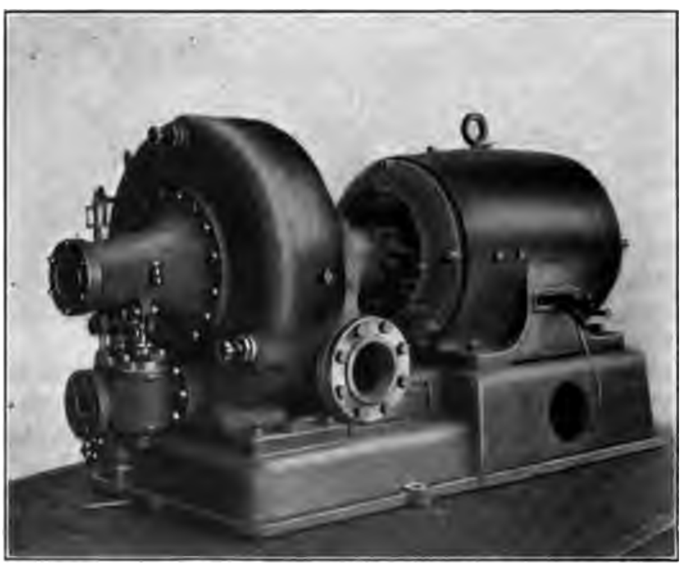


FIG. 6 STURTEVANT STEAM TURBINE, 30-IN.

the Terry and Sturtevant and has been on the market only a few months. The casing and steam chamber are cast solid with one side and the nozzle and return chambers bolted in. The wheel is milled from a steel casting, or forging in the smaller sizes, and the partitions

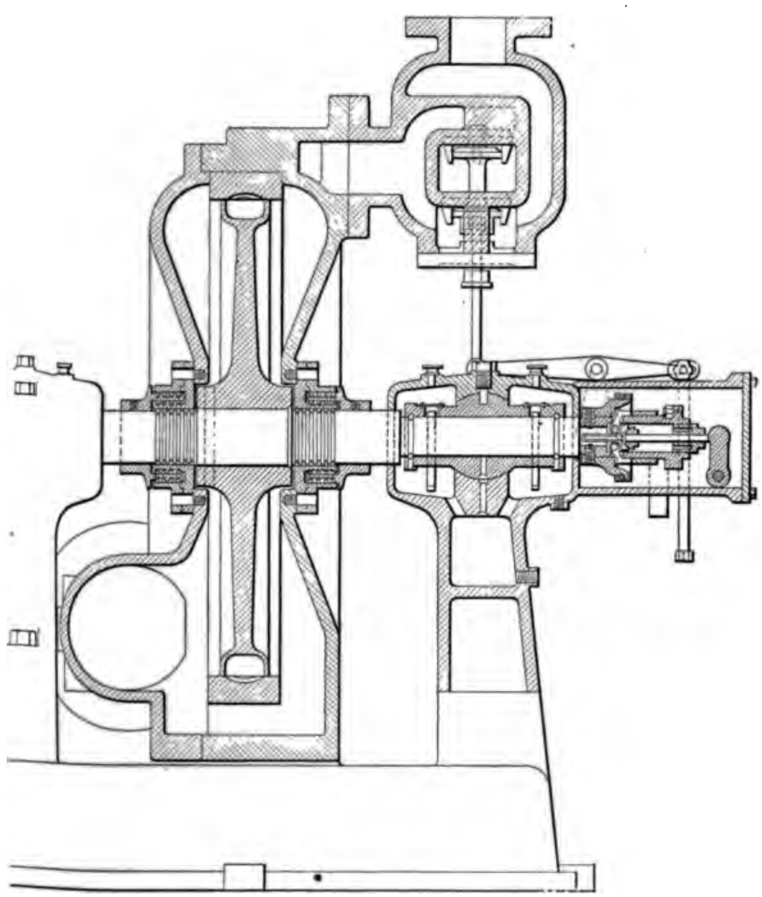


FIG. 7 SECTION OF STURTEVANT TURBINE

separating the buckets are inserted and held in place by three bands of steel shrunk on the face of the wheel. The return passages are peculiar in having no partitions. Two sizes of wheel have been built, the 42-in. and 30-in., but designs have been developed for the 12, 18, 24, 36, 48 and 60-in., covering powers from 10 h.p. to above 600 h.p.

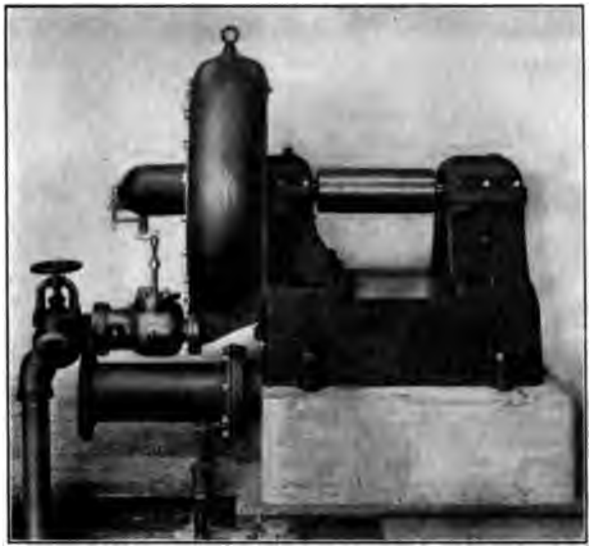


FIG. 8 BLISS TURBINE, 30-IN.

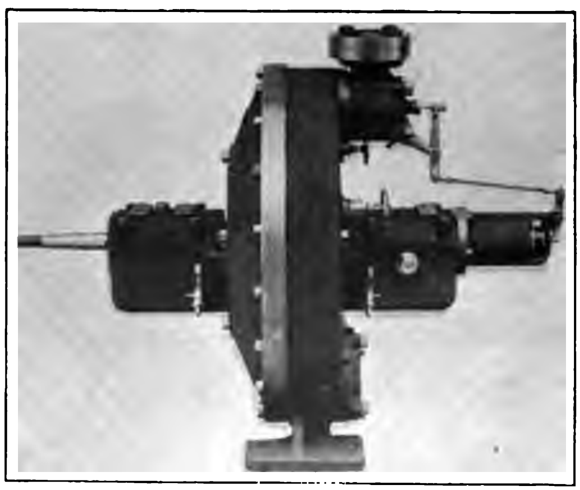


FIG. 9 DAKE STEAM TURBINE, 24-IN.

12 The Dake turbine, made by the Dake-American Steam Turbine Company of Grand Rapids, Mich., is a single-stage impulse turbine. The wheel is made of two bucket discs, with milled buckets and inserted partitions, bolted together over a wheel center. In their Headlight turbine the governor is enclosed between the sides of the wheel. The nozzles and return-passages are placed between the bucket discs. The machine is built in sizes of from 5 h.p. to 100 h.p., the diameter of the smallest wheel being 12 in.

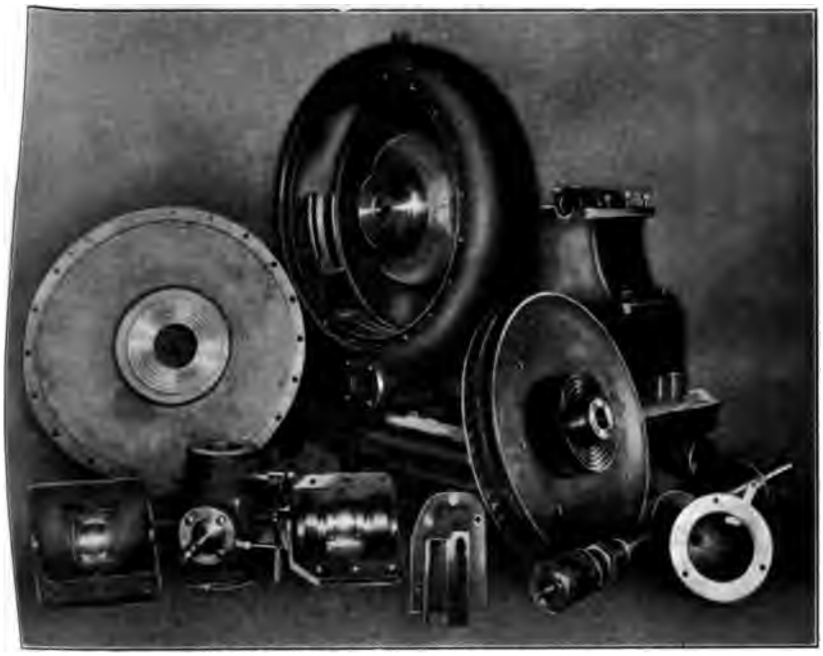


FIG. 10 PARTS OF THE BLISS TURBINE

13 Coincident with the development of the large Curtis turbines, the General Electric Company, at their Lynn Works, have developed and placed on the market a line of small generating sets ranging from 5 kw. to 300 kw. This range is covered by eight sizes, the smaller machines being single-stage with two or three passes per stage. The buckets and nozzles are of the well-known Curtis type.

14 The Kerr Turbine, made by the Kerr Turbine Company of Wellsville, N. Y., is of the compound impulse type. It is generally built in from two to eight stages. The buckets are of the double

Pelton type, inserted like saw teeth in the wheel disc. Five sizes of of wheels, 12, 18. 24, 30 and 36-in., are made and cover a range of from 10 h.p. to 300 h.p. The nozzles are in the plane of revolution of the

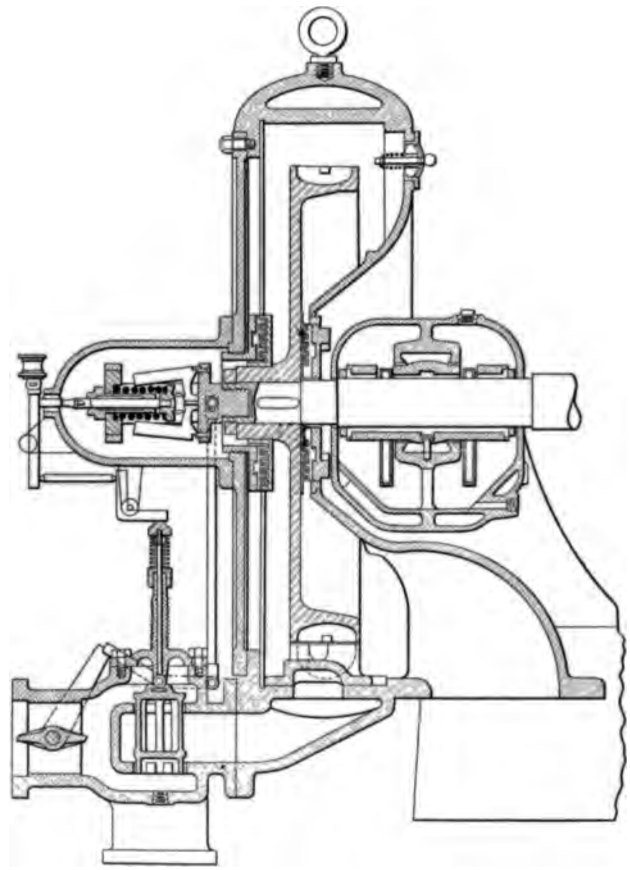


FIG. 11 SECTIONAL VIEW OF BLISS TURBINE

wheel and are screwed into the stage partitions and held in place by a lock nut.

15 As in large turbines, details of these small turbines, to which reference has been made, show the skill and knowledge of the designer, and that the same problem may be solved in different ways is well illustrated by the sections here reproduced.

DESCRIPTION OF DETAILS

16 *Nozzles.* The diverging nozzle is used by all makers except Kerr, whose multi-stage wheel requires a converging nozzle. In the De Laval, Sturtevant and Kerr turbines, the nozzles are screwed into their seats; that of the Terry is held in place by a bolt. The nozzles

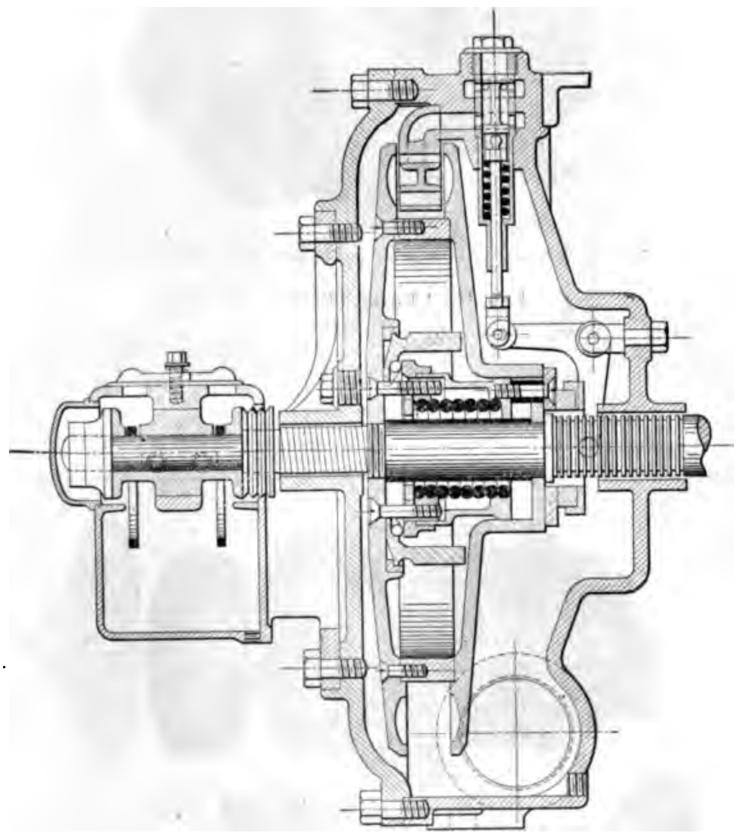


FIG. 12 SECTION OF DAKE HEADLIGHT TURBINE; EXTERIOR SHOWN IN FIG. 9

of the Curtis, Dake and Bliss turbines are reamed out of the solid. The larger sizes of the De Laval machine which have been put on the market lately have a large number of reamed nozzles instead of the older construction.

17 *Buckets.* The constructions employed in the Curtis and De Laval wheels are well known and have been described many times.

Fig. 13 Curtis Turbine 50 h.p.

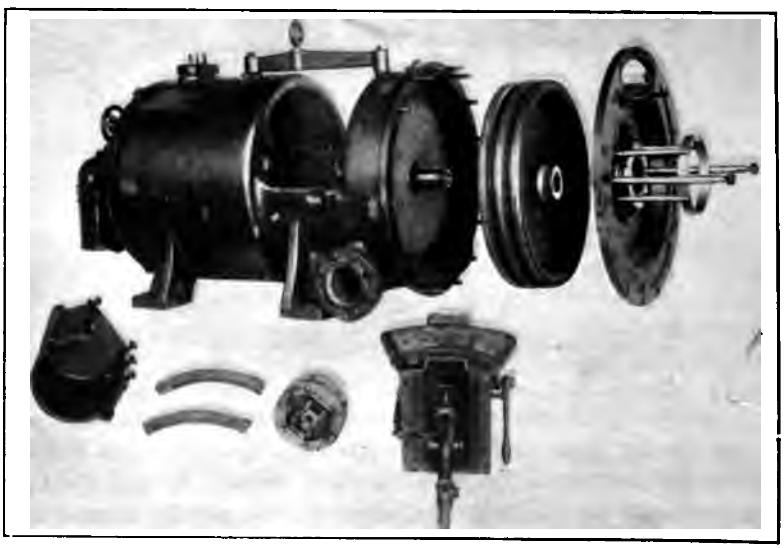


Fig. 14 Curtis Turbine in Process of Assembly

The Terry, Dake, Bliss and Sturtevant buckets are practically semicircular in form. The Terry bucket is constructed entirely of steel punchings assembled between grooves in the two steel discs forming the sides of the wheel. The Sturtevant wheel is milled out of a steel casting. The Bliss buckets are milled out, but the partitions are

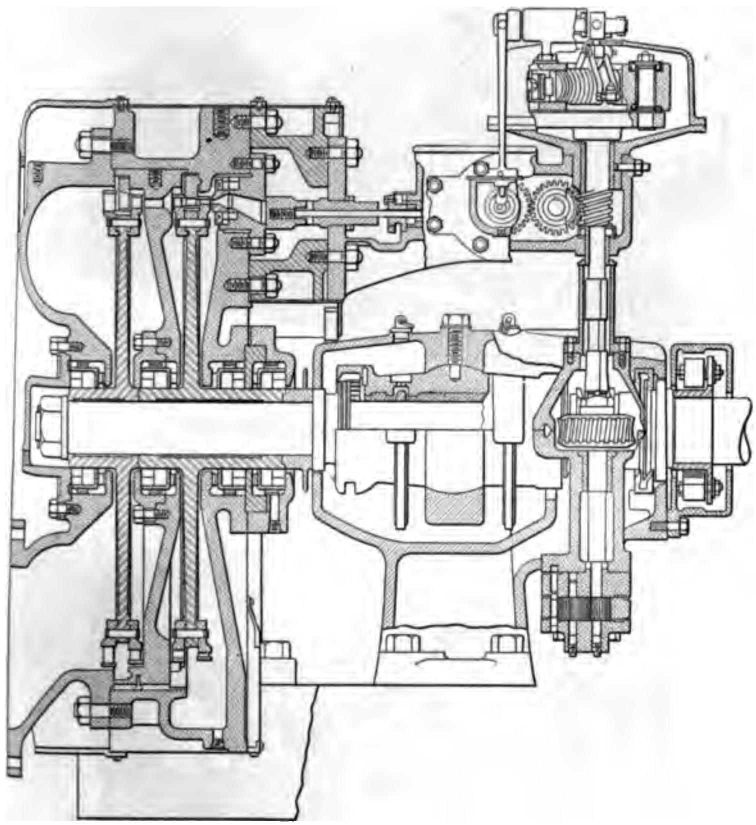


FIG. 15 SECTION OF CURTIS TWO-STAGE, NON-CONDENSING TURBINE, 160 H.P.

inserted and held in place and steel rings are shrunk on. The Dake buckets are turned out of the solid, the recesses for the partitions milled out and the partitions inserted; the wheel is then bolted together. The Kerr buckets are very similar to the original Pelton buckets and are inserted in the wheel in a manner similar to the De Laval buckets.

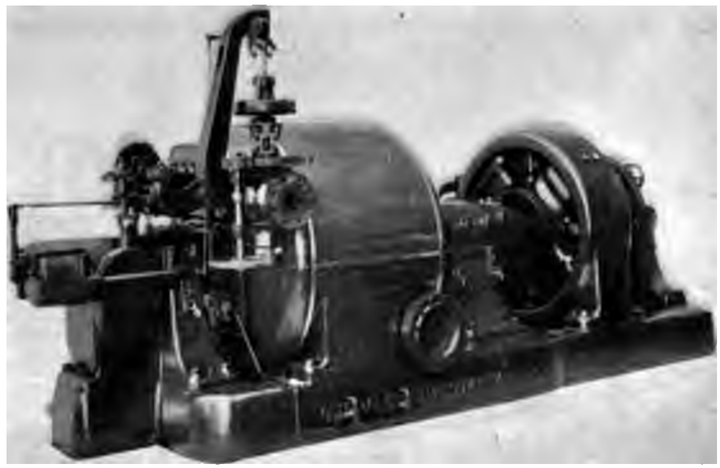


FIG. 16 CURTIS TURBINE, 200 H.P.

FIG. 17 REVOLVING ELEMENT OF CURTIS TURBINE IN BEARINGS

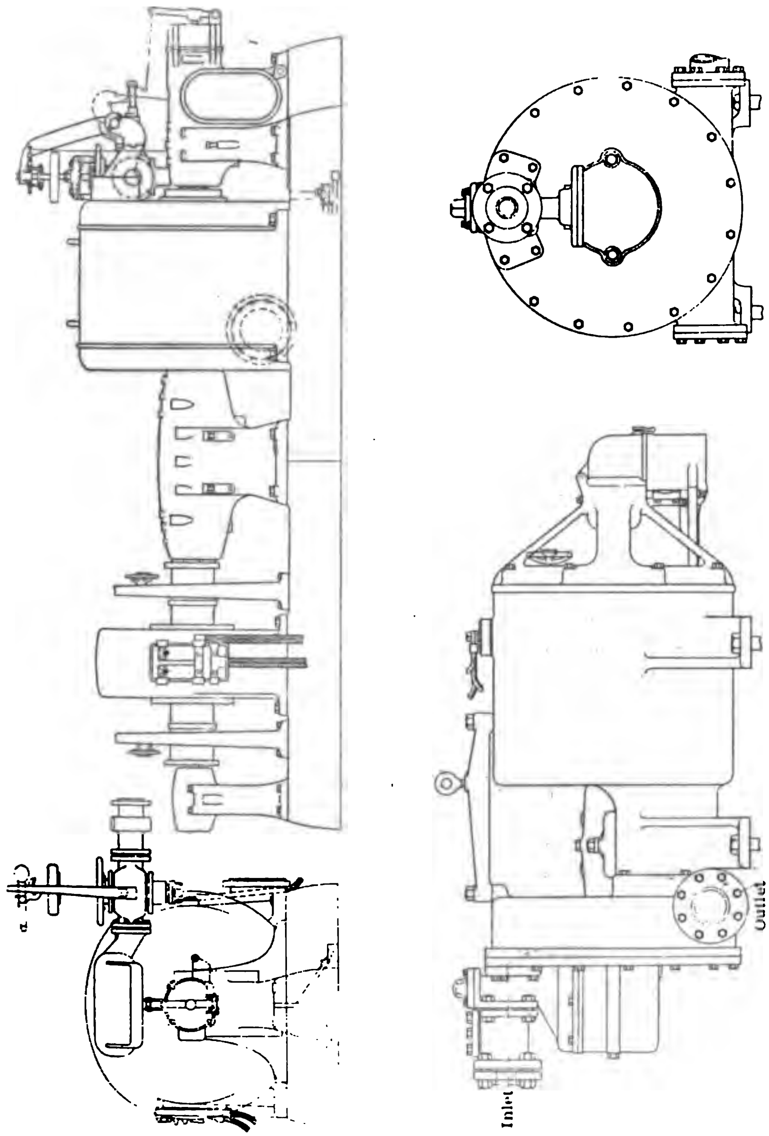


FIG. 18 DIAGRAMS OF CURTIS TURBINES, 200 H.P. AND 50 HP.

FIG. 19 KERR TURBINE, 18-IN.

18 *Return Chambers.* The Sturtevant returns are milled out of the solid ring. Bliss casts them in the nozzle piece and finishes them by hand; Terry casts each one separately, finishes by hand and assembles with bolts; Dake casts the return chambers solid, mills the passages and covers them with a shrouding.

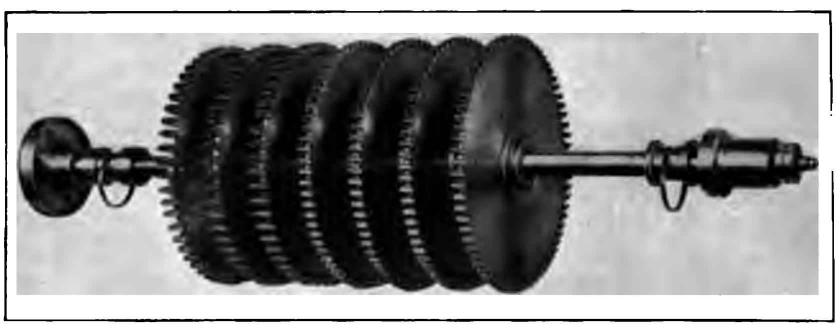


FIG. 20 COMPLETE ROTATING PART, 18-IN., 7-STAGE, KERR STEAM TURBINE

19 *Wheel Centers.* De Laval, Curtis, Sturtevant and Bliss make the wheel centers of steel castings or forgings integral with the wheel. Terry uses a steel casting but bolts the wheel disc to it. Kerr uses a screwed coupling, the inner part cut in three pieces and keyed to

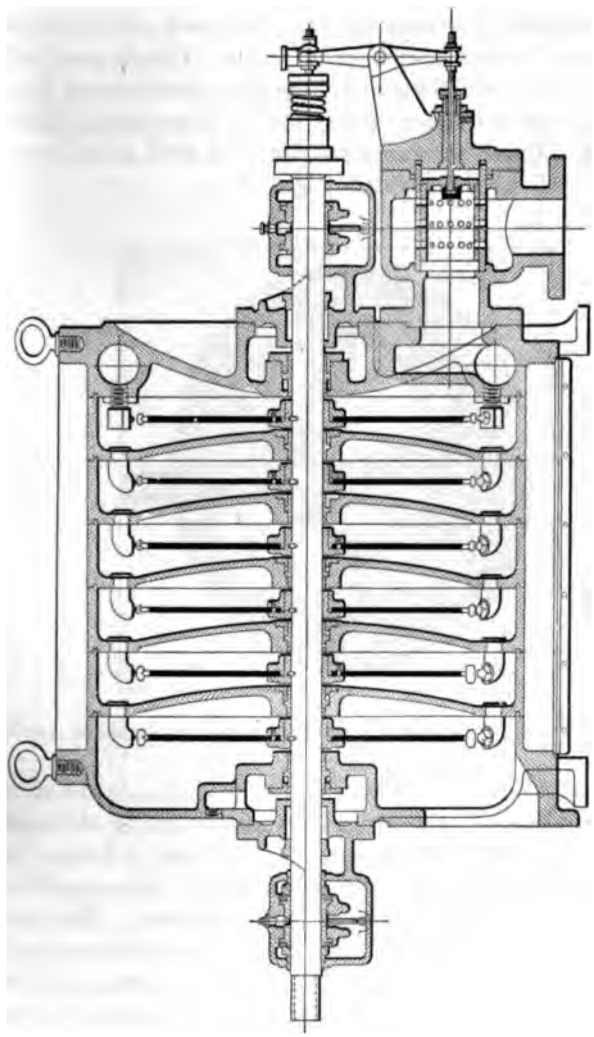


FIG. 21 KERR TURBINE, SECTIONAL VIEW

the shaft with round keys, clamping the wheel disc. Dake's wheel centers are an integral part of the wheel in small sizes, but in the larger machines are steel castings, in some cases a part of the shaft.

20 *Governors.* De Laval, Terry, Sturtevant, Bliss, Dake and Kerr use a flyball governor on the shaft end, which actuates the throttle valve through a system of levers. Curtis uses the flyball governor on the shaft for small sizes and slower-speed spring-controlled governors of different forms for the larger sizes. The Sturtevant, Bliss and Curtis machines are provided with an emergency stop governor as well as the throttling governor.

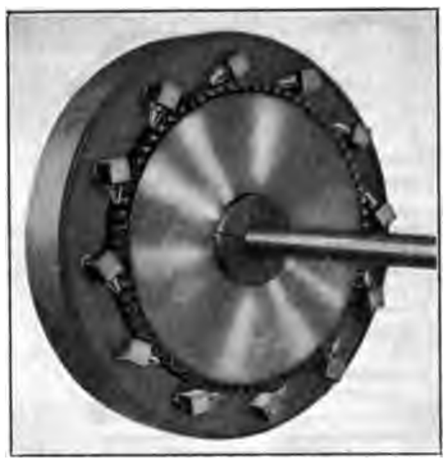


FIG. 22 ONE STAGE OF KERR TURBINE, SHOWING NOZZLES AND WHEEL

21 *Glands.* For non-condensing machines glands are not troublesome, as the difference of pressure between the casing and atmosphere is rarely more than a few pounds. Terry uses a bronze ball-and-socket gland with a long loose fit on the shaft. Sturtevant and Dake use a set of ring packing, either cast-iron or bronze. Bliss has a labyrinth packing without contact. Kerr has a floating bronze bush with soft packing behind it. Curtis uses a metallic packing held in place by a gland ring, and for condensing service a carbon-ring packing, steam-sealed.

22 *Clearance.* In none of these machines is clearance an important factor. The clearance between buckets and guide passages on a 24-in. wheel is usually from $\frac{1}{16}$ in. to $\frac{3}{32}$ in. when hot. Striking or rubbing is practically unknown.

23 ***Thrust.*** Theoretically, there should be no thrust in any turbine of these types. Practically, there is always a very small

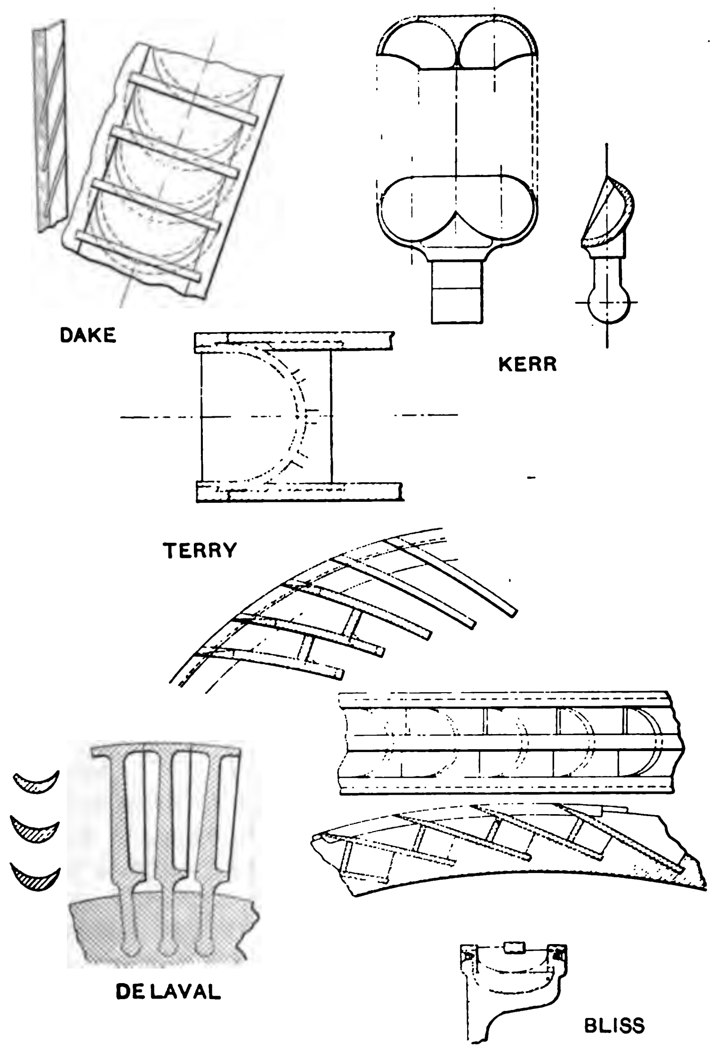


FIG. 23 TYPICAL TURBINE BUCKETS

thrust one way or the other. This thrust is usually taken care of by small thrust collars or washers next to the bearings. Thrust from the outside is prevented by the use of a flexible coupling between the turbine and the machine it drives.

24 *Bearings.* The bearings are always ring-oiled with large oil reservoirs, sometimes, on the larger sizes, provided with water jackets

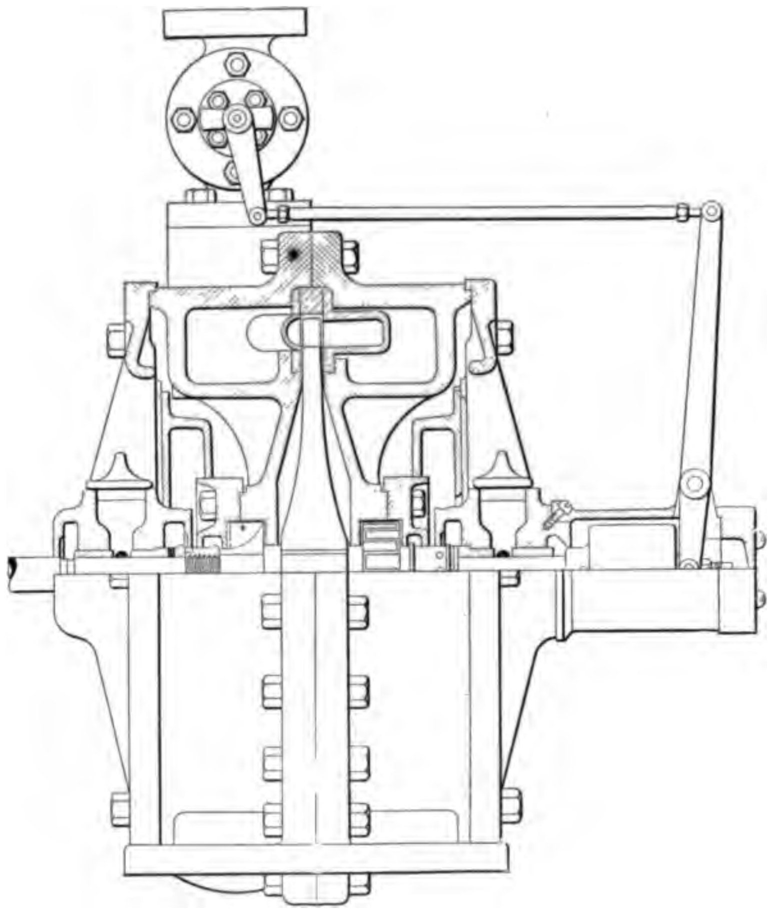


FIG. 24 SECTION OF WILKINSON STEAM TURBINE, 20-IN.

or water cooling pipes for an emergency cold-water circulation. The lubrication of the thrust is obtained at the same time.

25 *Operation.* These machines are nearly automatic in their operation. When the machine is once properly set, the coupling properly adjusted and the bearings supplied with oil, the machine may

run for years without an overhauling. The bearings must be looked after to see that no heating takes place and that the ring is carrying the oil to the shaft. The coupling should be examined from time to time to make sure that no thrust is communicated through it to the turbine. With these precautions a three months' continuous run is common and a number of turbines have to my knowledge run more than eighteen months without a cent spent on them for maintenance. Apparently there is no wear in nozzles, buckets, or return chambers

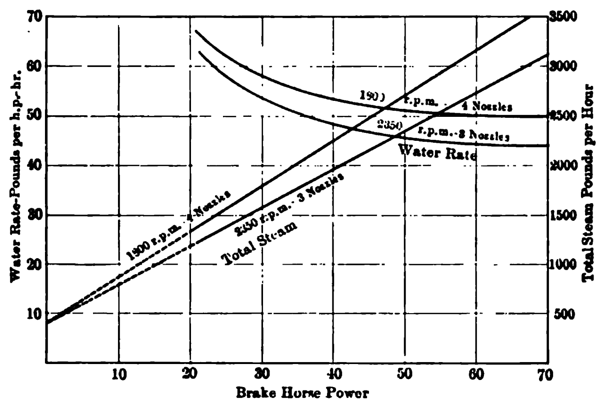


FIG. 25 STEAM CONSUMPTION CURVES, TERRY TURBINE

24-IN. WHEEL, 150-LB. PRESSURE, NO SUPERHEAT, NON-CONDENSING. TESTED BY WESTINGHOUSE MACHINE CO., PITTSBURG, PA.

The only wearing parts are the bearings and these are generously proportioned.

26 These machines may be taken apart and reassembled in half a day; some of them in two hours. The over-hung machines may be overhauled in an even shorter time.

27 *New Turbines.* The Hachenberg turbine, made by Wm. Gardam & Son, New York, is a compound impulse turbine resembling in construction the Dow turbine so frequently illustrated twenty years ago. Some experimental machines have been built, one of which was tested at Columbia University, and the commercial machine will soon be on the market.

28 James Wilkinson, of Providence, R. I., has a small steam turbine nearly in the commercial stage. A number of these machines are running, and within the next few months it is expected they will be on the market.

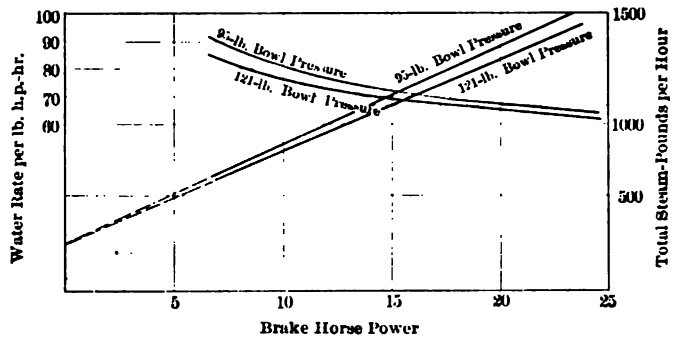


FIG. 26 STEAM CONSUMPTION CURVES, STURTEVANT TURBINE

20-IN. WHEEL, SINGLE-STAGE, NON-CONDENSING, 2400 R.P.M.

29 The Church turbine, lately completed by the Watson-Stillman Company and tested at Stevens Institute, is another promising turbine.

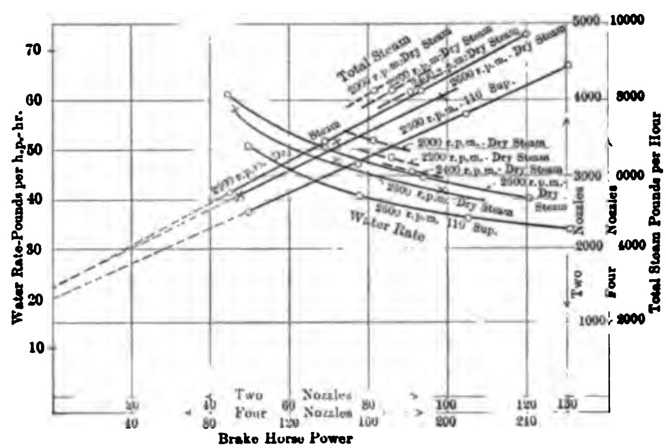


FIG. 27 STEAM CONSUMPTION CURVES, BLISS TURBINE, NON-CONDENSING

TESTED BY F. L. PRYOR AT HOBOKEN, N. J.
O = Two-nozzle, X = Four-nozzle

STEAM ECONOMY

30 The curves of steam economy have in most cases been obtained from the manufacturers. For the Curtis turbine speed-economy

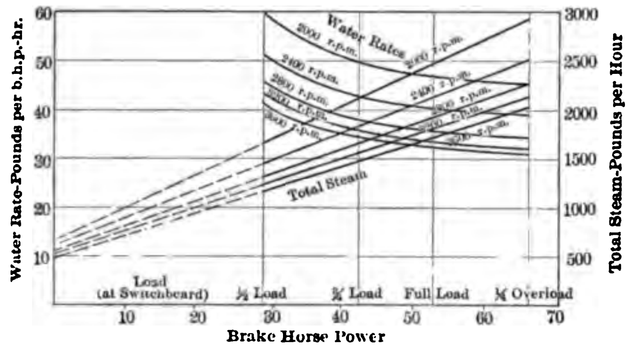


FIG. 28 STEAM CONSUMPTION CURVES, 50 H.P. CURTIS TURBINE

ONE-PRESSURE-STAGE, THREE ROWS OF BUCKETS, 25½-IN. WHEEL, CURVES CORRECTED TO 150-LB. BOILER PRESSURE, NO SUPERHEAT, ATMOSPHERIC EXHAUST

curves are given for the 50 h.p. and 200 h.p. sizes. These curves represent the average of a large number of tests and have been corrected to bring them to standard conditions. The averages were consistent, and the variation from the average in any case was not large.

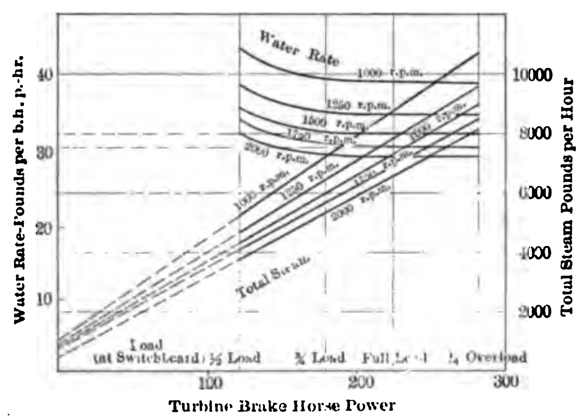


FIG. 29 STEAM CONSUMPTION CURVES, 200-H.P. CURTIS TURBINE

THREE-STAGE, 36-IN. WHEEL, CORRECTED TO 165-LB. ABS. BOILER PRESSURE, NO SUPERHEAT, NON-CONDENSING

31 The curves for the Terry turbine were plotted from fourteen tests made at East Pittsburg by the Westinghouse Machine Company. The curves for the Bliss turbine were plotted from twenty-four tests

made at Stevens Institute by Prof. F. L. Pryor. The curves for the Kerr turbine were plotted from tests made by the Kerr Turbine Company in their testing plant at Wellsville, N. Y.

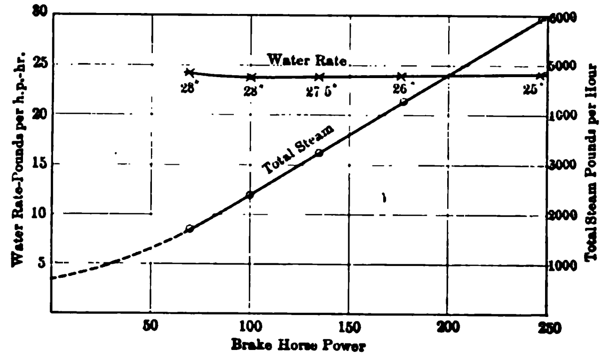


FIG. 30 STEAM CONSUMPTION CURVES, 24-IN. KERR TURBINE

SIX-STAGE, CONDENSING, VARYING VACUUM, 70-LB. GAGE PRESSURE

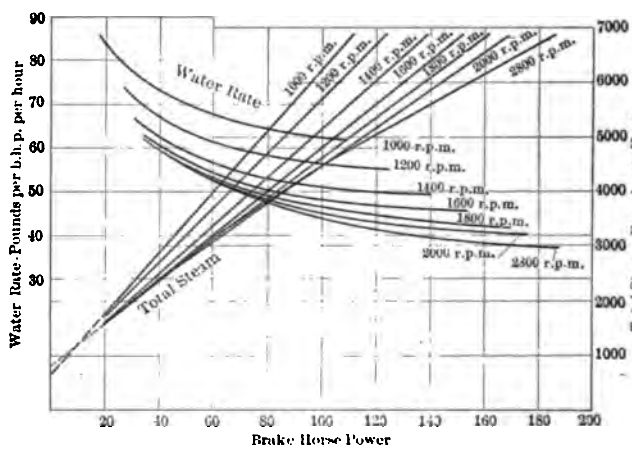


FIG. 31 LOAD CURVES OF KERR TURBINE

24-IN. WHEEL, 8-STAGE 175-LB. GAGE PRESSURE, NON-CONDENSING.

32 There seems to be no change in steam economy use. It may be too early to make this statement, but machines running regularly for three years have shown no increase in steam consumption.

33 The field of the small steam turbine is somewhat narrow when

compared with the high-speed steam engine. The small turbine has its place, however, and with the development of a more economical machine at the lower speed ranges, will have a much wider field. The turbine-driven centrifugal fan, for both high and low pressures, will have an increasing use, and the turbine-driven centrifugal pumps have marked advantages over reciprocating apparatus because of the absence of shock on the pipe line and their adaptation to space conditions.

34 The promise of development on these lines has led many manufacturers to enter the small-turbine field and the great expansion of the large-turbine business without doubt presages a like future for the small steam turbine.

DISCUSSION

Charles B. Rearick. Small turbines are being used extensively for hot-well service for surface condensers, the turbine driving a centrifugal pump which carries the condensed water away from the condenser, supplanting in these cases the usual reciprocating pump. They are more efficient than the reciprocating pump, they take less space, there are fewer parts to maintain, and the service seems to be very popular. They have also proved their worth in larger installations for driving boiler-feed pumps of the multistage turbine type.

2 Some of the newest work taken up by turbine drive is for circulating pumps for surface and jet condenser work. We have a representative lot of such installations using turbines from 50-h.p. to 250-h.p., operating such pumps under low heads and at speeds as low as 600 r.p.m., direct-coupled to the rotor shaft of the turbines.

3 The question of economy is not touched upon to any great extent in Mr. Orrok's paper, except to give some curves which cover only one condition of service in most cases and cannot very fairly be compared for the different makes. High economy in small turbine units is in many instances of minor importance. Reliability of service is most important of all. Under the very low speeds for

The paper on Small Steam Turbines was discussed both at the Washington meeting, May 4-7, and at the Boston meeting, June 11, 1909. The discussion is here given in abstract only, eliminating the matter presented at the Boston meeting which duplicated that given at the Washington meeting. The complete discussion was published in the September 1909 issue of The Journal.

driving circulating pumps and similar pump work the economy cannot be especially good; but in nearly all these large power plants the exhaust steam is all utilized in feed-water heaters, and approximately 80 per cent of the heat is returned to the boilers.

4 There is only one class of service in which high economy is absolutely necessary, and that is, when the unit becomes the prime mover or the main unit for a plant. In that case the turbine, both condensing and non-condensing, compares well with the engine; for such work is usually driving dynamos and other high-speed apparatus and the speed can be chosen for the best economy.

W. D. Forbes. Mr. Francis B. Stevens of Hoboken, who died a year ago in his ninety-fifth year, seeing in my shop some small Pelton water wheels, told me that in 1854 he had seen a steam turbine in a candy establishment in New York, which ran 1200 revolutions for some twelve years, with little or no attention. It drove a small fan. Mr. Stevens described the machine as practically the same as a Pelton water wheel, except that the bucket was cut in two, each half being placed on a separate disc, and the steam was led to these two buckets by a "splitter" between them, which of course was stationary. Each bucket was made fast to the disc, which was of course keyed to the shaft.

2 What seems strange to me is, if steam turbines are such excellent things and have been known so long, that they are not more generally used.

Richard H. Rice. The author describes the construction of seven different turbines, which may be divided into four classes according to the method of using steam, as follows:

a Single-stage, expanding nozzle, one bucket row, one velocity extraction: De Laval.

b Multistage, conveying nozzle, one bucket row per stage, one velocity extraction per stage: Kerr.

c Single-stage, expanding nozzle, one bucket row, multiple-velocity extraction: Terry, Sturtevant, Bliss, Dake.

d Single or multistage (depending on capacity), two to three bucket rows, mutiple velocity extraction but only one per bucket row: Curtis.

2 The value of these various methods of using steam is clearly set forth in the curves presented by the author (Fig. 25 to Fig. 30),

giving the water (steam) consumption of the various turbines in Classes *b*, *c* and *d*. In the diagram in Fig. 1 all these curves are drawn to the same scale so that they may be readily compared.

3 It will be seen that with one exception, a very small machine, which suffers somewhat from this fact, the water-rates of all the turbines in Classes *b* and *c* fall rather closely together, while Class *d*,

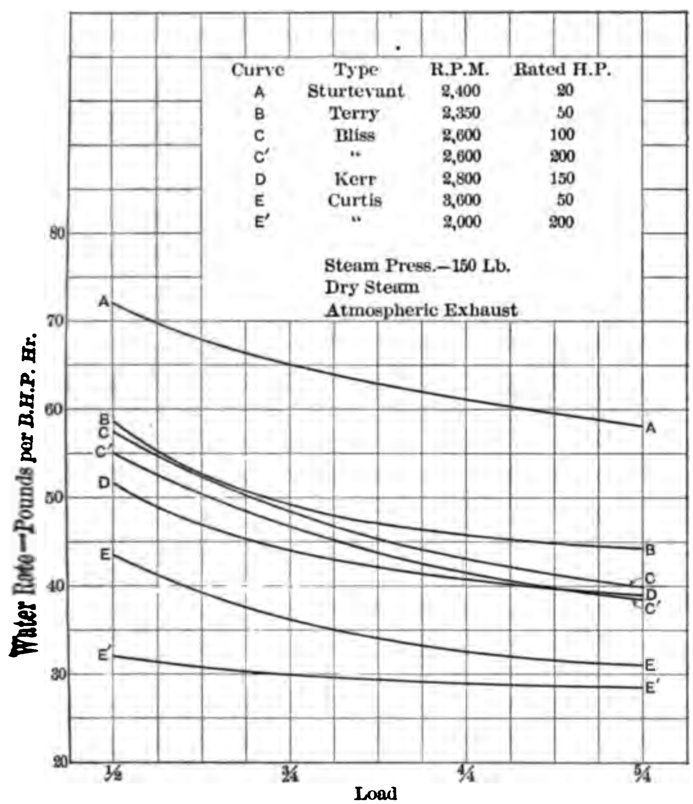


FIG. 1 ECONOMY CURVES OF SMALL TURBINES

even with a 50-h.p. machine, shows considerably better results. This is much more marked in the case of the 200-h.p. machine. Class *b* is represented by an eight-stage machine of 150-h.p. and shows slightly better results than the machines of Class *c*, particularly at light loads. This result is insignificant as compared with the complication of the large number of wheels, diaphragms, packings, and length of machine, and this complication is therefore apparently not justified. The reason such multistage machines do not give better water rates is due to a considerable extent to the high frictional losses caused by rotating wheels in a dense atmosphere of steam at comparatively high pressure.

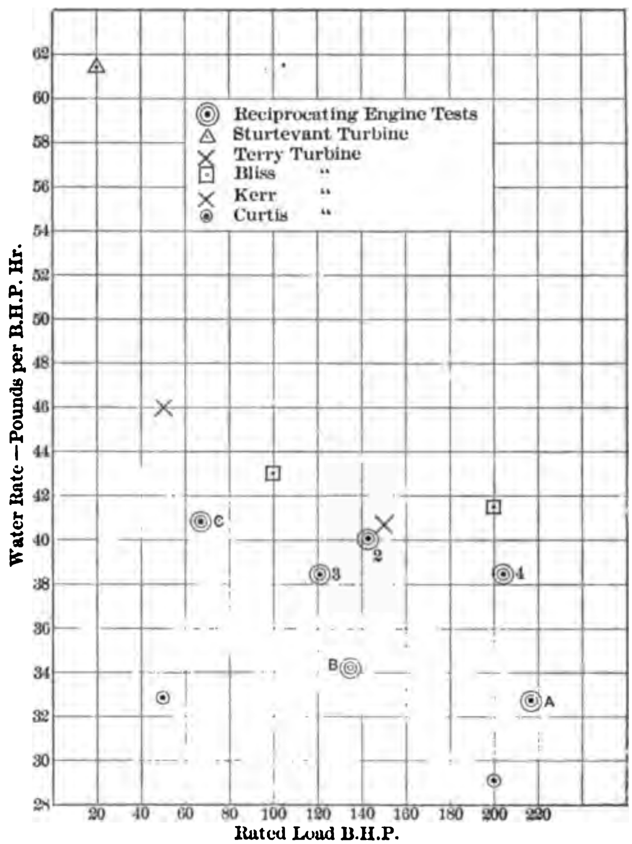


Fig. 2 Comparison of Economy of Reciprocating Engines and Turbines

4 The turbines of Class *c* labor under two disadvantages, due to using the steam repeatedly in the same buckets; either (as in the case of the Terry turbine) the steam has to be turned at high velocity through an angle of 180 deg. in the return chambers, or it has to be

used in buckets which have in general the wrong angle of entrance; for it is easy to see that if the bucket angle is correct for receiving the jet at its highest speed, it cannot be correct when the jet has considerably slowed down.

5 The general principle employed in these turbines was used first by Professors Riedler and Stumpf in the years 1902–1903, and a number of these machines were built and put into service. The company which exploited them, however, abandoned the principle about the latter year and since then has built under a Curtis license, this step having been taken by reason of the superior economies obtainable by the Curtis construction, which the paper seems fully to confirm.

6 At the Detroit meeting, June 1908, Messrs. Dean and Wood presented a paper giving results of tests on high-speed engines of sizes comparable with the turbine figures given by Mr. Orrok. Fig. 2 shows the results compared with the full-load water rates given by the author.

7 Messrs. Dean and Wood tested engines which had been in service for some time and many of which had evidently seriously deteriorated in efficiency due to wear and leakage. Mr. Orrok confirms the statement made by the writer, that the turbine does not fall off in efficiency after similar length of service and in further confirmation of this is the test on a 75-kw. Curtis turbine made by Professor Carpenter.

8 This fact has also been established by many tests made by the writer on turbines which have been in use for considerable lengths of time, and is subject only to the qualification that when steel bucket constructions are used, as in all the turbines described except the Curtis, wear may be expected under certain conditions of wet steam and light loads which will increase steam consumption after a very moderate length of service. The use of bronze buckets of the proper composition to resist this deterioration is therefore essential to secure the best results under all conditions.

Prof. R. C. Carpenter. During the past year I have given considerable study to the results from the use of small turbines, arriving at practically the conclusion of the author (Par. 33), that the field of the small turbine is somewhat narrow as compared with the high-speed steam engine. This conclusion applies to small turbines running non-condensing, however; when large turbines are operated condensing the economy is very high, and I think the results will usually be superior to those obtained with piston engines.

2 On testing one turbine, which I think had been in use for three years, I was pleased to find that my results practically agreed with those obtained when the turbine was first installed. This seems to indicate that so far as that turbine is concerned, there was no deterioration from use. The general results which I obtained in the economy tests were substantially those shown on these curves, and indicate that the economy is not good compared with the piston engine; the advantages of the small steam turbine must therefore be other than simply that of economy. The results of the tests of this machine are shown in the table.

3 I have recently had an opportunity of getting figures from a small turbine operated with a high degree of superheat and running

TEST OF CURTIS TURBINE, 75-KW. CAPACITY, OPERATED NON-CONDENSING
October 13, 1908

Test No.	1	2	3
Electric load, kilowatts	57.7	22.6	35.1
Pressure at throttle, pounds gage	121.6	95.0	93.3
Pressure at nozzle, pounds gage	108.4	86.0	82.6
Back pressure, pounds gage	0.24	0.07	−0.25
Barometer reading, inches	30.0	30.0	30.0
Total water to boiler, pounds	14971	1986.5	7454
Wet steam to turbine, pounds	14121	1833	6973
Quality of steam, per cent	98.3	99.0	99.0
Dry steam to turbine, pounds	13881	1815	6903
Dry steam to turbine per hour, pounds	3085	1815	2301
Dry steam to turbine per kw-hr., pounds	53.5	80.4	65.6
Equivalent per i. h.p. (provided 1 kw. = 1.6 i.h.p.)	33.5	50.3	41.0

NOTE—The pressure at the throttle is practically the same as at the boiler, which stood about 80 ft. away.

non-condensing. The results of that test were satisfactory in many ways: 350 deg. superheat seemed to have aboutthe same effect as 18 in. of vacuum, and a machine having a water rate given as approximately 50 lb. per b.h.p. went down to about 22 lb. per b.h.p. The small steam turbine has special advantages for many kinds of work where a high rotative speed and small torque are desirable; for those kinds of work I believe it will ultimately supersede the small piston engine.

H. Y. HADEN. A field for small turbines not touched by this excellent paper is that of installations where exhaust steam can be utilized to advantage. There is an installation in Pittsburg of a 150-h.p. turbine connected to centrifugal pumps, which operates under very unusual conditions. Primarily, it takes the exhaust of reciprocating pumps, without any regenerator, and develops the full power when exhausting into a vacuum of 25 in., but it is also capable of automatically taking high-pressure steam at 125-lb. pressure, should the supply of exhaust steam fluctuate too much or be entirely cut off. The same turbine also operates taking steam at 125-lb. pressure and exhausting into the atmosphere, and it had further to be guaranteed by the manufacturers to take high-pressure steam at 65-lb. pressure when exhausting freely. I believe the above four conditions could not be met by any reciprocating engine. Of course maximum economy will not be attained under each, but it is attained under two: that of exhaust-pressure condensing and high-pressure condensing.

2 The turbine is of the De Laval type, which is particularly adaptable for changing steam conditions, and is the only machine in which any desired plan of operation can be attained by simply changing nozzle ratios, without any change in the angle of the bucket or velocity of the pump, and without sacrificing either capacity or efficiency. I believe there is a large field for turbines operating under conditions such as the foregoing, whether connected to generators or centrifugal pumps—a field where one can depend upon a unit irrespective of the supply of exhaust steam and without a regenerator or other expensive auxiliary.

Fred. D. Herbert. Mr. Orrok did not mention tests made on the Terry turbine at the New York Edison Company's plant some years ago, in which the steam consumption is much below that shown in the Terry curve (Fig. 25). The accompanying curve and tables show the water rates of 50-h.p., 25-h.p. and 12-h.p. Terry turbines respectively.

TABLE 1 TESTS OF A 25-H.P. TERRY TURBINE

Test No.	Steam Pres.	Back Pres.	Quality or Superheat	Speed R.P.M.	Load B.H.P.	Per Cent of Rating	Total Water per Hr.	Water per H.P.-Hr.
			Degrees					
1	90	0	62.50	2500	25.52	102.0	1068	41.85
5	90	0	71.70	2500	18.60	74.5	851	45.72
2	90	0	82.42	2500	12.39	49.5	680.5	54.93
10	90	0	50.00	2100	24.29	97.2	1068	44.00
11	90	0	60.26	2100	15.816	63.4	808.25	51.09
6	90	0	45.60	1800	22.90	92.0	1077.25	47.10
7	90	0	79.86	1800	10.21	40.8	638.58	62.54
9	90	0	44.70	1800	15.59	62.4	807	51.76

TABLE 2 TESTS OF A 12-H.P. TERRY TURBINE

Test No.	Steam Pres.	Back Pres.	Quality or Superheat	Speed R.P.M.	Load B.H.P.	Per Cent of Rating	Total Water per Hr.	Water per H. P.-Hr.
			Degrees					
1	136.5	0	58	2501	13.65	113.6	577.3	42.0
2	136.8	0	15	2513	12.08	100.7	547.2	45.3
3	128.5	0	12	2490	11.70	95.6	545.2	46.6
4	130.9	0	0	2501	7.23	60.2	421.5	58.3

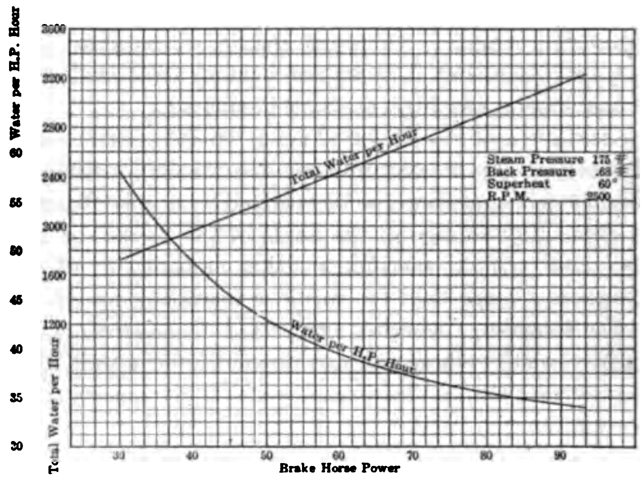


FIG. 1 CURVES OF 90 H. P. TERRY TURBINE

2 Regarding the statement that turbines are built of the single stage only, the Terry Company has in operation and under construction several two-stage condensing turbines, the largest of which is 600 h.p. and the smallest about 125-h.p. Because of the successful results obtained, the builders claim that the turbine is superior to the reciprocating engine as an operating proposition, and in none of these turbines now running has a bucket been replaced.

W. E. SNYDER. Almost all the emphasis has been laid on steam economy. Another point which should receive careful consideration is the lower cost of maintenance, particularly where turbines are used for boiler feed, replacing the direct-acting boiler-feed pumps ordinarily used; for pumping circulating water to condensers; or for driving centrifugal pumps pumping water at comparatively low heads. In all this work the turbine replaces direct-acting pumps which are very expensive to operate, not only from the standpoint of steam-consumption, but from the cost of supplies, such as cylinder oil, packing and pump valves. I have in mind one central condensing plant served by a direct-acting pump where the costs are from $75 to $100 a month for packing, cylinder oil and valves. A similar condensing plant served by a small turbine involved practically no expense for these supplies.

2 The first question, three or four years ago, before our company had installed any of these machines, was not a question of steam economy so much as of reliability of operation. If we put in a turbine to pump water for a central condensing plant, into which exhaust a large number of steam engines of various kinds used for varied service, would the turbine break down, just when it was most needed? For the purpose of investigating this point I went to a plant which had a small unit in operation driving a generator and which had been in use for about four years. The gears showed no appreciable wear, and there had been practically no shutdowns.

3 The result of that investigation was the adoption of turbines in central condensing plants in a number of works with which I have to do, and also later for boiler feed and for pumping water under low heads; and they have proved generally reliable regardless of make. This is also true in regard to small turbine air-compressor units for the cupola, etc., in steel works.

4 I think it is, therefore, in the displacement of the direct-acting pump, always expensive to operate apart from steam consumption, that the small turbine will find its greatest field of usefulness. In the Waterside Station in New York, the turbine-driven boiler feed pumps have been continuously operating for months, running almost automatically, and requiring practically no attention or supplies. Steam economies are of course important, yet in large plants where all of the large units are condensing, the steam from the auxiliaries is needed to heat the feed-water, and a few per cent more or less in steam consumption of turbine auxiliaries does not materially change the total economy of the plant.

5 Other advantages in favor of the small turbine as compared to direct-acting pumps, are the small space required and the fact that they can always be kept clean and present a good appearance. Direct-acting pumps are usually very difficult to keep in presentable condition on account of water leakage and of the excessive use of lubricants. The turbine and pump are entirely enclosed, the case can be wiped very conveniently, and it presents nothing of the unsightly appearance which is so often characteristic of the direct-acting pump.

F. H. Ball. The conclusions of the author regarding the future of the small steam turbine may fairly be questioned. On the score of efficiency the showing made by the several types, even when tested by the parties interested in their success, is very poor. From these test figures, it appears that the best performance ranges from about 30 lb. per h.p. per hr. to nearly 70 lb.

2 It must be noted also that very high steam pressure is generally used, and in some cases superheat. Under these conditions any good reciprocating engine, even of the single-valve type, run as a non-condensing compound, would easily develop power on 20 lb. or less: the user of one of the non-condensing turbines described must therefore expect to increase his coal bill from 50 to 200 per cent over a single-valve non-condensing compound engine having the simplest possible form of valve gear.

3 Moreover, the speed of these turbines, from 2000 to 3600 r.p.m., will generally be considered objectionably high. Buyers of electric motors generally prefer motors of moderate speed, even at the extra cost, and generally speeds above 1000 r.p.m., even for motors as small as 10 h.p., are considered objectionable. This same objection must inevitably be urged against speeds of 2000 to 3600 for engines of considerable power.

Chas. A. Howard. As far as a comparison of the merits of different turbines goes, it must be remembered that the economy is affected by the bucket speed even more than by steam pressure. In all of these tests the bucket speeds are different, and any attempt to make a comparison of the steam economy, as in the diagram by Mr. Rice, would thus lead only to an erroneous view. While his curves show in general what may be expected from turbines of this size, no correct comparisons can be drawn between individual machines.

W. J. A. London.[1] With reference to Richard H. Rice's comparison of Fig. 25 and Fig. 28, showing the steam consumption of the Terry turbine and that of the Curtis turbine, if the curve of the Curtis turbine be produced and the peripheral speed of the two types be made the same, the curve of the Terry turbine will cross that of the Curtis type at about 1950 r.p.m. See Fig. 1. There is therefore not much room for discussion of the difference of efficiency of the two types. Moreover, with a large turbine an increase of a pound on the steam consumption would increase the cost bill from

[1] Terry Steam Turbine Co., Hartford, Conn.

$2000 to $4000 a year, but with a small turbine it would mean an increase of only from $10 to $25 a year, which would be offset by the difference in first cost.

2 The greatest value of the paper lies in the fact that, better than a salesman, it shows to men having reciprocating engines, the great simplicity of construction of the turbine. Men familiar with reciprocating engines know what to do in case of breakdown, but with a

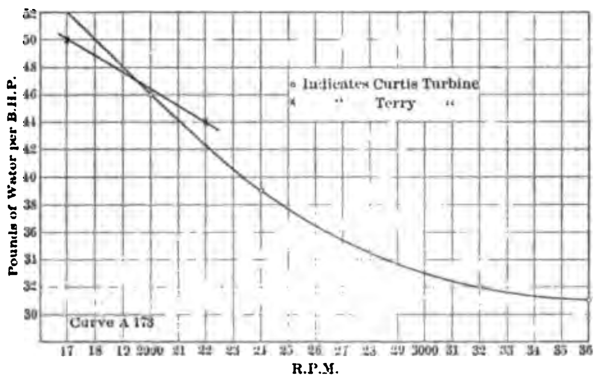


FIG. 1. COMPARISON OF TESTS IN FIG. 25 AND FIG. 28 REDUCED TO THE SAME WHEEL VELOCITY.

turbine a failure means a shutdown for several days. In a few years, however, every engineer will thoroughly understand the construction of a steam turbine and will be able to make his own repairs. Another fact which hindered the more frequent use of the steam turbine was that generators, pumps, blowers and other machines had to be designed specially for operation with turbines. That this is now being done is an acknowledgment that the steam turbine is here to stay.

CHAS. B. REARICK. The matter of speed for turbines driving centrifugal pumps is often a compromise, as the pump speeds are not always ideal for the turbine. Especially is this true for circulating work where the heads are often only 15 or 20 ft. and the delivery 20,000 gal. or more per min. It may be necessary to sacrifice some efficiency of the pump in order to run at a speed suited to the turbine. In such instances all the exhaust steam may be used to advantage in heating feed water, while the low cost of operation and the saving in oil and supplies will overcome the cost of increased steam consumption. When efficiency is of importance, as in isolated dynamo work, the speeds are usually such as to give results quite as good as those of the reciprocating engine, and in many cases better. The turbine has demonstrated its ability to give economical results in all cases where the speeds are favorable.

2 Regarding the rating of turbines, there is only one point of maximum efficiency of any design, so far as I know, and that is its maximum load. It is very similar to a gas engine in that respect. If we want maximum efficiency the turbine must carry its maximum load. That can be brought about in some cases by the nozzle system of design in which better economy is obtained at light loads by shutting off nozzles. But where this is done by hand regulation there is always danger of the load coming on without notice or without the engineer having opened any of the hand adjustments. Turbines should therefore be designed to eliminate hand regulation, and to accomplish this some builders provide for automatic operation of these valves. While this is successful within a limited degree, these valves may become leaky in service, and when once leaky the control of the turbine is beyond the operator, which may result in over-speeding to a bursting point in case the load is suddenly thrown off.

3 It follows that the number of controlling valves should be reduced to a minimum. If there is only one valve to control in a machine there is but one valve to look after and to keep tight. Few operators of turbines appreciate how serious the leakage of the controlling valve is to the proper governing of the turbine, especially on very light loads or no load. On the other hand, if turbines are kept well loaded these leaky valves are not noticed and as a result steam consumption is increased through their use rather than diminished, unless all the valves are wide open and the turbine is working up to its full capacity. The turbine which eliminates these dangers, it would seem, is the better machine, and in small units the difference in economy is entirely outweighed by the complications and dangers above cited.

F. B. Dowst. The B. F. Sturtevant Company have for years built reciprocating engines—single engines, multiple-single engines, and multistage engines. Later we built direct-current and alternating-current motors. We first came into touch with the steam turbine principle in 1883 when our attention was first called to the Wise steam motor. This motor was an impulse wheel, with four jets, I think, the steam impinging on buckets, no endeavor being made to expand the steam in nozzles. One of our engineers left us at that time to exploit the Wise steam motor. He returned after a year's sad experience in connection with the amount of steam that would flow through a small opening.

2 Our next experience was with the Dow steam motor. Mr. Dow came to us early in the nineties, I think it was, backed by Mr. Chisholm of the Chisholm Shovel Works of Cleveland, O. This turbine had previously been developed and used successfully to drive the flywheel in the Howell torpedo. Associated with Mr. Dow was Mr. Howard, for some time connected with the Fore River Ship and Engine Company, as it was then called. The Dow turbine was built in the Sturtevant works and was really a meritorious machine. It was what might be called an inward-flow reaction turbine. A motor of bronze was built and the method which Mr. Dow developed is now used in balancing our rotors.

3 A little later, during the development of the Curtis turbine, a representative of Mr. Curtis came to us for journals for high-speed work. He had trouble in finding a journal box capable of withstanding the high speed of his rotor shaft. A box that we used was very successful in solving the problem. We knew there was a somewhat limited field for high-speed motors for use with our fans, however, and were not quite ready to take up the Curtis turbine commercially.

4 A completed Dow turbine was frequently connected with one of our No. 6 blowers. The governing device was not developed, but that was not necessary in order to connect the turbine with a fan. I believe this turbine was tested in our works, and afterwards tested at the Massachusetts Institute of Technology for water consumption, which was found to be high. I have always thought that the Dow turbine possessed great possibilities and wondered why someone did not develop it.

5 Experiments with a steam turbine of our own for use with our fans resulted in the turbine described as the Sturtevant turbine. Mr. Orrok's description of the rotor is substantially correct, except that he omitted the fact that in the manufacture of this part we use an open-hearth steel forging of the best quality. Among the uses of the turbine are direct connection with generators, with fans for blowing blast furnaces, and with multivane fans for work on shipboard. Four fans with geared connection, recently built for a heating system in the West, have done good work.

6 An interesting problem was recently presented when the Navy Department insisted on fans for forced draft for the new torpedo boats

where oil fuel is to be used. This of course demanded a slow-speed turbine, but we think we have worked out a satisfactory combination by effecting a compromise between the fan and the turbine element. It is interesting to recall that a few years ago the Navy Department did not consider any motive power except a reciprocating engine. Later the electric motor came into use and now many of the new ships are equipped with forced-draft fans driven by electric motors.

7 There is not the slightest doubt that the general type of turbine discussed in Mr. Orrok's paper is here to stay. Engineers like it and engine builders must get ready to furnish turbine engines.

Chas. B. Edwards.[1] We are more particularly interested in the development of the large marine turbine, but our attention has been recently directed to the smaller turbines owing to the Navy Department specifying them for blowers; there is also a possibility of their use for circulating pumps and other auxiliary machinery on board ship. The Navy Department has increased the steam allowance for turbine installations over what it was a few years ago when it was limited to 50 lb. per h.p. hour.

2 The great problem in marine installations, particularly for naval purposes, is that of weight. The navy contracts specify a certain weight of machine and if we exceed that weight we must pay for it at the rate of about $500 a ton. In considering the turbine proposition, therefore, we must look at it not only from the mechanical side but also from the standpoint of weight. One of the difficulties, of course, is that of the exhaust. The weight of piping, fittings, valves, etc., runs up rapidly and it is therefore desirable that turbine auxiliaries should be placed as close to the condenser as possible; and in order to secure economical results it is also desirable to secure a low velocity of exhaust in the pipe lines.

V. F. Holmes.[2] The DeLaval Company has recently brought out a combination high-and-low-pressure steam turbine. Many plants where condensing water is available have an excess of exhaust steam from auxiliaries and the question has arisen whether a machine could not be devised for this class of service. That would necessitate storing up energy in times of an excess of exhaust steam to carry the machine over the periods of limited exhaust steam, and would involve expense and complications. What is desirable is a machine in which

[1] Chief Engineer, Fore River Shipbuilding Co., Quincy, Mass.

[2] Power Equipment Company, Boston, Mass.

both the exhaust and the live steam can be used economically without regenerators and other heat-storing devices.

2 The DeLaval combination high-and-low-pressure turbine is built with two nozzle compartments, one for high pressure and the other for low pressure. Each compartment is furnished with nozzles having the proper ratio of expansion for the conditions under which they operate. Some of the nozzles are furnished with shut-off valves for regulation under variable conditions.

3 Two steam connections are provided, one for high-pressure steam and the other for low-pressure steam, each connection leading to its own governor valve, which in turn is operated by a separate governor. The operation is entirely automatic, the low-pressure governor being set for a speed slightly higher than that of the high-pressure governor. On the total or partial failure of the low-pressure steam supply the machine will automatically draw from the high-pressure steam supply the steam necessary to make up the deficiency. Also in case of the complete failure of the low-pressure steam supply, the machine will operate on high-pressure steam, and under this condition will give practically the same economy as a high-pressure steam turbine.

4 The combination high-and-low-pressure turbine is built for conditions where continuous operation is essential and where the supply of low-pressure steam is intermittent or is apt to fail completely. The regulation when changing from one steam pressure to the other varies from 2 to 3 per cent, this being on an instantaneous change from one condition to the other, such as seldom occurs in actual service. A by-pass valve allows the admission of high-pressure steam into the low-pressure compartment, for operation under full-load conditions non-condensing. This by-pass valve is not automatic, and is simply to enable the machine to carry full load in case of failure of, or during repairs to, the condensing apparatus.

5 Both the low-pressure turbine and the combination high-and-low pressure turbine are built for steam conditions varying from 5 lb. pressure above atmosphere to 10 in. of vacuum at the steam inlet. They are also built for low vacuums for conditions where the temperature of the circulating water or existing condensers prohibits the maintenance of a high vacuum. The steam consumption of the machine varies somewhat with the sizes and operating conditions; the average machine operating with steam at atmospheric pressure and exhausting into a vacuum of 27 in. to 27½ in. will use from 28 lb. to 32 lb. of steam per b.h.p-hour.

6 The DeLaval Company is also building a high-speed, low-pressure turbine particularly adapted for direct connection to centrifugal pumps and blowers. This class of machine is built in both the low-pressure and combination high-and-low-pressure types, and consists of the DeLaval wheel direct-connected to the machinery to be driven. On account of the direct connection of the wheel and the elimination of the usual DeLaval reduction the machine can be economically operated only at high speed, and for this reason is not suited to direct-current generator work, but is particularly adapted for high-speed pumping and blower work, such as power-plant auxiliaries, boiler-feed pumps, elevator pumps, etc.

J. S. Schumaker. An error has crept into these figures that I am sure was not intended. That is, the figures given for the economy of the Terry steam turbine were obtained from a turbine with nozzles designed for 100 lb. pressure. But the steam pressure used on the test was, I believe, 150 lb. One other point that may in fairness be brought out is that the Terry turbine tests as offered here were made without representatives of the Terry Steam Turbine Company being present, while in the majority of the other cases cited the tests are shop tests.

Prof. Carleton A. Read. I am interested from the fuel side of the question in the use of non-condensing turbines in small manufacturing plants of from 75-kw. to 300-kw. capacity, where there is an excess of exhaust that can be used only for feed-water heating and heating the buildings in cold weather. We all agree that it is well not to have oil in the exhaust if the condensation is to return to the boilers, but many plants have a good and cheap water supply and after using as much of their exhaust as possible still have some going to waste. Nearly all of the tests quoted are from the manufacturers and without doubt are correct for the conditions under which they were made, but data as to coal consumption under actual working conditions would be of interest to the man buying an equipment for a small plant.

Prof. Ira N. Hollis. One aspect of the subject impresses me as important. The curves of efficiency used for comparing different turbines relate particularly to the thermodynamic efficiency of the machine or the number of pounds of steam per horsepower developed. It seems to me that where a steam turbine is connected with a pump, such as one used for feeding a boiler or for circulating water in a condenser, the machine ought to be treated as a whole. From this point of view, the number of gallons of water delivered per pound of steam or per pound of coal is an important factor and should be given in every case. Naturally the pressure against which the water is pumped is another factor. Ordinary reciprocating engines driving feed pumps are very uneconomical machines. I have had experience with pumps that used 100 lb. of steam per i.h.p. or even more. However, the efficiency of the pumps as a whole for delivering water into a boiler was never worked out.

2 It may be that the steam turbine is to replace the steam engine for all purposes about a power station, particularly if the high-pressure centrifugal pump can be developed into a highly efficient machine in connection with the turbine. It seems to me, therefore that it would be useful in connection with all tests of turbines used to drive pumps, to give the combined efficiency of the unit.

Prof. Edw. F. Miller. In looking through these figures of steam economies it will be noticed that the greater the load the smaller the amount of steam per horsepower. All the turbines I have had to do with would stand considerable overload, in some cases 80 per cent. I would like to know what decides the maker in rating his turbine. Apparently the economy line runs down as the overload goes on. Why not rate the turbine higher and get better economy?

John T. Hawkins. I was a pretty old engineer when the turbine was born and consequently know little about it except what I have learned by reading and observation. I am not going to try to impart information but I wish to ask a question. It seems to be a well-known fact that with the turbine engine, the higher the load the greater the efficiency within its limits. To what is the fact due that the turbine is more efficient under high load?

Richard H. Rice. Just a few words in explanation of why the turbine water rates decrease as the load goes up and of the effect of the various governing mechanisms on this action. The impulse turbine is essentially a turbine of partial admission. In a multi-stage condensing turbine of this type the buckets in the last stage are usually designed of the right height and the nozzles of the right proportion to use the entire circumference of the wheel. In a four-stage machine, the next to the last stage will use perhaps one-half the circumference of the bucket wheel for steam admission. It could be designed to use all the circumference, but that would involve undue shortening of buckets. In the second stage there is a further shortening of the arc of steam flow, and in the first stage, a still shorter arc is used, perhaps 90 deg.

2 In non-condensing turbines, if we were to attempt to use the entire circumference of the wheel the buckets would be so small that the machine would be impracticable and inefficient. We must therefore use a short arc, decreasing the cost of the governing mechanism and making a reasonable bucket speed possible. It is evident that the bucket speed must depend on the size of the machine and that, in connection with the operating speed, it is the prime consideration in the cost of the machine. It would be a mistake to make a 25-kw. machine with the same bucket speed as a 300-kw. machine, because the former would be so large in diameter and so expensive as to be impracticable.

3 It follows that one reason why the turbine increases in economy as the load increases is that a larger circumference of bucket wheel is used; a smaller percentage of the total power is wasted and therefore efficiency increases. Therefore, if the governing mechanism works by throttling we have this condition: the steam pressure and area of the nozzle system determine the amount of steam that can be used in the turbine. In a machine with nozzles wide open, the latter must be so designed that the turbine will carry maximum load, as otherwise the turbine would shut down at maximum load. It follows that nozzles designed for full pressure at maximum load will be greatly throttled when running with light load, and consequently the efficiency will decrease. Therefore it is advisable to govern the nozzle system in such a way that nozzles can be designed for full boiler pressure. By using a larger or smaller number of nozzles, and hence a larger or smaller arc of wheel, the full economy of the nozzles is obtained and only the proper number of nozzles are open for a given load.

Chas. H. Manning. The diagrams confirm the opinion I had formed of the small steam turbine to the effect that it is a steam thief. But that its virtues will outclass its sins I am thoroughly convinced. Recent developments of high-speed centrifugal pumps, fans and generators open a field for the turbine in which it is sure to succeed.

2 A small practical point is that almost all of these small high-

speed machines use the ring oiler, which is in general bad practice. It has a very small contact on the shaft and any small thing will stop its running. Furthermore, the rings frequently break. If for the ordinary ring oiler a chain with a large arc of contact is substituted, preferably a window-cord chain, it will never fail and will bring up ten times as much oil as a ring oiler. While this is a small point, any machine depends more on the perfection of its detail than it does on the theory on which it is built.

C. P. Crissey.[1] There is one type of the small turbine to which the author has given scant space; that is, the small condensing machine. While, perhaps, the field is not so wide for this type as it is for small turbines exhausting at or above atmosphere, it cannot be ignored. Practically all marine work requires condensing prime movers, and many small stations use this type. Only one example of a condensing machine is referred to by the author, the results of tests being shown in Fig. 30. It would be a mistake to consider this curve as representative of small condensing turbines.

2 Small turbines as well as large derive great benefit in economy from high vacuum, and a vacuum of 28 in. is easily obtained on small machines of proper design. In a well-designed small turbine the vacuum shows no greater tendency to fall with the increase of load than in large machines. Why the Kerr turbine shows a loss in vacuum as the load and hence steam flow increase, we are unable to tell definitely from this paper. It will be noted that the steam is discharged from the buckets on each side of the wheels. It is therefore necessary for one-half of the total flow to pass about the wheels in order to reach the succeeding nozzles. Excessive velocities and throttling will occur in the low-pressure stages where the volumes encountered are great, unless large areas are provided for this steam. I understand that in the Kerr turbine this throttling is obviated as much as possible by providing holes in the wheels. These holes, however, increase the windage loss.

3 One of the reasons for abandoning the Riedler-Stumpf turbine in Germany was the inability of its buckets to handle large volumes of steam successfully. The same objection holds against all machines of the Riedler-Stumpf type.

4 The only small turbines having buckets capable of caring efficiently for large volumes of low-pressure steam are the De Laval and the Curtis types. The DeLaval turbine is seriously handicapped

[1] General Electric Co., West Lynn, Mass.

by its high rotative speed, while the Curtis turbine, due to its pressure and velocity stages, is capable of moderate speeds. In order to show that the results of Fig. 30 are not typical of all small condensing turbines, I will say that Curtis condensing turbines of from 100 to 200 h.p. give economies of 18.5 to 15.5 lb. of steam per b.h.p. hour when operating with 150 lb. dry steam and 28 in. vacuum.

5 Regarding the curves of this paper, I believe they should be compared at rated speed, because the bucket angles are designed for this speed. The rated speed may be taken as the maximum stated upon the curves.

W. J. A. LONDON. I wish to add something to my remarks in connection with the curve mentioned by me earlier in the discussion; namely, a comparison between Fig. 25 and Fig. 28. In making this curve, I used only absolute tests according to the figures mentioned and made no deductions whatever except in the question of relative peripheral speeds. If the curves plotted in Fig. 28 are reproduced for a series of full-load points on a speed basis, a positive curve will be formed. On this curve is plotted the two full-load tests shown in Fig. 25. Now, then, as there are only two points given in the Terry tests, it is impossible from these tests to obtain the nature of the curve, but the point I particularly wished to bring forward was that these two points practically coincide,—one test, as a matter of fact, being better and one worse,—both of them being so near the Curtis curve as to make little difference. They are not so far away as Mr. Rice would have us believe from his diagram.

2 The point has also been raised as to whether emergency governors were fitted on other makes of turbines besides the Curtis. Particularly with the Terry turbine and I believe with the majority of the other makes, an emergency governor is not provided for the reason that a positive type of governor is fitted on the main shaft. The worst that can happen is the breaking of a spring, which would immediately close the valve. With a governor driven by a gear shaft an emergency governor is more essential for the reason that the gears are likely to break; hence some form of governor is used on the main shaft. If the direct-connected governor on the main shaft is likely to get out of order, why then is the emergency governor not likely to get out of order when placed in the same position? Up to the present time the Terry Turbine Co. has not had a machine burst, and with the type of governor adopted and the speeds em-

ployed, the designers consider an emergency governor an unnecessary luxury.

R. H. Rice. Mr. London claims that the steam consumption of the Terry turbine is the same as that of the Curtis turbine, when operating the Terry turbine at its designed speed and reducing the speed of the Curtis turbine to two-thirds of its designed speed of 3600 r.p.m. The inaccuracy of this comparison can be readily understood when it is known that the angles of the buckets in the Curtis turbine would be radically changed if designed to run at two-thirds of the present rated speed.

2 In discussing emergency governors, it must be realized that we are dealing with much higher speeds than those usual with reciprocating engines. It has been found best in many plants to install emergency governors on reciprocating engines. If this is desirable on slow-speed apparatus, how much more desirable, and even necessary, is it on high-speed apparatus like steam turbines. Many other accidents besides the breaking of a spring can happen to a positive type of governor fitted to the main shaft, and any one of these is sufficient to cause a dangerous increase in speed of the turbine. An emergency governor can be made to possess the utmost certainty and reliability of action, since its function is to shut down a machine and not to regulate its speed.

J. H. Libbey. The applications of small steam turbines mentioned by the author, except for driving high-pressure fans, refer to uses with auxiliary apparatus in a central power station. For this purpose a small steam turbine must be considered in competition with a reciprocating engine, and in general the choice will be decided by the following considerations: First cost, attendance required, maintenance and repairs, space, economy and influence on design of power station.

2 *First Cost.* At present, when the service permits operation at speeds approximating those for which the turbine was designed, the cost of the turbine is somewhat lower than that of a corresponding reciprocating engine.

3 *Attendance Required.* The attendance required for a small steam turbine is less than that required for any other type of steam machinery. It approaches very closely that required for an electric motor.

4 *Maintenance and Repairs.* In Par. 32, the author indicates

that small turbines have been running for only three years. Obviously in this time no data of great value could be obtained to enable a decision to be made in regard to maintenance and repairs. The evidence, however, strongly indicates that they will be materially less than for a reciprocating engine.

5 *Space.* Steam-turbine-driven apparatus is generally characterized by the small space required. In a great many cases, a turbine unit can be installed where a reciprocating engine would be impossible.

6 *Economy.* An inspection of Fig. 28 and Fig. 29 shows that for the best conditions a turbine can deliver a horsepower with as little steam as, or less steam than, the same size reciprocating engine. In installations where the conditions are not favorable, the economy is reduced. Unfavorable conditions for a steam turbine are low superheat, low steam pressure, high back pressure or reduced speed of the turbine, on account of the characteristics of the driven machine. The last condition is the most likely to cause reduction in the economy.

7 It should be borne in mind that in the ordinary large central station where fairly large generating units are installed, the steam consumption of the auxiliaries does not in general amount to more than 10 per cent of that of the main generating units. In such cases, the auxiliary exhaust will heat the feed water to about 175 or 180 deg. fahr. A considerable increase of steam consumption of the auxiliaries can be permitted before there is sufficient exhaust to heat the feed water to 212 deg. fahr. In most cases, therefore, the steam consumption of these small auxiliaries is a matter of secondary consideration.

8 In the various auxiliaries generally used the inherent conditions which would affect the steam consumption would be in general as follows:

Exciter, favorable.
Circulating pump, speed low for best results.
Hot-well pump, favorable.
Forced or induced-draft fans, speed low for best results; special design of fan required.
Feed pump, speed low; however, the steam consumption of a turbine-driven multistage centrifugal feed pump is much lower than that of a reciprocating pump of the same capacity.

9 In this connection Mr. Orrok's statement in Par. 32, that there

seems to be no change in steam use with length of service is of importance as it is well known that the steam consumption of engines or pumps increases greatly with wear of valves, rings, pistons, cylinders, etc.

10 Many central stations are today operating the auxiliaries with superheated steam. Very few changes are required in the structure of a steam turbine to adapt it to superheated steam by the use of which the economy is improved. The reciprocating engine gains in economy from superheat, but greater changes in the design are required to obtain satisfactory operation, and the expense of the engine is therefore increased.

11 *Influence of Design on Power Station.* Turbine-driven exciters are generally light in weight and compact. They can be set on the engine-room floor without a heavy foundation or resulting vibration.

12 Circulating-pump units are of simple design. In many cases, a combination of auxiliaries may often be effected. There is on the market a jet condenser, the centrifugal pump and air pump of which are on the same shaft with the turbine. When in surface-condenser work the conditions are such that the speed of the circulating pump is subject to little variation, the turbine, circulating pump, and hot-well pump can be mounted on the same shaft. One manufacturer is prepared to add a rotary vacuum pump to these, either direct-connected or chain-driven. This arrangement gives practically one auxiliary for a surface condenser in place of three.

13 *Future Designs.* The small steam turbine has sufficiently justified its existence. The future will undoubtedly show types with improved economy, especially at reduced speeds, simplicity of design, rugged characteristics, ability to operate without attention, interior construction that is easily accessible and such that few repairs due to wear are required.

THE AUTHOR. The author is greatly pleased with the reception accorded his paper and the amount of discussion which it brought out. He must take exception to a comparison of water rates plotted on percentages of load as abscissæ, and a new diagram has been prepared, Fig. 1, showing the results of all the water-rate curves plotted with bucket speed or peripheral velocity as abscissæ, obviously a much better measure of the performance of these machines. The author would like to take up the question of improper entrance and discharge bucket angles in machines of the Riedler-Stumpf type, as well as the fluid friction question, both mentioned in the discussion of Mr. Rice; but these should be the subject of a mathematical paper and are not of serious importance in a small turbine. The author feels that Mr. Ball has failed to grasp the fact that with small rotating masses speeds of from 600 to 3000 r.p.m. are not as objectionable as a speed of 150 r.p.m. in a modern four-valve engine, or 100 double strokes per minute in a direct-acting pump.

2 Replying to Prof. Hollis' discussion, the author knows of many power plants where entire reliance is placed on turbine-driven multi-

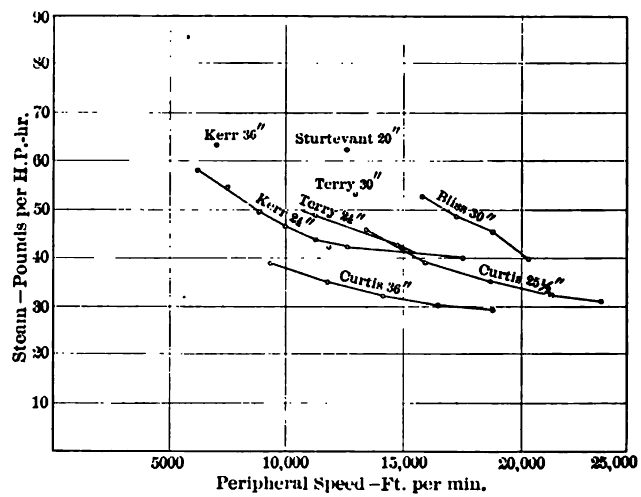


FIG. 1 STEAM CONSUMPTION OF SMALL TURBINES PLOTTED WITH PERIPHERAL SPEEDS AS ABSCISSAE

stage centrifugal pumps for feed-water service. He knows of no case where an attempt has been made to find the coal consumption of the feed pumps directly; in other words, the duty of the pumps. It has usually been obtained through the steam consumption with a knowledge of the evaporation constant of the plant. The use of Venturi meters in the feed lines and in the steam connections to the turbine-driven feed pumps would give this duty directly, and a partial installation of this nature has been made at the Waterside Station of the New York Edison Company.

No. 1243.

TESTS UPON COMPRESSED AIR PUMPING SYSTEMS OF OIL WELLS

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When the Louisiana oil fields at Evangeline were in full operation, they offered exceptional opportunities for the study of air lifts. Nearly every known method of piping the wells was in use. The air plants originally installed were the crudest affairs imaginable, having been erected in feverish haste during the boom several years ago. When the production of the fields began to decrease, and the price of oil also declined, it was realized for the first time that the operating expenses were abnormal, and that unless greater economy were practiced, disastrous results would follow. Few changes were made, however, up to eighteen months ago, beyond the purchasing of additional equipment.

2 Each concern has a central station or air plant and all the compressors therein are connected to a manifold from which the air lines lead to the various wells on the property held by that concern. The manifold design is such that by manipulating the valves, any machine may be made to operate any of the wells.

3 Often the air lines reach the wells by a roundabout way, and have innumerable short bends, valves, double swings to avoid pipe cutting, and plugged tees instead of elbows. All of this tends further to decrease the economy of the operation, and taking all things into consideration, it is little wonder that the efficiencies of the plants were low. The size pipe used for these air lines is designed neither for the amount of air to be transmitted nor for the distance it is to be carried, but is with one exception 2 in. in diameter.

4 The boilers of the air plants are of 40 h.p., of a portable contracted waist type, and few were covered with asbestos. The boilers were so set that one-fifth of their lengths projected into the open, as

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indicated in Fig. 1, in order to avoid the necessity of perforating the roof to receive the stacks and to provide cooler boiler-rooms, regardless of the heat wasted.

5 The redeeming feature in all the plants is the type of compressor in general use. These compressors are generally of high grade, and display remarkable endurance. It is common for a machine designed for 350-lb. pressure to operate under a pressure of 500 lb., and at speeds far in excess of those for which it was designed. The most popular type of compressor has the duplex steam end and compound or two-stage air end. The steam cylinders are fitted with Meyer adjustable cut-off valves and the air cylinders in some instances with piston and in others with Corliss intake valves and poppet discharge valves. Plain speed governors are used and the capacities of the

Fig. 1 A Typical Air Plant

compressors range from 100 to 1000 cu. ft. of free air per minute and operate at pressures of from 150 to 750 lb. per square inch. The machine best adapted to the purpose, however, is the 500-cu. ft., 500-lb. type.

TERMS

6 An explanation of certain terms to be used may not be out of place.

"Submergence in feet" refers to the number of feet below the surface of the fluid (after the well has been pumped down, and is operating under its normal conditions) that the air under pressure is admitted.

"Per cent of submergence" is the submergence in feet divided by the total number of feet of vertical discharge line, measured from the point of admission of the air to the point of discharge of the fluid.

"Volumetric efficiency" of the compressor is the actual amount of free air that is compressed and discharged by the cylinder, divided by the cubical contents of that cylinder.

"Free air" is air at standard temperature and pressure.

"Pumping head" is the vertical distance in feet (after the well is pumped down, as before stated) from the fluid level in the well to the point of discharge.

$$\text{"The Constant"} = \frac{\text{Gal. per minute} \times \text{pumping head in feet}}{\text{Cu. ft. of free air per minute}}$$

$$\text{"The Ratio"} = \frac{\text{Cu. ft. of free air per minute}}{\text{Cu. ft. of fluid per minute}}$$

DESCRIPTION OF SYSTEMS

7 Fig. 2, 3, 4, and 5, illustrate the air lift systems that are and have been in use on the oil fields.

8 Fig. 2 shows the Straight Air or Sanders system. The well top is sealed as shown at A. Compressed air is forced through the pipe B into the space between the discharge or eduction pipe C, and the well casing D.

9 When without air pressure, the fluid in the well will stand at some point such as E, the level in the air space and the discharge line being identical. When air is forced through B, the level of the fluid in the air space is gradually forced down until the end of C is uncovered. Instantly some of the air escapes into the discharge pipe C, lowering the air pressure in the air space F. This causes the fluid to rise in and up the air space and discharge pipe until a point is reached where air and water pressure balance. Then, more air coming in, the pressure again rises, the fluid level is forced down as before, more air escapes into the discharge pipe, and thus the cycle is repeated. As may be readily seen, the air that rushes into the discharge line carries the "slug" of water that has just previously entered.

10 Fig. 3 shows what is commonly known as the Central Pipe system. The discharge line A is placed inside of the well casing as before and inside of the discharge is suspended a small air line usually 1¼ in. in diameter. The end of the 1¼-in. line is plugged and a number of ⅛-in. holes are drilled inclining upwards in the last joint of pipe. Air is forced down through the small air line shown, passes out of the ⅛-in. holes, and mingles with the fluid carrying it out through the discharge line A. It is generally supposed that the fluid in this

case is discharged because of the aëration of the fluid in the discharge pipe which in turn is caused by the intimate commingling of air and fluid. The weight of the fluid column inside of the discharge pipe is therefore less in pounds per square inch than that without and the energy due to this difference in weight is utilized to lift the fluid and overcome the various losses.

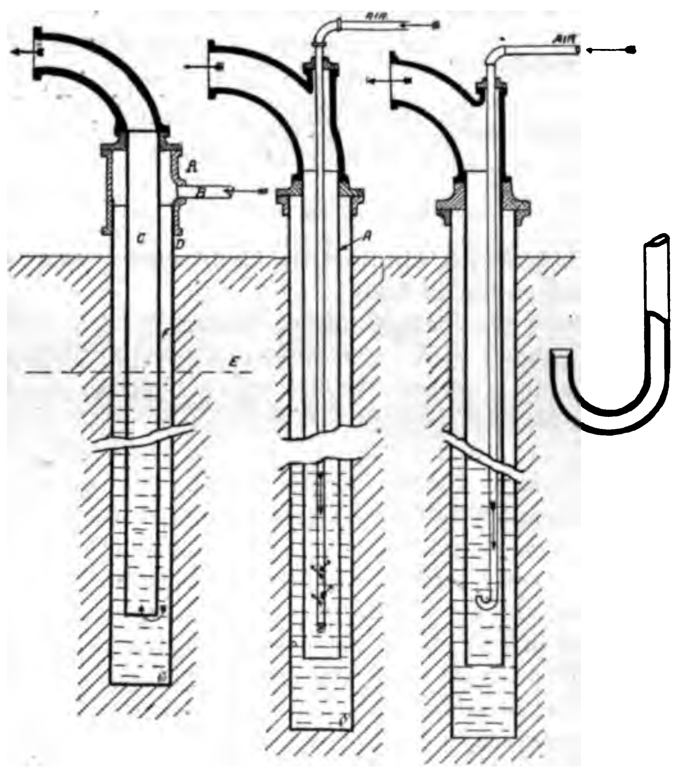

FIG. 2 STRAIGHT AIR LIFT SYSTEM — FIG. 3 CENTRAL PIPE SYSTEM — FIG. 4 RETURN BEND SYSTEM

11 What is commonly known as the Open End system of air lift was at one time in quite extensive use on the field. It is similar to the system just described except that the small air line is open at the lower end, and of course there are no holes drilled in the air line.

12 Fig. 4 illustrates a form of the Return Bend system. It is claimed by the inventor that: "It consists in improved processes and

apparatus whereby the compressed air is delivered in bulk into the lower end of the water eduction pipe, and the water and air are caused to ascend through said pipe in distinct alternate layers of definite dimensions."

13 The use of this system has been discontinued in Evangeline because, as the field managers told the writer, it failed to produce as large a quantity of fluid as that produced by other systems.

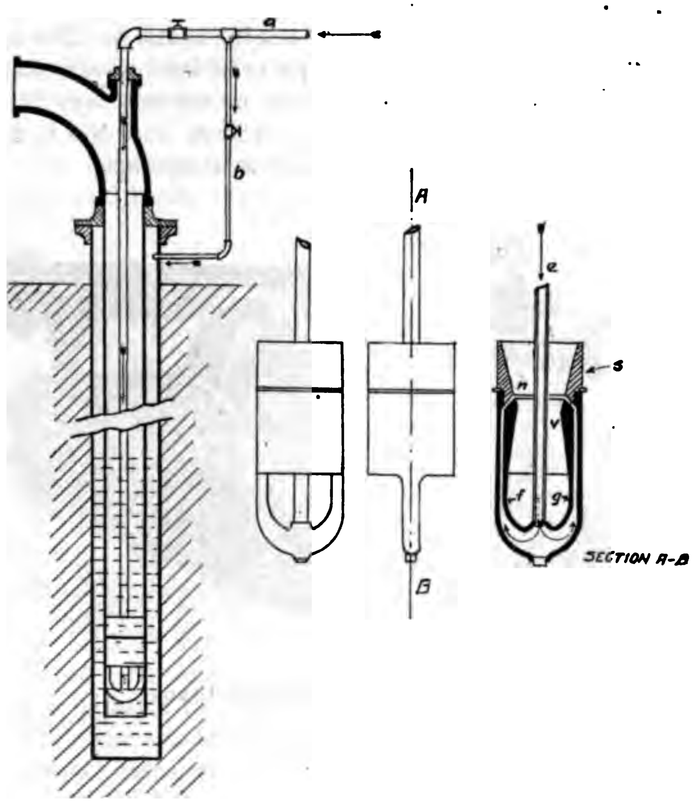

Fig. 5 System Combining Features of other Systems Described

14 Fig. 5 shows a patent system which in reality is a combination of the several systems already described. The claims of the inventor are: less submergence, and hence less air pressure necessary, decreased air consumption, or with an equal amount of air, increased fluid yield.

15 Compressed air is forced through *a* down into the foot piece, which is placed at that point of submergence shown by test to be most economical. The well top is sealed and air under pressure

is also admitted between the casing and discharge pipe on the water head by means of the branch shown at *b*. This forces the fluid to a higher level in the discharge pipe and also prevents fluid in the air space or chamber from vibrating and foaming. This is quite an advantage in oil well pumping as the liability of making "riley oil" is thereby greatly lessened.

16 The footpiece shown in section is made of cast brass and is in two parts. The air on reaching the foot piece divides and goes up through the hollow prongs *f* and *g* and out the nozzle *n*. The nozzle is adjusted to receive the quantity of air to be used by screwing the upper part *s* of the footpiece, in or out as the case may be. To increase the velocity of the fluid in the discharge line, the footpiece is restricted and formed into a "venturi" as shown at *v*.

FIG. 6 TYPE OF COMPRESSOR USED

TEST NO. 1

17 The Crowley Oil and Mineral Company was the first to take active steps for the improvement of their plant and pumping equipment. They decided to install the patent air lift last described (Fig. 5). A test of the old system was first made to determine the amount of compressed air used and the fluid yield. The new equipment was next installed and a similar test made of the same duration and under the same conditions. The tests and installation were conducted on Well No. 32, 1805 ft. deep, and located 542 ft. from the compressor operating it. The air to the well was controlled by means

of a manifold in the plant and was conveyed to the well top through a 2 in. pipe line which as usual was in poor condition and badly designed.

18 The system of pumping was that illustrated in Fig. 2. The well casing was 6 in. in diameter, suspended inside of which was a 4-in. discharge line.

19 The compressor was a duplex steam and compound air type made by the Ingersoll-Rand Company and designed to compress 1000 cu. ft. of free air per minute to 350 lb. pressure. The steam end was fitted with Meyer adjustable cut-off valves and the air end with Corliss intake and poppet discharge valves. The machine is shown in Fig. 6.

20 The discharge pipe from the well top was run up into a steel tank of known dimensions and the amount of fluid pumped during the

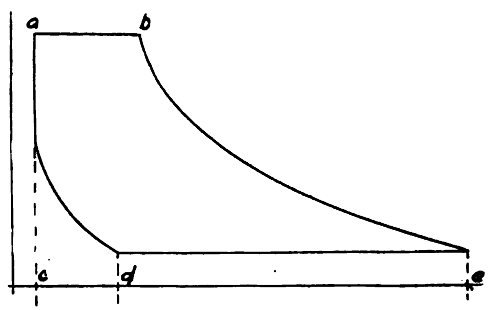

Fig. 7

test ascertained by direct measurement. Air gages, previously tested, were placed both at the compressor and at the well top thereby making it possible to determine the friction losses in the manifold and air line and also the actual pressure at the well top. Simultaneous indicator cards were taken from the steam and air ends of the compressor, and from these cards were obtained the volumetric and mechanical efficiencies, the steam and air horse powers, and the steam consumption (theoretical) of the machine.

21 The volumetric efficiency was assumed to be the ratio of the piston travel during admission stroke to the total piston displacement, ($\frac{de}{ce}$ on the indicator card, Fig. 7). Sometimes this method is inaccurate and unsatisfactory (1) because the entering air at atmospheric temperature and pressure is heated by

contact with the cylinder walls and piston; and (2) because of leakage of air from the compression side of the moving piston to the suction side. Neither the expansion resulting from the first condition nor the reduction in volumetric efficiency resulting from the second are observable on the indicator card.

22 The first inaccuracy was partially overcome by placing a recently calibrated thermometer as far down in the intake pipe of the compressor as possible, noting the temperature and making the necessary corrections as will be observed in the log of results.

23 The method of ascertaining the volumetric efficiency, that the writer would have used, but for his inability to obtain the necessary apparatus, was in brief as follows:

Connect the air discharge of the compressor to an enclosed tank. From this tank, connect to a cooler and from thence to a second enclosed tank of known dimensions.

Place a regulating valve between the first tank and the cooler, setting the valve to maintain the pressure in the first tank at that point at which the efficiency is to be determined.

Attach test gages to both tanks and a reliable thermometer to the second tank.

Start the compressor and note the temperatures of the intake air and of the air in the second tank both at the beginning and end of the run. Note also the initial and final air pressures and the reading of the barometer, and the speed in revolutions per minute of the compressor.

The volume of air compressed is then determined from the formula:

$$V = v \frac{273 + T}{R} \left(\frac{29.92 \times P_1}{273 + T_2} - \frac{29.92 \times P}{273 + T_1} \right)$$

where

V = Volume of air compressed.

v_1 = Cubical contents of the second tank.

T = Room, or intake air, temperature.

T_1 = Initial temperature of the air in the tank.

T_2 = Final temperature of the air in the tank.

R = Reading of the barometer in inches of mercury.

P and P_1 = Initial and final air pressures in the tank.

The volume of air thus obtained divided by the total piston displacement equals the volumetric efficiency.

OBSERVATIONS

24 Every thirty minutes for a period of six hours, readings were taken of the boiler pressure gage, the air gage at the compressor and at the well top, the r.p.m. of the compressor, the temperature of the intake air and of the barometer. A set of indicator cards, and also a sample of the fluid pumped from the well, were taken at each interval.

25 The temperature of each sample of fluid was noted; it was then placed in a proper receptacle, and at the end of the test, the weight of a gallon was ascertained, together with the specific gravity of the oil. The amount of fluid pumped was determined, as before stated, by direct measurement, due allowance having been made for the samples that were withdrawn.

26 When these tests were run, no attempt was made by the writer to re-design the air lines, or to correct in any manner the numerous other defects. The old system was tested just as it had been operated, and the new system was installed and tested under the same adverse conditions. After both systems had been tested, some few of the defects were corrected in the manifold and air line design, thereby insuring more economical operation in the future.

TABLE 1 SUMMARY OF RESULTS

THE CROWLEY OIL AND MINERAL COMPANY, EVANGELINE, LA.

	Old System	New System
Duration of test, hours	5.5	6.0
Mean i.h.p.	122.56	89.19
Mean water h.p.	9.97	10.36
Mean air h.p.	107.38	79.16
Gallons of fluid per second	0.542	0.608
" " " " hour	1953.6	2188.2
Barrels of fluid per hour	46.51	52.1
Weight of 1 gal. of fluid	8.7	8.69
Mean temperature of fluid, deg. fahr.	111.5	113.2
Percentage of salt water in fluid	87.3	86.7
" " sand "	2.2	1.9
" " crude oil "	10.5	11.4
Barrels of oil per hour	4.86	5.94
Barometer reading, inches of mercury	29.95	29.94
Specific gravity of oil	0.9	0.9

	Old System	New System
Constant	63.1	101.8
Size of air line, inches		1.25
Total depth of well, feet	1805.0	1805.0
Size of casing, inches	6.0	6.0
Height above ground to which the fluid was pumped, feet	18.5	18.5
Size of discharge line used, inches	4	4
Total length of vertical discharge line	1513.5	1513.5
Total length of vertical air line in well	1513.5	1493.0
Dimensions of compressors, inches*	10x22x16x20–	7½x18x16x20
Number operated	1	1
Kind of fuel used†	Crude oil	Crude oil
Gallons of fuel used per hour	48.36	35.27
Barrels of fuel used per hour	1.15	0.835
Price of 1 bbl. of oil at time of test, dollars	0.90	0.90
Cost of fuel for producing 1 bbl. of fluid, dollars	0.0222	0.0146
Cost of fuel for producing 1 bbl. of crude oil, dollars	0.212	0.126

*Type of compressor used, Rand Drill Co. Imperial Type X, Steam Cylinders, compound air cylinders.

†Type of boiler, oil well supply, portable contracted waste.

Test No. 2

WELLS NO. 12, 30, AND 32 OF THE CROWLEY OIL AND MINERAL COMPANY

27 The saving in air volume accomplished by the new system led those interested to endeavor to operate two wells with one machine, something before considered impossible in the field.

28 Well No. 30 was forthwith tested, though not with sufficient accuracy to warrant the publication of the results, and the approximate pumping head and submergence established. The new system was then installed with the requisite pipe to equalize the submergence (hence working pressure) of this well with that of No. 32. How successfully the working pressures of the two wells were equalized may be seen by reference to Table 2 of the Appendix.

29 The two wells in question were then connected to one air compressor with gratifying results. No trouble was experienced in starting, and the machine furnished air in abundance for steady operation.

30 Preparations were being made to run the usual test when the compressor operating Well No. 12 "went dead." This last named well had been previously tested and equipped with the new system. This shutdown, of course, would mean a loss of at least a day's pro-

duction from the well, amounting to quite an item, so the writer advised that this well be also connected to the machine already operating No. 30 and No. 32. By speeding the machine up a few revolutions, the additional load was easily taken care of as may be more fully noted by reference to the accompanying log (Table 2).

TABLE 2 SUMMARY OF RESULTS

WELLS No. 12, 30, 32, CROWLEY OIL AND MINERAL COMPANY

Duration of tests, hours	6.0
Mean (total) i.h.p.	151.1
" " w.h.p.	25.14
" " a.h.p.	129.05
Total gallons of fluid per hour	6168.0
" barrels " " " "	146.87
" " " oil " "	16.17

WELL No. 12

Weight of 1 gal. of fluid	8.5
Temperature of fluid	118.5
Per cent of salt water in fluid	87.2
" " " sand " "	1.3
" " " crude oil " "	11.5
Barrels of oil per hour	6.44
Specific gravity of oil	0.87
Total depth of well in feet	1705.00
Size of casing, inches	6.00
" " discharge line, inches	4.00

WELL No. 30

Height above ground to which fluid was pumped, feet	17.5
Total length of vertical discharge line	1025.5
" " " " air line	992.58
Weight of 1 gal. of fluid	8.65
Temperature of fluid	120.2
Per cent of salt water in fluid	88.3
" " " sand " "	1.5
" " " crude oil " "	10.2
Barrels of oil per hour	4.83
Specific gravity	0.9
Total depth of well in feet	1920.00
Size of casing, inches	6.00
" " discharge line, inches	4.00

Height above ground to which fluid was pumped, feet	18.00
Total length of vertical discharge line	1516.3
Total length of vertical air line in well	1494.2

WELL No. 32

Weight of 1 gal. of fluid, pounds	8.7
Temperature of fluid	114.5
Per cent salt water in fluid	86.9
" " sand " "	1.8
" " crude oil " "	11.3
Barrels of oil per hour	4.90
Specific gravity	0.9
Total depth of well, feet	1901.00
Size of casing, inches	6.00
" " discharge "	4.00
Height above ground to which fluid was pumped	18.5
Total length of vertical discharge line	1513.0
Total length of vertical air line in well	1493.0
Size of air lines in well, inches	1.25
Barometer reading, inches of mercury	29.95
Dimensions of compressor, inches*	10x22x16x20
Number operated	1
Kind of fuel used†	Crude oil
Barrels of fuel used per hour	1.45
Price of 1 bbl. of oil at time of test, dollars	0.90
Cost in fuel of producing 1 bbl. of oil, dollars	0.074

*Type of compressor used, Rand Drill Co. Imperial Type X, duplex steam cylinders, compound or two stage, air cylinders.

†Type of boilers, oil well supply, portable contracted waste.

TEST No. 3

WELL NO. 2, MAMOU POWER COMPANY

31 This test was run in the same manner as those preceding except that the fluid field was ascertained by means of a two-foot rectangular weir placed between the earthen fluid and oil pits, the salt water bleeds of the former having been closed. The old system used was that illustrated in Fig. 3.

32 The depth of fluid over the crest of the weir was measured by means of the ordinary hook gage calibrated to read accurately in hundredths of a centimeter. The weir constant was previously determined by testing in the usual way, using a sample of the fluid as pumped from the well.

TABLE 3 SUMMARY OF RESULTS

WELL NO. 2, MAMOU POWER COMPANY

	Old System	New System
Duration of tests, hours	10.0	10.0
Mean i.h.p.	99.1	62.8
" w.h.p	9.85	13.36
" a.h.p	82.5	50.4
Gallons of fluid per second	0.694	0.849
" " " " hour	2499.6	3056.4
Barrels of fluid per hour	54.75	72.77
Weight of 1 gal. of fluid, pounds	8.72	8.75
Mean temperature of fluid, deg. fahr.	118.3	117.9
Percentage of salt water in fluid	87.7	86.1
" " sand " "	1.2	1.6
" " crude oil in fluid	11.1	12.3
Barrels of oil per hour	6.08	8.95
Specific gravity of oil	0.9	0.9
Barometer reading, inches of mercury	29.94	29.93
Weir constant	24.39	24.39
Pumping constant	97.1	202.9
Total depth of well in feet	1901.0	1901.0
Size of casing, inches	6.0	6.0
Height above ground to which fluid was pumped, feet	3.33	3.33
Size of discharge line used, inches	4.0	4.0
Size of air line in well, inches	1.25	1.25
Total length of vertical discharge line	1500.0	1500.0
Total length of air line in well	1489.5	1489.5
Dimensions of compressor, inches*	7½x18x16x16	7½x18x16x16
Number operated	1	1
Kind of fuel used†	Crude oil	Crude oil
Gallons of fuel used per hour	44.22	30.53
Barrels of fuel " " "	1.05	0.727
Price of 1 bbl. of oil at time of test, dollars	0.85	0.85
Cost in fuel of producing 1 bbl. of fluid, dollars	0.0163	0.0085
Cost in fuel of producing 1 bbl. of crude oil, dollars	0.128	0.069

*Type of compressor used, Hall Steam Pump Co.,Duplex steam cylinders, compound air cylinders, Plain "D" valves on steam end, poppet valves on air end.

†Type of boiler, 72"x18' horizontal return tubular, manufactured by the Lockout Boiler Co.

CONCLUSION

33 A careful examination of the tests brings out several points that may require explanation.

34 The loss of air pressure by friction in the small 1¼-in. air line in the well, to which the footpiece of the new system was attached,

was approximately determined as follows: Pipe connections were made at the well top, so that by the manipulation of various valves the air from the main line could be sent either through the $1\frac{1}{4}$-in. air line or into the space between the well casing and the discharge line. By noting the pressure gage readings in each instance, the friction loss (assuming that there is no loss by friction when air is forced between casing and discharge) is represented by the difference in the readings. Corrections were made, of course, for that part of the discharge line below the footpiece.

35 It was impossible to obtain the actual friction loss in said $1\frac{1}{4}$-in. line by other means more accurate than those employed. While some little error may be involved in assuming no friction loss in the one instance, a comparison of the loss thus obtained with the theoretical loss is quite favorable, the former loss being the greater.

36 Reference to Table 3 will show that the working submergence of the new system is less than that of the old, in spite of the fact that there is the same amount of pipe in the well in each case. This is due to the additional drop in pumping head caused by the increase of fluid yield. All calculations of submergence and pumping head were made from the observed air pressures after correcting for friction losses, etc. The mean of these calculations was verified as far as possible by actual measurement. This was done by shutting down the compressor after the well had been in steady operation for several hours and pulling the discharge line. The point at which the fluid stood, while the well was being pumped, was plainly defined on the pipe. The time required after shutting down the compressor to pull the first "triple" from the well was a fraction less than two minutes. Comparison of the actual pumping head and submergence thus obtained with those obtained by calculations from the pressure gage readings was in each case very close, a difference of 10 ft. 2 in. being the maximum.

37 Acknowledgment of valuable aid during tests is hereby made to the following who checked the writer in his various observations: on Well No. 32, to Mr. B. Brand, of the Crowley Oil and Mineral Co.; on Wells No. 12, 30 and 32, to Mr. Brand, Mr. J. Murphree and Mr. S. Bolin, of the Crowley Oil and Mineral Co.; on Well No. 2 of the Mamou Power Co., to Mr. J. A. Sonet of that company and his able assistants; and especially is the writer grateful to Mr. J. W. Smith for courtesies extended during the former's sojourn on the field.

APPENDIX

TABLE 1 TESTS OF WELL NO. 32

CROWLEY OIL AND MINERAL CO., EVANGELINE, LA.

Old System, November 4, 1907

Number	Time	Cu. Ft. of Free Air per Minute (Displacement)	Volumetric Per Cent Efficiency	Temperature of Air Degrees Fahr.	Cu. Ft. of Free Air per Minute (Actual)	Fluid Level in Tank	Gallons per Minute	Cu. Ft. of Fluid per Minute	Ratio	Air Pressures: Discharge	Air Pressures: Intercooler	Air Pressures: At Well	Pumping Head	I. H. P.	W. H. P.	A. H. P.	R. P. M.	Submergence in Feet	Submergence Per Cent	Pumping Efficiency Per Cent
1	1:15	632.16	92.0	86.3	555.75	1 ft. 8¾ in.	32.56	4.35	127.7	155.0	60	150.0	1167.0	118.3	10.03	103.37	72	346.5	22.8	9.7
2	1:45	623.38	91.5	86.3	545.06	....	32.56	4.35	125.2	157.2	60	153.5	1159.82	119.1	9.98	103.6	71	353.58	23.2	9.6
3	2:15	632.16	92.0	86.3	555.75	....	32.56	4.35	127.7	155.1	60	150.0	1167.0	118.3	10.03	103.4	72	346.5	22.8	9.7
4	2:45	632.16	92.0	86.3	555.75	....	32.56	4.35	127.7	157.4	60	149.0	1169.3	118.2	10.00	102.8	72	344.19	22.6	9.8
5	3:15	667.28	90.0	86.3	573.83	....	32.56	4.35	131.9	155.0	62	152.0	1162.7	123.4	9.99	107.3	76	350.82	23.1	9.3
6	3:45	658.50	91.0	86.0	571.79	....	32.56	4.35	131.5	166.3	60	150.0	1167.0	121.2	10.03	105.4	75	346.5	22.8	9.6
7	4:15	693.62	91.0	86.0	603.51	....	32.56	4.35	138.6	166.0	61	155.0	1155.45	130.5	9.92	113.5	79	358.05	23.6	8.8
8	4:45	667.28	90.0	86.3	573.83	....		4.35	131.9	166.0	60	155.5	1054.35	135.5	9.92	117.8	76	359.15	23.7	8.3
9	5:15	658.50	91.0	86.3	571.79	....	32.56	4.35	131.5	166.4	60	155.7	1053.84	123.6	9.92	107.5	75	359.66	23.7	9.2
10	5:45	693.62	91.0	86.0	603.51	....	32.56	4.35	138.6	155.0	62	156.4	1052.22	121.2	9.92	113.5	79	361.28	23.8	8.8
11	6:15	632.16	92.0	86.3	555.75	....	32.56	4.35	127.7	160.0	60	150.0	1167.0	118.3	10.03	103.4	72	346.5	22.8	9.7
12	6:45	649.72	91.6	85.7	569.34	4 ft. 11½ in.	32.56	4.35	130.6	163.0	60	155.0	1155.45	123.1	9.92	107.04	74	358.05	23.6	9.3

TABLE 1 TEST OF WELL No. 32—Continued

New System, November 6, 1907

Number	Time	Cu. Ft. of Free Air per Minute (Displacement)	Volumetric per Cent Efficiency	Temperature of Air Degrees Fahr.	Cu. Ft. of Free Air per Minute (Actual)	Fluid Level in Tank	Gallons per Minute	Cu. Ft. of Fluid per Minute	Ratio	Air Pressures: Discharge	Air Pressures: Intercooler	Air Pressures: At Well	Pumping Head	I. H. P.	W. H. P.	A. H. P.	R. P. M.	Submergence in Feet	Submergence per Cent	Pumping Efficiency per Cent
1	12:00	448.4	92.0	82.0	395.8	1 ft. 6 in.	36.47	4.87	81.2	207	77	202	1076.5	91.5	10.3	79.6	76	416.5	27.9	13.0
2	12:30	442.5	91.5	82.0	388.4	....	36.47	4.87	79.8	207	77	202	1076.5	89.8	10.3	78.1	75	416.5	27.9	13.2
3	1:00	436.6	90.0	82.0	377.01	....	36.47	4.87	77.5	205	77	200	1081.1	86.7	10.4	75.4	74	411.19	27.6	13.8
4	1:30	436.6	90.0	82.0	377.01	....	36.47	4.87	77.5	205	77	200	1081.1	86.7	10.4	75.4	74	411.19	27.6	13.8
5	2:00	442.5	91.5	81.3	388.9	....	36.47	4.87	79.9	205	77	200	1081.1	89.5	10.4	77.8	75	411.19	27.6	13.3
6	2:30	442.5	91.5	81.5	388.8	....	36.47	4.87	79.9	205	77	200	1081.1	89.4	10.4	77.7	75	411.19	27.6	13.3
7	3:00	448.4	92.0	81.6	396.1	....	36.47	4.87	81.4	210	77	205	1069.6	92.5	10.3	80.4	76	423.4	28.4	12.8
8	3:30	442.5	91.5	81.2	389.1	....	36.47	4.87	80.0	207	77	202	1076.5	89.9	10.3	78.7	75	416.5	27.9	13.2
9	4:00	442.5	91.5	81.3	388.9	....	36.47	4.87	79.9	210	77	205	1069.6	90.7	10.3	78.9	75	423.4	28.4	13.1
10	4:30	436.6	90.5	81.0	379.7	....	36.47	4.87	77.8	205	77	200	1081.1	87.3	10.4	75.9	74	411.9	27.6	13.7
11	5:00	436.6	91.2	81.0	382.7	....	36.47	4.87	78.6	205	77	200	1081.1	87.9	10.4	76.5	74	411.9	27.6	13.6
12	5:30	442.5	90.2	80.5	383.9	....	36.47	4.87	78.1	205	77	200	1081.1	88.5	10.4	76.8	75	411.9	27.6	13.5
13	6:00	442.5	91.0	80.5	387.4	5 ft. 1¼ in.	36.47	4.87	79.5	205	77	200	1081.1	89.1	10.4	77.5	75	411.9	27.6	13.4

TABLE 2 TEST OF WELLS NO. 12, 30, AND 32

CROWLEY OIL AND MINERAL CO., NOVEMBER 13, 1907

NEW SYSTEM INSTALLED IN EACH

Number	Time	Cu. Ft. of Free Air per Minute (Displacement)	Volumetric Efficiency (Mean)	Temperature of Intake Air, Degrees Fahr.	Cu. Ft. of Free Air per Minute (Actual)	Gallons per Minute No. 12	Gallons per Minute No. 30	Gallons per Minute No. 32	Total Gallons per Minute	Cu. Ft. of Fluid per Minute	Ratio	Air Pressures Discharge	Air Pressures Intercooler	Air Pressures At Well No. 12	Air Pressures At Well No. 30	Air Pressures At Well No. 32
1	11:00	719.9	....	76.5	628.02	39.2	33.2	30.4	102.8	13.74	45.7	217	65	128	202	205
2	11:30	719.9	....	76.5	628.02	39.2	33.2	30.4	102.8	13.74	45.7	215	65	128	205	202
3	12:00	710.2	....	76.7	619.3	39.2	33.2	30.4	102.8	13.74	45.1	215	65	128	205	200
4	12:30	702.4	....	76.7	612.3	39.2	33.2	30.4	102.8	13.74	44.5	215	65	128	202	200
5	1:00	684.8	....	77.0	584.04	39.2	33.2	30.4	102.8	13.74	42.5	215	65	128	202	202
6	1:30	693.6	....	77.0	604.5	39.2	33.2	30.4	102.8	13.74	43.9	215	65	133	202	202
7	2:00	719.9	90	77.0	627.4	39.2	33.2	30.4	102.8	13.74	45.6	217	65	128	205	205
8	2:30	693.6	....	76.5	605.7	39.2	33.2	30.4	102.8	13.74	44.1	215	65	128	202	200
9	3:00	710.2	....	76.5	619.7	39.2	33.2	30.4	102.8	13.74	45.1	215	65	128	202	200
10	3:30	702.4	....	76.0	613.3	39.2	33.2	30.4	102.8	13.74	44.6	215	65	123	202	202
11	4:00	702.4	....	76.0	613.3	39.2	33.2	30.4	102.8	13.74	44.6	215	65	128	205	205
12	4:30	710.2	....	75.5	620.7	39.2	33.2	30.4	102.8	13.74	45.2	215	65	128	202	202
13	5:00	702.4	....	75.5	613.9	39.2	33.2	30.4	102.8	13.74	44.7	215	65	128	202	202

Number	Gallons per Minute per 100 Cu. Ft. of Free Air	I. H. P.	W. H. P. No. 12	W. H. P. No. 30	W. H. P. No. 32	Total W. H. P.	R. P. M.	A. H. P.	Per Cent Submergence No. 12	Per Cent Submergence No. 30	Per Cent Submergence No. 32	Pumping Heads No. 12	Pumping Heads No. 30	Pumping Heads No. 32	Pumping Efficiency
1	16.4	155.7	7.1	9.4	8.6	25.1	82	132.3	25.9	28.0	28.4	696.9	1077.7	1069.6	18.9
2	16.4	155.1	7.1	9.3	8.65	25.05	82	131.9	25.9	28.5	27.9	696.9	1070.8	1076.5	19.0
3	16.6	152.9	7.1	9.3	8.7	25.1	81	129.9	25.9	28.5	27.6	696.9	1070.8	1081.1	19.3
4	16.8	151.2	7.1	9.4	8.7	25.2	80	128.5	25.9	28.0	27.6	696.9	1077.7	1081.1	19.5
5	17.6	144.3	7.1	9.4	8.65	25.15	78	122.5	25.9	28.0	27.9	696.9	1077.7	1076.5	20.4
6	17.0	149.3	6.9	9.4	8.65	24.95	79	126.9	26.9	28.0	27.9	685.5	1077.7	1076.5	19.7
7	16.4	155.6	7.1	9.3	8.6	25.0	82	132.3	25.9	28.5	28.4	696.9	1070.8	1069.6	18.9
8	16.9	149.6	7.1	9.4	8.7	25.1	79	127.0	25.9	28.0	27.6	696.9	1077.7	1081.1	19.7
9	16.6	153.1	7.1	9.4	8.7	25.2	81	130.0	25.9	28.0	27.6	696.9	1077.7	1081.1	19.3
10	16.5	151.5	7.2	9.4	8.65	25.25	80	128.7	24.5	28.0	27.9	708.5	1077.7	1076.5	19.5
11	16.8	151.5	7.2	9.3	8.65	25.1	80	128.7	25.9	28.5	28.4	696.9	1070.8	1069.6	19.5
12	16.6	153.3	7.2	9.4	8.6	25.25	81	130.3	25.9	28.0	27.9	696.9	1077.7	1076.5	19.3
13	16.8	151.5	7.2	9.4	8.65	25.25	80	128.7	25.9	28.0	27.9	696.9	1077.7	1076.5	19.5

TABLE 3 TEST OF HILL NO. 2

MAMOU POWER CO., EVANGELINE, LA.

Old System, February 27, 1908

Number	Time	Cu. Ft. of Free Air per Minute (Displacement)	Volumetric Efficiency	Temperature of Air Degrees Fahr.	Cu. Ft. of Free Air per Minute (Actual)	Depth of Fluid over Weir, Cm.	Gallons per Minute	Cu. Ft. of Fluid per Minute	Ratio	Air Pressures: Discharge	Air Pressures: Intercooler	Air Pressures: At Well	Pumping Head	I. H. P.	W. H. P.	A. H. P.	R. P. M.	Submergence Feet	Submergence, per Cent	Pumping Efficiency per Cent
1	7:45	413.6	85.2	75.1	342.46	1.71	41.7	5.57	61.5	260	80	252.5	916.72	91.8	10.1	75.3	88	583.28	38.8	13.4
2	8:45	441.8	84.7	77.5	362.04	1.72	41.95	5.61	64.5	260	80	252.5	916.72	93.3	10.2	75.3	94	583.28	38.8	13.5
3	9:45	498.2	85.0	77.8	409.51	1.71	41.7	5.57	73.4	255	80	247.4	866.14	108.9	9.6	89.4	106	633.86	42.3	10.7
4	10:45	470.0	85.0	78.2	386.01	1.70	41.46	5.54	68.5	255	80	247.4	866.14	98.8	9.5	81.5	100	633.86	42.3	11.6
5	11:45	526.4	86.1	78.7	436.53	1.72	41.95	5.61	77.9	255	80	247.4	866.14	115.7	9.6	94.5	112	633.86	42.3	10.2
6	12:45	462.2	85.3	78.8	380.52	1.71	41.7	5.57	69.1	255	80	247.4	866.14	101.2	9.6	82.3	98	633.86	42.3	11.7
7	1:45	470.0	84.5	81.3	381.55	1.71	41.7	5.57	68.5	255	80	247.4	866.14	101.5	9.6	81.7	100	633.86	42.3	11.7
8	2:45	432.4	84.5	81.6	350.83	1.71	41.7	5.57	62.9	260	82	253.0	915.57	95.5	10.1	77.5	92	584.43	38.9	13.0
9	3:45	441.8	84.7	77.5	362.04	1.71	41.7	5.57	64.9	260	82	252.5	916.72	93.3	10.1	76.4	94	583.28	38.8	13.2
10	4:45	441.8	84.7	76.2	362.04	1.70	41.46	5.54	65.4	260	82	252.5	916.72	93.3	10.0	76.3	94	583.28	38.8	13.1
11	5:45	535.8	85.1	75.0	443.21	1.70	41.46	5.54	80.0	260	82	252.5	916.72	97.1	10.0	97.1	114	583.28	38.8	10.3

New System, March 3, 1908

1	7:20	282.0	86.3	72.3	237.80	2.09	50.98	6.81	34.9	220	80	213.5	996.32	59.7	13.5	48.5	60	493.18	33.1	27.8
2	8:20	282.0	86.3	72.3	237.80	2.09	50.98	6.81	34.9	226	80	218.3	985.23	60.5	13.3	49.2	60	504.27	33.8	27.1
3	9:20	291.4	85.7	73.5	243.52	2.10	51.22	6.84	35.6	220	80	214.0	995.16	61.3	13.5	49.5	62	494.34	33.2	27.4
4	10:20	282.0	86.3	73.8	237.08	2.08	50.73	6.78	34.9	220	80	214.0	995.16	59.7	13.4	47.6	60	494.34	33.2	28.2
5	11:20	291.4	85.7	74.2	243.10	2.08	50.73	6.78	35.8	220	80	214.0	995.16	61.1	13.4	48.8	62	494.34	33.2	27.6
6	12:20	338.4	86.1	75.0	283.21	2.09	50.98	6.81	41.6	220	80	214.0	995.16	71.3	13.4	57.1	72	494.34	33.2	23.6
7	1:20	310.2	86.5	76.1	260.27	2.09	50.98	6.81	38.2	225	80	217.5	987.07	65.8	13.3	53.2	66	502.43	33.07	25.0
8	2:20	310.2	86.5	78.3	259.21	2.08	50.73	6.78	38.2	225	80	217.5	987.07	65.5	13.2	51.7	66	502.43	33.07	25.9
9	3:20	291.4	85.6	77.1	241.98	2.09	50.98	6.81	35.5	225	80	217.5	987.07	61.2	13.3	49.3	62	502.43	33.07	27.1
10	4:20	282.0	86.3	76.3	234.12	2.10	51.22	6.84	34.2	220	80	214.0	995.16	58.8	13.5	47.3	60	494.34	33.2	28.6
11	5:20	310.2	86.5	72.8	261.88	2.09	50.98	6.81	38.4	220	80	214.0	995.16	65.7	13.4	52.2	66	494.34	33.2	26.2

DISCUSSION

F. A. HALSEY. The air-lift pump is one of those things of which our knowledge is almost exclusively experimental. Its analysis presents such serious difficulties that rational equations for the performance of the apparatus have not been derived by anyone to my knowledge. Under these circumstances, we are reduced to experiment for the determination of the fundamental laws of the performance of the apparatus, and these experiments should cover a wide range in the conditions which lead to variation in the performance of apparatus, namely, depth of submergence and lift.

2 The most complete experiments now on record were presented in a paper before the British Institution of Civil Engineers in 1906,* these experiments being of sufficiently wide scope to supply a satisfactory guide for the design of these pumps under a considerable range of conditions.

3 One curious feature attends the behavior of the air-lift pump and the Taylor hydraulic air compressor, which are essentially the same apparatus, reversed in action. In each case we have a pair of vertical pipes communicating at the bottom, one filled with water and the other with a mixture of air and water—a sort of suds. If the pipes are of indefinite length the columns will take levels corresponding with their respective gravities, but if the suds pipe is cut off below this level it will overflow, the suds will rise, and we have an air-lift pump; while on the contrary, if the water pipe is cut off below this level, it will overflow, the column of water will rise, and we have a Taylor hydraulic air compressor. The point to which I refer is that while the first of these constructions has a low, the second has a high efficiency. The highest figure given in this paper for the efficiency of the pump is about 28 per cent, and while 40 per cent has been reached in exceptional cases, the average is probably not more than 20 per cent. On the other hand, the efficiency of the hydraulic air compressor is in the vicinity of 75 or 80 per cent.

DR. SANFORD A. MOSS. The method used by the author for comput tion of volumetric efficiency is subject to serious inaccuracies, as points out. For the purpose of the tests in question, where relat rather than absolute volumetric efficiencies are desired, the met is quite proper. The volumetric efficiencies given must be underst to be relative and not absolute, however, as the errors may be c

*Abstracted in the American Machinist, August 16, 1906.

serious. I have used the method which the author states he would have preferred for ascertaining volumetric efficiency, Par. 23, and found it very good. Everything operated as would be expected and a reputable value of volumetric efficiency was obtained. In one case, the method as given in Par. 23 was checked by use of calibrated orifices and also by use of a single tank with rise of pressure, the rate of increase of pressure when the pressure reached the rated value giving the desired volumetric efficiency. All three methods checked very closely. The volumetric efficiency in this case was 58 per cent, showing the serious losses which can occur due to the inaccuracies mentioned.

J. G. CALLAN.[1] Par. 21 to Par. 23, with diagram (Fig. 7), refer to the method of determining the efficiency of the air compressor, and Mr. Ivens states in Par. 23 the method which he would have preferred for determining this quantity.

2 The method of pumping up pressure in a large tank is open only to the objection that the large tank is rarely obtainable, particularly when the pressures which it is proposed to use are high, and the volume large. A more available and almost equally exact method is afforded by the use of calibrated orifices of which the discharge coefficient has been duly determined. These orifices can be used in connection with well-known formulæ, which involve a knowledge of corrected pressure and temperature of air delivered to orifice, and of velocity and temperature of emergent jet; all factors which can be readily and accurately determined.

3 The losses of a compressor are so considerable and so complex that the assumption of volumetric efficiency from the indicator card may be in error by extremely large percentages and is quite as likely to be misleading as enlightening, unless results are strictly comparative rather than absolute. The value of the present tests would apparently be somewhat affected by the errors in this method of determining delivery, since certain elements of compressor loss would assume different percentage values with different discharge rates.

4 This brings up the desirability of some well-understood standardized method of testing compressors and other apparatus for delivering air. Commercial loss is certain to arise from the considerable variations in terminology, and a definition of terms by an authoritative body such as this Society would be of marked advantage. For example, volumetric efficiency undoubtedly should mean the ratio

[1] General Electric Co., Schenectady, N. Y.

between the air actually discharged (when restored to standard atmospheric conditions of temperature and pressure) to the air which would have been discharged had the compressor cylinder been completely filled on each stroke with air under standard atmospheric conditions, and had all the air displaced been completely discharged. The exact method of determining the approximation of real to theoretical discharge obviously should be only a secondary consideration in such a definition, but could properly be taken up as corollary.

5 The segregation and determination of losses in air compressors of different types form an interesting and useful line of investigation upon which little has been published. A considerable amount of work with which I am familiar indicates that these losses are usually materially greater than the customary assumptions, particularly in compressors which have been in service for some time. Various methods have been devised for independently estimating loss due to heating of intake, leakage of inlet, discharge valves and piston, clearance loss, throttling, and losses due to improper setting of mechanical valve gears, but usually these determinations will involve more labor than a direct measurement of air output by orifice and impact tube. It is my belief, however, that the computation which neglects all of these losses except the clearance loss, particularly where the pressure is high and the compressor is somewhat worn, is almost sure to be very gravely in error.

THE AUTHOR. The discussions of the paper on oil-well pumping presented at the Washington meeting were centered on the method of ascertaining volumetric efficiency of air compressors. The object of the paper was rather to compare the relative efficiencies of various air-lift systems when being operated on extremely high pumping heads. However, as all the discussions were agreeable to the writer's views and methods, there is no room for argument.

No. 1244

THE SPECIFIC VOLUME OF SATURATED STEAM

By Prof. C. H. Peabody,[1] Boston, Mass.

Non-Member

For many years the specific volume of saturated steam has been computed from the thermodynamic equation

$$s = \frac{r}{AT} \cdot \frac{1}{\frac{dp}{dt}} + \sigma$$

in which the quantities have the following significance:

s is the specific volume, for example the volume in cubic meters of one kilogram.

r is the heat of vaporization in calories.

A is the heat equivalent of a unit of work.

T is the absolute temperature obtained by adding 273 to the temperature by the centigrade thermometer.

$\frac{dp}{dt}$ is the differential coefficient of the pressure with regard to the temperature, the pressure being in kilograms per square meter.

σ is the specific volume of water (0.01 cubic meters per kg.)

2 For this paper French units are used because the original data are given in them and comparison with experimental values is convenient.

3 All the quantities entering into this equation are now determined with a certainty and precision that must be considered satisfactory for engineering purposes and a comparison with experimental determinations of the specific volume shows an exceptionally good concordance.

[1] Professor Naval Architecture and Marine Engineering. Mass. Inst. Tech.

Presented at the Spring Meeting, Washington, May 1909, of The American Society of Mechanical Engineers.

4 To make the exposition of this statement clear it is necessary to review the experimental data and to state the precision that can properly be attributed to them.

5 The mechanical equivalent of heat as determined by Rowland[1] may be taken as 427 meter-kilograms (778 foot pounds) at 15 deg. cent., which corresponds nearly with 62 deg. fahr. There have been more recent investigations which on the whole confirm this result, though there is some indication that it is a trifle small. The uncertainty may be one in a thousand or one in two thousand.

6 Callendar[2] gives for the absolute temperature of freezing point 273.1 deg. cent., with a probable error of one in two thousand.

7 For the range of temperature from 30 deg. to 100 deg. Henning[3] gives the equation

$$r = 94.210\ (365 - t)^{0.31249}$$

in calories at 15 deg. cent. In English units the equation may be written

$$r = 141.124\ (689 - t)^{0.31249}$$

Experiments by Dieterici,[4] Griffiths[5] and A. C. Smith[6] confirm his results and extend the equation to freezing point. The probable error of this equation is one in one thousand.

8 In his paper, The Total Heat of Saturated Steam, read at the Annual Meeting, 1908, Dr. Harvey N. Davis gives for the total heat of steam from 212 deg. to 400 deg. fahr.

$$H = H_{212} + 0.3745\ (t - 212) - 0.000550\ (t - 212)^2$$

Transformed into French units this may be written

$$H = 638.9 + 0.3745\ (t - 100) - 0.00099\ (t - 100)^2$$

provided that the constant term be taken as the sum of Henning's value for r at 100 deg. cent. and the heat of the liquid be taken as 100.2, according to a consideration to be taken up later in this paper. To conform with the conditions already accepted, this equation should give the total heat in calories at 15 deg. cent., while Dr. Davis used for the calories 1/100 of the heat required to raise one kilogram

[1] Proc. Am. Acad., vol. 15 (n.s. 7), 1879.
[2] Phil. Mag., Jan. 1903.
[3] Annalen der Physik, vol. 21, p. 849, 1906.
[4] Annalen der Physik, vol. 16, p. 912, 1905.
[5] Phil. Trans., 186, p. 261, 1895.
[6] Physical Review, vol. 25, 1907.

of water from freezing to boiling point. The difference amounts to 2/1000, as indicated by the heat of the liquid just mentioned ($q = 100.2$). Now the total heats at 100 deg. and 200 deg. cent. are 638.9 and 666.5, and their difference is 27.6 calories, so that the total effect is less than one-tenth of a calorie.

9 As for the heat of the liquid we have the three following sources of information:

a Barnes'[1] determinations of the specific heat of water from 0 deg. to 95 deg. cent.

b Dieterici's[2] determinations of the same property from freezing point to very high temperatures.

c Regnault's[3] determinations of the heat of the liquid.

Barnes' experiments were made by an electrical method for which great relative precision is claimed, and they showed a good concordance with Rowland's work on the mechanical equivalent, which in reality was an investigation also of the specific heat. Dieterici's investigation consisted essentially in heating water in a quartz tube, which was then transferred to the ice calorimeter. His results appear to be systematically larger than Barnes'; at 95 deg. cent., the discrepancy is $\frac{4}{10}$ of 1 per cent.

10 In 1907 the author endeavored to join Regnault's values for the heat of the liquid to those deduced from Barnes' values of the specific heat. Now Regnault's experiments consisted in running hot water into a calorimeter partly filled with cold water and noting the rise of temperature in the calorimeter. There were 40 tests in all, cattered irregularly from about 100 deg. to 190 deg. cent. for the temperature of the hot water; there were in a way three groups of tests, one near 110 deg., one near 160 deg., and the third near 190 deg. cent.

11 The average rise of temperature in the calorimeter for the first group was not far from 9 deg. cent., which item appears to account for the considerable irregularity of results at that place. The experiments with the highest temperatures had nearly twice that rise of temperature in the calorimeter and about half the dispersion of results.

12 In order to use Regnault's results his values for the heat of the liquid were recomputed, allowing for the true specific heat of the water in the calorimeter, and then a diagram was plotted as shown

[1] Phys. Review, vol. 15, p. 71, 1902.
[2] Annalen der Physik, vol. 16, p. 593, 1905.
[3] Memoirs de l'Institut de France, vol. 26.

by Fig. 1, in which the abscissae are temperatures and the ordinates are values of $q - t$.

13 This allows of the use of a large vertical scale which much accentuates the apparent scattering of points. A curve was then drawn to join a curve from 0 deg. to 100 deg. cent., from Barnes' results for the specific heat of water. This curve passes near the highest group of points, above the middle group and below the lowest group.

14 It should be said that Barnes' results were first transformed to allow for the use of 62 deg. fahr. for the standard temperature, instead of 20 deg., which he had taken in his report; also that his

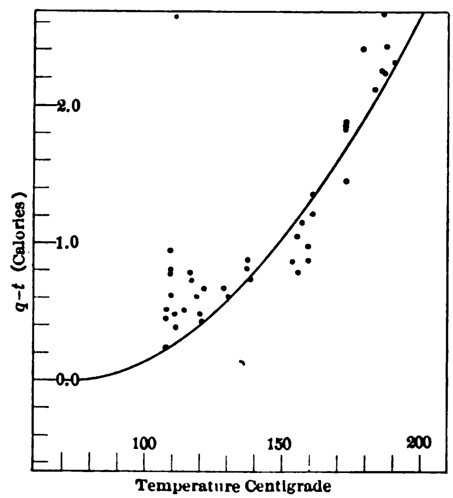

FIG. 1 RECOMPUTATION OF REGNAULT'S EXPERIMENTS ON THE HEAT OF THE LIQUID OF WATER

values were slightly increased at temperatures approaching 100 deg. so as to avoid a break in the curve. The last had the effect of increasing the heat of the liquid at 100 deg. by one one-thousandth.

15 Finally a table of specific heats was drawn off for temperatures from 0 deg. to 220 deg. cent., which served as the basis of a graphical integration for the value of $q - t$. Fig. 2 gives the curve representing the final value of this quantity and also a curve representing values that would be obtained if Dieterici's values for the specific heat were accepted.

16 The author is of the opinion that the full curve in Fig. 2 shows very nearly the true value of the property under consideration, and he has used it to determine heats of the liquid.

17 The maximum deviation of a single point from the curve in Fig. 1 is 0.8 of a calorie, which amounts to $\frac{3}{4}$ of 1 per cent of the heat of the liquid at that point. If we could consider that an error of 0.02 deg. might be attributed to the temperatures in the calorimeter it would account for one-third of that deviation. But to take the most pessimistic view of the situation and charge an error of 0.8 of a calorie against the method, we may still consider that for temperatures above boiling-point the heat of the liquid is always associated with the heat of vaporization, and that their sum is more than

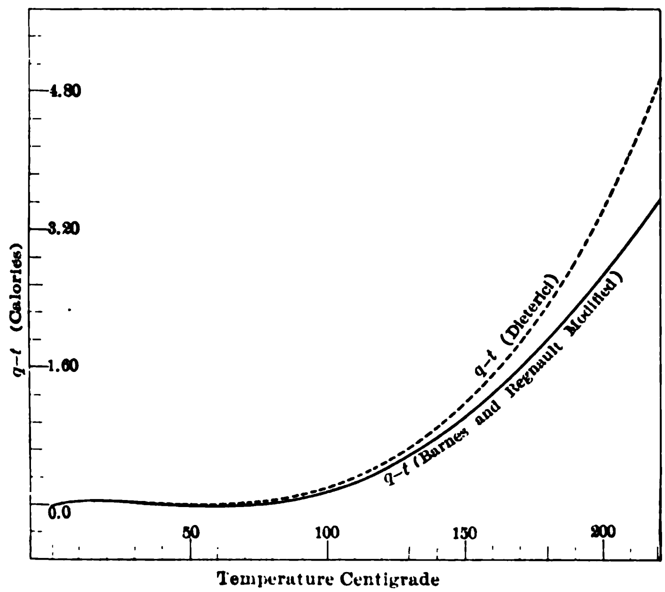

FIG. 2 VALUES OF THE QUANTITY $(q-t)$

THE FULL CURVE SHOWS THE QUANTITY DEDUCED FROM THE AUTHOR'S COMBINATION OF BARNES' EXPERIMENTS ON THE SPECIFIC HEAT OF WATER WITH REGNAULT'S EXPERIMENTS ON THE HEAT OF THE LIQUID, WHILE THE DOTTED CURVE SHOWS RESULTS FROM DIETERICI'S EXPERIMENTS ON THE SPECIFIC HEAT OF WATER.

630 calories, so that the deviation in this light amounts to $\frac{1}{8}$ of 1 per cent.

18 A more just view is clearly to take the deviation of the worst group of points. This occurs at 117 deg. and is about 0.3 of a calorie, that is, 0.25 per cent of the heat of the liquid. The most favorable view is to consider that the upper end of the curve is well fixed by Regnault's experiments, which were then under the most favorable conditions, and that the lower end is tied to Barnes' values, which have all desired precision. This matter is discussed with some detail be-

cause the original experimental results needed to be entirely recast for the present purpose.

19 But while important from some aspects, the quantities with which we are dealing are not affected by uncertainties that concern our main investigation, i.e., the specific volume of saturated steam, for the maximum variation between the author's value for the heat of the liquid, and a value determined from Dieterici's investigation, amounts to 0.8 of a calorie at 200 deg. cent. This is only $\frac{1}{8}$ of 1 per cent of the total heat at that place. However, we need for our specific volume the heat of vaporization, and the discrepancy then becomes $\frac{1}{5}$ of 1 per cent.

20 Recent determinations of the pressure of saturated steam have been made by Holborn and Henning,[1] with all the resources of modern physical methods including the platinum thermometer. They claim a precision of 0.01 deg. in the determination of temperature and that

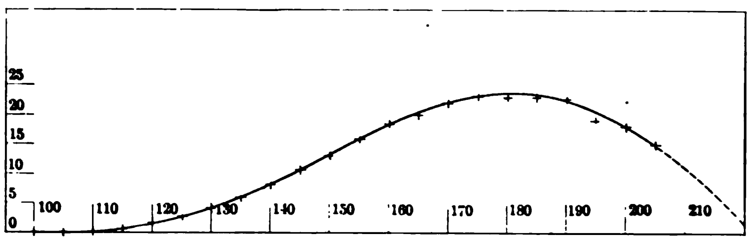

FIG. 3 CURVE TO EXTRAPOLATE PRESSURE OF SATURATED STEAM TO 220 DEG CENT.

their results reduced to the thermometric scale have a probable error of not more than 0.02 deg. at 200 deg. cent. Their own experiments cover the range of temperature from 50 deg. to 200 deg. cent. (122 deg. to 392 deg. fahr.), and they have extrapolated results to 205 deg. cent. Below 30 deg. they have made use of experiments by Thiesen and Scheel to extend results to freezing points; these experiments were not made with the same degree of precision as those by Holborn and Henning.

21 In order to extend calculations to 220 deg. cent., as has been the habit in computing steam tables, the author made use of a diagram shown by Fig. 3, in which the abscissae are temperatures centi-

[1] Annalen der Physik, vol. 26, p. 383, 1908.

NOTE. Since these results may not be easily accessible, it may be of interest to say that they have been transferred directly to Table 3, of the author's Steam and Entropy Tables, edition of 1909.

grade and the ordinates are differences between Holborn and Henning's value and pressures computed by the following equation:

$$\log p = 5.457570 - 0.4120021\,(9.997411296 - 10)^{t-100} + (7.74168 - 10)\,(9.997411296 - 10)^{t-100}$$

which was chosen as a matter of convenience and because it gave a curve which crossed the axis near 220 deg. cent. when produced. It is thought that the extrapolated values are not much in error, though there is no means of determining this question. Fortunately this part of the range of temperature, as well as that below 30 deg. cent., is not so important to engineers.

22 The degree of precision attained by Holborn and Henning in the determination of the pressure of saturated steam is far beyond any direct technical requirement, since pressures are seldom determined closer than one-tenth of a pound; it is, however, requisite, if the differential coefficient $\frac{dp}{dt}$ is to be determined with certainty and accuracy.

23 Since their results are presented in a table without attempting to represent it by an equation, it becomes necessary to replace $\frac{dp}{dt}$ by $\frac{\Delta p}{\Delta t}$ which can be most readily obtained as follows: for a given temperature, for example 100 deg., we may compute the ratio by taking two adjacent temperatures, such as 98 deg. and 102 deg., finding the difference of pressure, which is to be divided by the difference of temperature; and the result is to be multiplied by 13.5959, because that is the pressure of one millimeter of mercury on one square meter. This result is

$$\frac{\Delta p}{\Delta t} = 13.5959\,\frac{815.9 - 707.3}{4} = 369.1$$

24 A number of elements entered into the determination to use this method and to take an interval of 4 deg. If the relation of the pressure to the temperature could be represented by a second-degree curve, that is, if such a curve were a parabola with its axis parallel to the axis of pressure, the ratio $\frac{\Delta p}{\Delta t}$ for any interval would be precisely equal to $\frac{dp}{dt}$. A table of values that could be represented by such a curve would have constant second differences; by second differences are meant the results obtained by taking (*a*) the differences

of successive tabular values, and (*b*) the differences of these differences. An examination of the second differences of Holborn and Henning's values showed great regularity between 50 deg. and 100 deg., i.e., for their own determinations. The second differences increased slowly; for intervals of 4 deg. the increase was imperceptible, for 6-deg. intervals the increase was barely perceptible, but for 10-deg. intervals it was very apparent.

25 Now the possible precision of reading the height of a column of mercury, including allowance for variations of density, is better than the determination of temperature; consequently the probable error to be considered is that attributed to the determination of temperature, namely 0.01 deg., consequently the probable error of a single determination of the ratio $\frac{\Delta p}{\Delta t}$. To diminish the effect of local variations this ratio was computed for each degree of temperature and the regularity of the results thus obtained was tested by taking first and second differences. Where the second differences showed irregularity, the values of the ratio were changed to the extent of 1/1000 in order to improve the regularity of the second differences. This process is equivalent to drawing a smooth or fair curve to represent physical properties obtained by observation.

26 Having values of the ratio $\frac{\Delta p}{\Delta t}$ for each degree of temperature the specific volumes were computed by the thermodynamic equation in Par. 1. They were in turn tested for regularity by taking first and second differences: and again the values were changed when necessary to the extent of 1/1000 to improve the regularity of the second differences. The combined effect of both fairings is estimated not to exceed 1/500 in any case and the author believes that the probable error of the final determinations of the specific volumes is not greater than that amount for the range of 50 deg. to 200 deg. cent.

27 It may further be said that having computed the values of Apu at each fifth degree and plotted the results on a large diagram, no individual values were found to vary from a fair curve more than 1/750.

28 Fortunately there are extant experiments on the specific volume of saturated steam by Knoblauch, Linde and Klebe,[1] made with such a degree of precision as to give a satisfactory check on the computations made by the method described. These experiments

[1]*Mitteilungen uber Forschungsarbeiten*, vol. 21, S. 33, 1905.

consisted in measuring the temperature and pressure of superheated steam at constant volume, and the results were so treated as to give the volume at saturation by a straight-line extrapolation with great certainty. The experimenters give the following equation to represent the properties of both superheated and saturated steam:

$$p\,v = BT - p\,(1+a\,p)\left[C\left(\frac{373^3}{T}\right)-D\right]$$

$B = 47.10$; $a = 0.000002$; $C = 0.031$; $D = 0.0052$, volumes being in cubic meters per kilogram, pressures in kilograms per square meter, and the absolute temperature being on the centigrade scale.

29 For English units the equation may be written

$$p\,v = 85.85\,T - p\,(1+0.00000976\,p)\left[\frac{150{,}300{,}000}{T^3} - 0.0833\right]$$

the volumes being in cubic feet, the pressures in pounds per square foot and the temperatures in degrees fahr.

30 Knoblauch claims for this equation a mean probable error of 1/500, though admitting individual discrepancies of twice that amount. This equation applied to the computation of specific volumes of saturated steam shows a good concordance with results, computed by the thermodynamic equation, the greatest discrepancy being 1/300 at 165 deg. cent. (329 deg. fahr.).

31 Not satisfied with this apparent concordance, which after all was with an empirical equation which on examination showed somewhat larger variation from individual experimental values at saturation, the author had a diagram drawn of the 32 values of the specific volume reported by the experimenters. The diagram was drawn to a very large scale, using temperatures for abscissae and logarithms of volumes for ordinates, and a fair curve was drawn by aid of a stiff spline. From readings on this curve the volumes were determined at 5 deg. intervals, and are set down in the accompanying table together with values computed by the thermodynamic equation.

32 The greatest deviation of values in this table is 0.2 per cent, which is precisely the probable error assigned by the experimenters for their work. It may therefore be concluded that between the limits of temperature in this table and probably from 30 deg. to 200 deg. cent. (86 deg. to 392 deg. fahr.), the probable error of computations by aid of the thermodynamic equation is not in excess of 1/500.

COMPARISON OF EXPERIMENTAL AND COMPUTED VALUES OF THE SPECIFIC VOLUME OF SATURATED STEAM

Temperature	Volume, Cubic Meters			Temperature	Volume, Cubic Meters		
	Experimental	Computed	Per cent deviation		Experimental	Computed	Per cent deviation
100	1.674	1.671	+0.18	145	0.4458	0.4457	+0.02
105	1.421	1.419	+0.14	150	0.3927	0.3921	+0.15
110	1.211	1.209	+0.17	155	0.3466	0.3463	+0.09
115	1.036	1.036	0.	160	0.3069	0.3063	+0.20
120	0.8894	0.8910	−0.18	165	0.2724	0.2729	+0.18
125	0.7688	0.7698	−0.13	170	0.2426	0.2428	+0.12
130	0.6670	0.6677	−0.10	175	0.2168	0.2164	+0.19
135	0.5809	0.5812	−0.05	180	0.1940	0.1941	−0.05
140	0.5080	0.5081	−0.02				

33 This conclusion carries with it the attribution of at least the same degree of precision to all the properties entering into the thermodynamic equation. A little consideration will show that this conclusion covers all the properties given in steam-tables including the entropy. As an apparent exception we have the heat of the liquid at high temperatures which may be uncertain to the extent of $\frac{1}{2}$ of 1 per cent of itself, but as that quantity is then associated with the heat of vaporization the influence of such an error will be of no consequence in computations.

34 It may therefore be expected that steam tables based on the present information will have permanence.

DISCUSSION

Prof. William D. Ennis. The reason for the extrapolation of Fig. 3 is not quite clear to me. The ordinates of this diagram are differences between the Holborn and Henning values for the pressure of saturated steam, and those given by the Peabody formula, $\log p = a - b\,\alpha^{t-100} + c\,\beta^{t-100}$. The diagram is extended to include temperatures above 205 deg., the Holborn and Henning limit. Why would it not be just as satisfactory, if the Holborn and Henning values are satisfactory, to extrapolate directly the curve expressing their results?

2 There seems to be little room for uncertainty in any of the properties of saturated steam, excepting, possibly, the heat of the liquid. The maximum divergence in values for the former, comparing the Dieterici and the modified Regnault values adopted by Professor

Peabody, occurs at the highest temperatures: at 220 deg. (an extrapolated point) it is 1.31 cal. or 0.584 per cent. Now if Dieterici does not claim an accuracy exceeding 0.5 per cent at this temperature, and since Professor Peabody admits a possible fractional percentage of error, the true value may be within the limit of estimated error in both computations. The result of taking Dieterici's values would be to make the computed specific volume 0.01 per cent less at 100 deg., 0.02 per cent less at 140 deg. and 0.045 per cent less at 165 deg., than those tabulated by the author. The deviation from the Knoblauch, Linde and Klebe results would be thus generally decreased. There is still a possibility that the Dieterici values may be more nearly correct. If the preponderance of error in values of the other quantities entering into the volume formula were in such a direction as to make the computed volumes too small, the lower heats of the liquid used by Professor Peabody might lead to an apparently better result because of a balancing of opposite errors. It does not seem safe to say definitely that such may not be the fact. The uncertainty in the heat of the liquid at 200 deg. is 0.36 per cent rather than 0.25 per cent. The same uncertainty applies to the entropy of the liquid, and a possible error of about 0.16 per cent to the entropy of vaporization. The entropy of saturation at this temperature may then be wrong to as great an extent as 0.23 per cent, or $\frac{1}{25}$. This would introduce a barely noticeable error into computations involving vapor cycles.

3 It is questionable whether permanence, in a matter of this kind, is as desirable as a standard, a flexible standard. Would it not be within the scope of the Society to coöperate with national engineering organizations abroad in the preparation of an international steam table for saturation and superheat, embodying the generally accepted values and subject to modification whenever an undisputed conclusion is reached on the one or two remaining doubtful quantities?

The Author. In reply to Professor Ennis I will first explain that Holborn and Henning give a table of pressure for each degree of temperature, and make no use of an equation except as a means of fairing their results, for which purpose they chose Thiesen's equation which gives divergent values for higher temperatures. Now it happens that the equation which I chose gives values which converge toward Holborn and Henning's values so that it is possible to draw an extrapolation diagram as shown in my paper.

2 Secondly, I wish to say that Dr. Henning has kindly sent me in advance of publication an abstract of results which he has recently

obtained for the heat of vaporization of water from 100 deg. to 180 deg. cent. His memoir will soon be available and will show that the results which I have deduced from Dr. Davis' values of the total heat, show a close concordance with these new experimental values. It is not unlikely that we may have a conclusive determination of the remaining quantity, heat of the liquid, but as stated in my paper the concordance of all quantities involved in the computation of steam tables is even now very satisfactory so that there is no reason to anticipate any necessity for changing tables for engineers.

No. 1245

SOME PROPERTIES OF STEAM

By Prof. R. C. H. Heck, New Brunswick, N. J.

Member of the Society

The purpose of this paper is to present some recent experimental results as to two of the fundamental thermodynamic properties of water and steam, and to make certain comparisons between these determinations and the older values used in our steam tables. The two properties considered are, the relation between pressure and temperature of saturated steam, and the specific heat of water.

THE PRESSURE-TEMPERATURE RELATION

2 This relation is, from the point of view of experimental determination, the simplest of the properties of steam, and with accurate instruments and adequate skill can be very precisely measured. For this reason, the results obtained by various experimenters differ by relatively small amounts, and in discussing them we take up a question in the realm of scientific accuracy rather than one concerning effectively correct values for ordinary technical use. For certain purposes, however, it is most important that this relation be truly and accurately known.

3 In *Annalen der Physik*, 1907, vol. 22, p. 609 to 630, is published a paper by F. Henning, On the Saturation Pressure of Steam, in which are gathered together all the determinations that have been made on this relation, from Magnus and Regnault down to that time. These are compared by means of curves, which show, to a large scale, their departures from an assumed standard of reference. This standard is the formula of Thiesen,

$$(t+273)\log\frac{p}{760} = 5.409\,(t-100) - 0.508\times 10^{-8}\,[(365-t)^4 - 265^4]$$

where t is centigrade temperature and p is pressure in millimeters of mercury. From the comparison and discussion the conclusion was reached that up to 100 deg. cent. this formula is to be accepted,

Presented at the Spring Meeting, Washington, May 1909, of The American Society of Mechanical Engineers.

while above 100 deg. the determinations of Regnault are best—not as set forth by his formula, but as worked over by Henning, from a selection of his more reliable observations.

4 A new and very accurate determination by Holborn and Henning, over the range from 50 deg. to 200 deg. cent., is fully described in *Annalen der Physik*, 1908, vol. 26, p. 833 to 883, in a paper, On the Platinum Thermometer and the Saturation Pressure of Steam, while in *Zeitschrift des Vereins deutscher Ingenieure*, February 20, 1909, is given a brief presentation and comparison of results. Exceedingly close agreement is shown between these new observations, the recomputed Regnault values, and the work of Knoblauch, Linde, and Klebe —see Table 3 in *Zeitschrift* article. The final result is a table giving p for every degree from 0 deg. to 205 deg. cent., which follows Thiesen's formula up to 50 deg., and embodies the authors' work from that point.

5 This table is here reproduced in Table 1, but with pressure converted to pounds per square inch and interpolated for every degree fahrenheit from 32 deg. to 402 deg., or to just past 250 lb. abs. Later the writer hopes to extend this table, carrying forward the line of the Holborn-Henning determination in comparison with the observations of Regnault and others. This can be done even up to a pressure of 1000 lb. with sufficient accuracy for all practical purposes.

6 In the work of conversion and interpolation, it was necessary to carry the numbers to a higher degree of apparent accuracy, or to use more significant figures, than any experimental precision would call for. Without a mathematical formula, a function of this sort can be carried forward only by carefully smoothing out the differences until those of the second order follow a continuous rate of change. In this operation, the first differences were brought to a sufficient degree of smoothness to furnish effectively accurate values of the rate of change of p with t; and this differential coefficient, $\frac{dp}{dt}$ is also given in Table 1. It may be considered absolutely correct (as a derivative) within about four or five units in the last place, while as between successive values the closeness is much better. This is less precise than might be desired, but it is accurate enough for use in calculating specific volume, since the thermal data there involved are not of any greater degree of reliability.

7 In Fig. 1 is given a comparison between the pressures in Table 1 and some hitherto generally used values. The base is temperature fahrenheit, the ordinate the difference between the other value of p and that in Table 1. Curve 1, for the range up to 225 deg. fahr., is

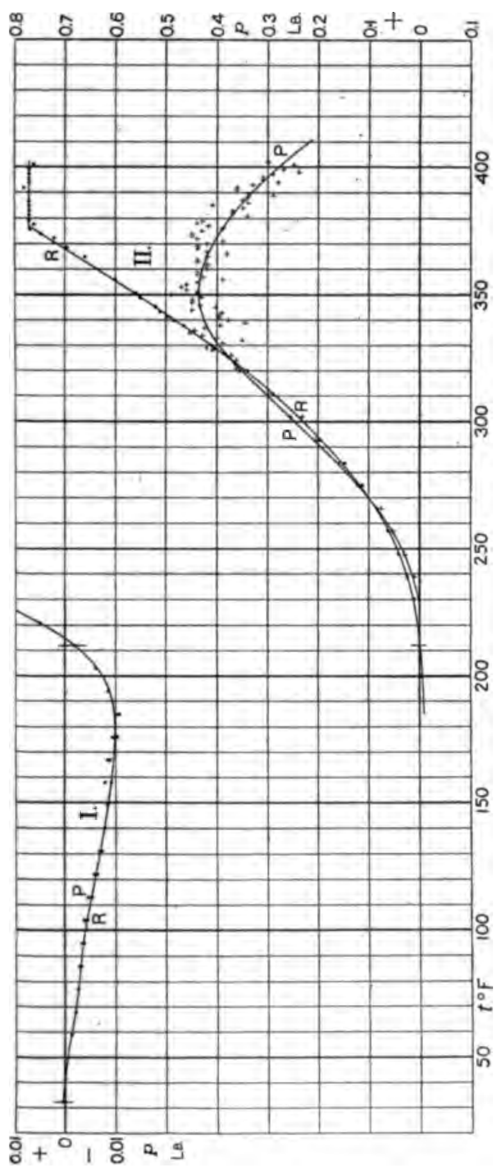

FIG. 1 COMPARISON OF PRESSURE-TEMPERATURE DETERMINATIONS

DIAGRAM SHOWS DEPARTURE OF REGNAULT (R) AND PEABODY (P) VALUES OF THE PRESSURE p FROM THOSE OF HOLBORN AND HENNING, AS GIVEN IN TABLE 1 drawn to the large scale at the left, and shows how Regnault's formula drops below the new determination. The curves at 2 have the ordinate scale at the right, only one-tenth as large as that for 1. The letter R marks the "standard" Regnault curve, here plotted from the table in Roentgen's Thermodynamics, which happened to be the most convenient in its manner of expression: note the abrupt change at about 380 deg. fahr. Curve P shows Peabody's values, which are based on Regnault, but with revised computations, and depart quite decidedly from the older table above 325 deg. The scattering of the points above that temperature is due to the coarseness of numerical expression, Peabody giving but one decimal place for the higher pressures. The curve is simply sketched through this band of points.

8 Holborn and Henning do not attempt to devise a formula, but base their table on a method of graphical interpolation. It will be noted that Curve 1 shows a faint waviness, indicating some departure from perfect mathematical smoothness; but the extreme smallness of the irregularities is really a proof of the skill with which the original interpolation was made.

THE SPECIFIC HEAT OF WATER

9 In Fig. 2 are plotted several important curves for the specific heat of water—the true or instantaneous, not the mean value. Curve R shows Regnault's formula, which in fahrenheit units is,

$$c = 1 + 0.0000222\ (t - 32) + 0.000000278\ (t - 32)^2$$

This curve differs radically from the newer and true determination of the specific heat over the lower part of the range, as shown by the other curves.

10 Curve B represents the experiments of H. T. Barnes and associates; these are described briefly in *Proceedings Royal Society*, 1900, vol. 67, fully in *Phil. Trans. Roy. Soc.*, 1902, vol. A 199; while in *Physical Review*, 1902, vol. 15, there is a description of the determination on supercooled water, which was carried to -5 deg. cent., and the tabulated values for the whole range up to 95 deg. cent. The body of this work was done by a continuous method, water flowing through a small tube and absorbing heat which was electrically supplied and measured; for the range below freezing, a method of mixing was found necessary.

11 Curve P, which begins at 140 deg. fahr., shows the values used by Peabody above this temperature; below it he accepts the work of Barnes. Peabody's line—it is almost straight—is based on Reg-

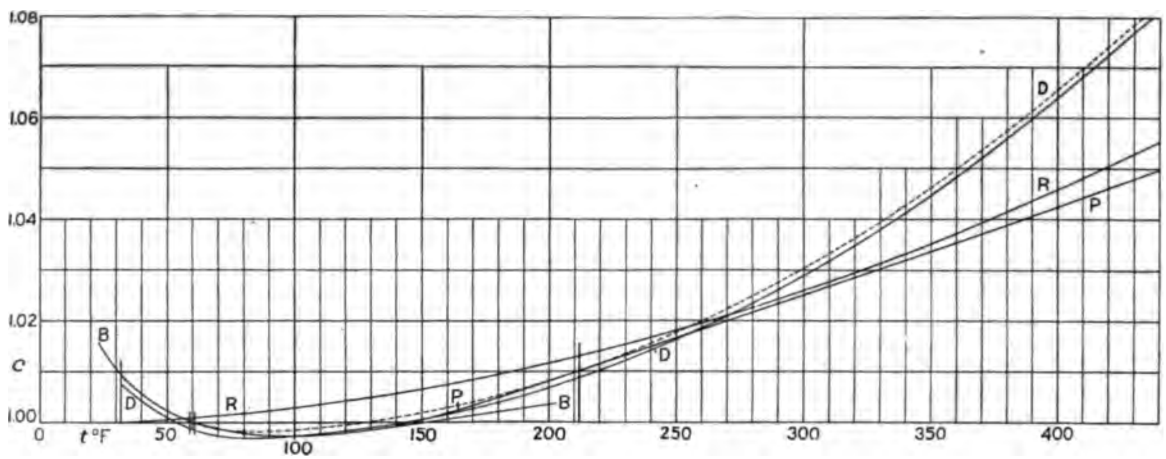

FIG. 2 THE SPECIFIC HEAT OF WATER, VALUES MEASURED OR USED BY REGNAULT (*R*), BARNES (*B*), PEABODY (*P*), AND DIETERICI (*D*)

THE LAST ARE IN MEAN CALORIES; CONVERSION TO THE 15-DEG. CENT. CALORIE RAISES THE *D* CURVE TO THE DOTTED POSITION nault's experiments: but it hardly seems reasonable to make c thus an almost straight-line function of t.

12 Curve D shows the very important experiments of Dieterici, described in *Annalen der Physik*, 1905, vol. 16. In these a small body of water, pure and free from air, was sealed in a tube of quartz. This little cartridge was heated to a certain desired temperature, then dropped into a Bunsen ice calorimeter, where the heat given off in its cooling to 0 deg. cent. is measured. The highest temperature reached was about 300 deg. cent. The drawback in this method is the relatively large heat capacity of the quartz tube, which has to be very carefully determined. From 100 deg. fahr. upward, Dieterici finds that his results conform very well to a parabolic equation like that of Regnault, which for fahrenheit units has the constants,

$$c = 0.99827 - 0.0000576\ (t - 32) + 0.00000064\ \ (t - 32)^2$$

Below 100 deg. fahr., tabulation from graphical interpolation is preferable to expression by formula. A numerical comparison of the several curves is given in Table 2.

DIFFERENT HEAT UNITS

13 Before discussing these data, something must be said as to the unit of heat measurement. Regnault intended to use the heat capacity of water at 15 deg. cent. as the heat unit—in other words, the 15-deg. calorie—but it was not until long after his time that the true manner of variation of the specific heat over the lower range of ordinary temperatures was either clearly perceived or accurately measured. Barnes' values are based on unity at 16 deg. cent., and it will be noted that the B curve on Fig. 2 crosses the base-line at just about 16 deg. cent. (the two short vertical cross-lines near 60 deg. fahr. are at 15 deg. and 16 deg. cent.). The now generally used numerical values of the mechanical equivalent of heat, 427 m-kg. or 778 ft. lb. are based on a heat unit at 15 deg. cent or 59 deg. fahr.

14 Dieterici's results are expressed in the mean calorie, which is one one-hundredth of the heat required to raise 1 kg. of water from 0 deg. to 100 deg. cent.; and his specific heat values check up to an average of unity over this range. Graphically, on Fig. 2, his curve cuts the 15-deg. cent. ordinate at 0.0012 below the unity base-line. In a special experiment, with electrical measurement analogous to that used by Barnes, he made the mechanical equivalent of the mean calorie bear to our standard Rowland value for the 15-deg. calorie the

ratio of the numbers 419.25 to 418.8, or 1.0011 to 1.00000. Disregarding some uncertainties which may exist in the minds of physicists as to the finality of this determination, it seems reasonable, for engineering purposes, to use this 0.0011 or 0.11 per cent correction in order to change from one system of units to the other.

15 The amount of attention here paid to this small point is justified by the importance given to it through the introduction of the mean calorie to the Society in the recent paper on The Total Heat of Saturated Steam, by Dr. H. N. Davis. Personally, I think we had better transform heat values in this unit by means of the ratio just offered, rather than change our mechanical equivalent of heat from 778 to 778.9.

16 Now the specific heat is the ratio of a certain absolute quantity of heat to an assumed unit quantity. If we use a larger unit, the ratio will be smaller, and *vice versa*. Assuming that the mean calorie is 1.0011 of the 15-deg. calorie, we change Dieterici's values to the 15-deg. unit if we increase them by 0.11 per cent. This would raise his curve to the dotted position on Fig. 2, and change his formula to $c = 0.99938 - 0.00005766\ (t - 32) + 0.0000006407\ (t - 32^2.)$

SPECIFIC HEAT OF WATER—CONCLUSION

17 It is pretty safe to say that the Holborn-Henning results for pressure and temperature, set forth in Table 1, are final, and that this relation is now known surely and accurately enough for all purposes of practical science. But in regard to the specific heat of water we are yet confronted by one of the annoying uncertainties which have so long surrounded many parts of this subject. Dieterici claims an experimental accuracy ranging from 0.1 per cent at low ranges to 0.5 per cent at high ranges of temperature; but his method is open to the objection that two heat-capacities have to be measured and their difference used.

18 In spite of some small doubt as to the accuracy of Dieterici's results, and a faint suspicion that his curve may rise too rapidly, I am of the opinion that his determination is to be accepted instead of Regnault's. Further, the idea of an increasing rate of increase in c, as expressed by a second-degree equation, seems to be far more reasonable than that of a nearly constant rate of increase.

19 It is hardly probable that the heat capacity of water will ever be so accurately determined that the heat for the external work of expanding the water will be more than a small fraction of the probable error in heat measurement.

TABLE 1 THE PRESSURE-TEMPERATURE RELATION

t	p	dp/dt	t	p	dp/dt	t	p	dp/dt	t	p	dp/dt
....	....	....	76	0.4433	0.01467	121	1.7362	0.04815	166	5.459	0.1277
32	0.0886	0.003575	77	0.4582	0.01510	122	1.7849	0.0493	167	5.588	0.1302
33	0.0922	0.00371	78	0.4735	0.01554	123	1.8348	0.0505	168	5.719	0.1327
34	0.0960	0.003845	79	0.4893	0.01600	124	1.8859	0.0517	169	5.853	0.1353
35	0.0999	0.003985	80	0.5055	0.01646	125	1.9382	0.05295	170	5.990	0.1380
36	0.1039	0.00413	81	0.5222	0.01694	126	1.9918	0.0542	171	6.129	0.1407
37	0.1081	0.00428	82	0.5394	0.01742	127	2.0466	0.05545	172	6.271	0.1434
38	0.1125	0.00443	83	0.5570	0.01792	128	2.1027	0.05675	173	6.416	0.1462
39	0.1170	0.004585	84	0.5752	0.01844	129	2.1601	0.0581	174	6.564	0.1490
40	0.1217	0.004745	85	0.5939	0.01898	130	2.2189	0.05945	175	6.714	0.1519
41	0.1265	0.00491	86	0.6132	0.01952	131	2.2790	0.0608	176	6.868	0.1548
42	0.1315	0.005075	87	0.6330	0.02008	132	2.3405	0.06215	177	7.024	0.1577
43	0.1367	0.00525	88	0.6533	0.02065	133	2.4033	0.06355	178	7.183	0.1607
44	0.1420	0.00543	89	0.6743	0.02123	134	2.4675	0.06495	179	7.345	0.1637
45	0.1475	0.00561	90	0.6958	0.02182	135	2.5332	0.06645	180	7.511	0.1668
46	0.1532	0.00580	91	0.7179	0.02243	136	2.6004	0.0680	181	7.679	0.1699
47	0.1591	0.00600	92	0.7406	0.02305	137	2.6692	0.0696	182	7.850	0.1730
48	0.1652	0.00620	93	0.7640	0.02368	138	2.7396	0.0712	183	8.025	0.1762
49	0.1715	0.00641	94	0.7880	0.02432	139	2.8116	0.0728	184	8.203	0.1794
50	0.1780	0.00663	95	0.8127	0.02498	140	2.8851	0.0744	185	8.384	0.1827
51	0.1847	0.00685	96	0.8380	0.02566	141	2.9603	0.0760	186	8.568	0.1860
52	0.1917	0.00708	97	0.8640	0.02635	142	3.0371	0.0776	187	8.756	0.1894
53	0.1989	0.00731	98	0.8907	0.02705	143	3.1155	0.0793	188	8.947	0.1929
54	0.2063	0.00754	99	0.9181	0.02776	144	3.1956	0.0810	189	9.142	0.1964
55	0.2104	0.00778	100	0.9462	0.02849	145	3.2775	0.0828	190	9.340	0.1999
56	0.2219	0.00803	101	0.9751	0.02923	146	3.3612	0.0846	191	9.542	0.2035
57	0.2301	0.00829	102	1.0047	0.02999	147	3.4467	0.0864	192	9.747	0.2072
58	0.2385	0.00856	103	1.0350	0.03077	148	3.5341	0.0883	193	9.956	0.2109
59	0.2472	0.00883	104	1.0662	0.03157	149	3.6233	0.0902	194	10.169	0.2147
60	0.2561	0.00911	105	1.0982	0.03240	150	3.7141	0.0921	195	10.385	0.2185
61	0.2653	0.00939	106	1.1310	0.03325	151	3.808	0.0940	196	10.606	0.2224
62	0.2749	0.00968	107	1.1647	0.0341	152	3.903	0.0960	197	10.830	0.2263
63	0.2847	0.00998	108	1.1992	0.0350	153	4.000	0.0980	198	11.058	0.2303
64	0.2948	0.01029	109	1.2347	0.0359	154	4.099	0.1001	199	11.291	0.2343
65	0.3053	0.01061	110	1.2711	0.03685	155	4.200	0.1022	200	11.527	0.2384
66	0.3161	0.01094	111	1.3084	0.03775	156	4.303	0.1043	201	11.767	0.2425
67	0.3272	0.01127	112	1.3466	0.0387	157	4.408	0.1064	202	12.013	0.2467
68	0.3386	0.01161	113	1.3858	0.0397	158	4.516	0.1086	203	12.261	0.2509
69	0.3504	0.01196	114	1.4260	0.0407	159	4.625	0.1108	204	12.514	0.2552
70	0.3625	0.01232	115	1.4671	0.0417	160	4.737	0.1131	205	12.771	0.2595
71	0.3750	0.01269	116	1.5093	0.0427	161	4.852	0.1154	206	13.033	0.2639
72	0.3879	0.01307	117	1.5525	0.04375	162	4.968	0.1178	207	13.299	0.2683
73	0.4012	0.01345	118	1.5968	0.0448	163	5.087	0.1202	208	13.569	0.2728
74	0.4148	0.01384	119	1.6421	0.0459	164	5.209	0.1227	209	13.845	0.2783
75	0.4289	0.01425	120	1.6886	0.0470	165	5.332	0.1252	210	14.124	0.2819

TABLE 1.—Continued

t	p	dp/dt	t	p	dp/dt	t	p	dp/dt	t	p	dp/dt
211	14.408	0.2866	256	33.085	0.5677	301	67.99	1.015	346	127.67	1.675
212	14.697	0.2914	257	33.657	0.5758	302	69.01	1.027	347	129.35	1.693
213	14.991	0.2962	258	34.236	0.5840	303	70.05	1.0395	348	131.05	1.711
214	15.290	0.3011	259	34.824	0.5922	304	71.09	1.052	349	132.77	1.729
215	15.594	0.3061	260	35.420	0.6005	305	72.15	1.065	350	134.51	1.746
216	15.902	0.3111	261	36.025	0.6088	306	73.22	1.0775	351	136.26	1.764
217	16.215	0.3162	262	36.638	0.6172	307	74.31	1.090	352	138.04	1.782
218	16.534	0.3214	263	37.259	0.6256	308	75.40	1.103	353	139.83	1.800
219	16.858	0.3266	264	37.888	0.6341	309	76.51	1.116	354	141.64	1.818
220	17.187	0.3319	265	38.526	0.6426	310	77.64	1.129	355	143.46	1.836
221	17.521	0.3372	266	39.173	0.6513	311	78.77	1.142	356	145.31	1.855
222	17.860	0.3426	267	39.828	0.6600	312	79.92	1.155	357	147.17	1.874
223	18.205	0.3480	268	40.492	0.6688	313	81.08	1.169	358	149.06	1.893
224	18.556	0.3535	269	41.165	0.6777	314	82.26	1.182	359	150.96	1.912
225	18.913	0.3591	270	41.848	0.6868	315	83.44	1.195	360	152.88	1.931
226	19.275	0.3648	271	42.54	0.6960	316	84.65	1.209	361	154.82	1.951
227	19.643	0.3705	272	43.24	0.7052	317	85.86	1.223	362	156.78	1.970
228	20.017	0.3763	273	43.95	0.7145	318	87.09	1.237	363	158.76	1.990
229	20.396	0.3821	274	44.67	0.7239	319	88.34	1.251	364	160.76	2.010
230	20.781	0.3880	275	45.40	0.7334	320	89.60	1.265	365	162.78	2.029
231	21.172	0.3940	276	46.14	0.7430	321	90.87	1.280	366	164.82	2.049
232	21.568	0.4000	277	46.88	0.7527	322	92.16	1.295	367	166.88	2.069
233	21.970	0.4061	278	47.64	0.7625	323	93.46	1.309	368	168.96	2.089
234	22.379	0.4123	279	48.41	0.7725	324	94.78	1.324	369	171.06	2.108
235	22.794	0.4185	280	49.19	0.7826	325	96.17	1.339	370	173.18	2.128
236	23.216	0.4248	281	49.98	0.7926	326	97.45	1.354	371	175.31	2.148
237	23.644	0.4312	282	50.77	0.8028	327	98.81	1.369	372	177.47	2.168
238	24.079	0.4377	283	51.58	0.8131	328	100.19	1.384	373	179.65	2.189
239	24.520	0.4442	284	52.40	0.8235	329	101.58	1.400	374	181.85	2.210
240	24.967	0.4508	285	53.23	0.8340	330	102.99	1.415	375	184.07	2.231
241	25.421	0.4575	286	54.07	0.8446	331	104.41	1.430	376	186.31	2.252
242	25.882	0.4643	287	54.92	0.8553	332	105.85	1.445	377	188.58	2.274
243	26.350	0.4711	288	55.78	0.8661	333	107.30	1.461	378	190.86	2.296
244	26.825	0.4780	289	56.65	0.8770	334	108.77	1.477	379	193.17	2.318
245	27.307	0.4850	290	57.53	0.8880	335	110.26	1.493	380	195.50	2.341
246	27.795	0.4920	291	58.42	0.8991	336	111.76	1.509	381	197.86	2.364
247	28.290	0.4991	292	59.33	0.9103	337	113.27	1.525	382	200.23	2.387
245	28.793	0.5063	293	60.25	0.9216	338	114.81	1.542	383	202.63	2.410
249	29.303	0.5136	294	61.17	0.9330	339	116.36	1.558	384	205.05	2.433
250	29.820	0.5210	295	62.11	0.9445	340	117.92	1.574	385	207.49	2.456
251	30.345	0.5285	296	63.06	0.9561	341	119.50	1.591	386	209.96	2.479
252	30.877	0.5361	297	64.03	0.9678	342	121.10	1.607	387	212.45	2.502
253	31.417	0.5438	298	65.00	0.9796	343	122.72	1.624	388	214.96	2.525
254	31.965	0.5517	299	65.98	0.9915	344	124.35	1.641	389	217.50	2.548
255	32.521	0.5596	300	66.98	1.0035	345	126.00	1.658	390	220.06	2.571

TABLE 1.—Continued

t	p	dp/dt	t	p	dp/dt	t	p	dp/dt	t	p	dp/dt
391	222.64	2.594	395	233.20	2.687	398	241.37	2.759	401	249.75	2.832
392	225.24	2.617	396	235.90	2.711	399	244.14	2.783	402	252.60	2.857
393	227.87	2.641	397	238.62	2.735	400	246.93	2.807			
394	230.52	2.664									

TABLE 2 THE SPECIFIC HEAT OF WATER

Temperature		Regnault	Dieterici	Barnes	Peabody
Cent.	Fahr.				
−5	23	....	....	1.0158	....
0	32	1.00000	1.0075	1.0094	....
+5	41	....	1.0037	1.00530	....
10	50	1.00049	1.0008	1.00280	....
15	59	....	0.9987	1.00080	....
20	68	1.00116	0.9974	0.99895	....
25	77	....	0.9970	0.99806	....
30	86	1.00201	0.9971	0.99759	....
35	95	....	0.9972	0.99735	....
40	104	1.00304	0.9974	0.99735	....
50	122	1.00425	0.9983	0.99800	....
60	140	1.00564	0.9995	0.99910	0.99940
70	158	1.00721	1.0012	1.00035	1.00150
80	176	1.00896	1.0032	1.00166	1.00415
90	194	1.01089	1.0057	1.00305	1.00705
100	212	1.01300	1.0086	(1.0044)	1.01010
120	248	1.01776	1.0157	....	1.01620
140	284	1.02324	1.0244	....	1.02230
160	320	1.02944	1.0348	....	1.02850
180	356	1.03636	1.0468	....	1.03475
200	392	1.04400	1.0605	....	1.04100
220	428	1.05236	1.0758	....	1.04760
240	464	(1.06144)	1.0928	....	....
260	500	(1.07124)	1.1115	....	....
280	536	(1.08176)	1.1318	....	....
300	572	(1.09300)	1.1538	....	....

Regnault: from formula, par. 9. Above 200 deg. cent. his formula is an extrapolation.
Dieterici: from table in original publication, computed by formula from 40 deg. cent. upward.
Barnes: from *Physical Review*, with last value extrapolated.
Peabody: from Steam and Entropy Tables, p. 10.
Dieterici: values in mean calor[illegible] (heat units), others in 15 deg. cent. units.

DISCUSSION

Dr. Sanford A. Moss. In Regnault's original paper on pressure and temperature of saturated steam was given an empirical formula, first suggested by Roche, but reconstructed by Regnault and called "Formula K." This represented Regnault's results more closely than any other single formula and has also been shown to represent other experimental results, particularly those for very high pressures and temperatures. This formula was discussed by Ramsey and Young in the Philosophical Transactions, vol. 183, 1892, page 111, and also in London Engineering, vol. 83, January 4, 1907. I have given some discussion of this matter in the Physical Review, vol. 25, no. 6, December 1907. It would be interesting if Professor Heck would give a comparison of the values of his table with those computed by this formula. If it can be demonstrated, as is the conclusion in the papers above mentioned, that this formula represents all of the experimental results very closely, it is highly desirable that it be used. Thermodynamic computations can be carried on very readily, if we have a single formula for the entire range.

Prof. G. A. Goodenough.[1] Since the experimental results of Holborn and Henning have been generally accepted, it seems highly desirable to have an analytical relation that will express these results with a sufficient degree of accuracy. For this purpose the formula of Bertrand,

$$\log p = k - n \log \frac{T}{T - a} \qquad [1]$$

seems to be quite suitable. In this formula k, n and a are constants and T denotes the absolute temperature.

2 The values of the constants can be so chosen that the formula will give fair results throughout the ordinary range of temperatures with one set of constants. A closer agreement, however, is obtained by dividing the temperature range into three parts. The values of the constants are:

For t = 32 deg. – 90 deg., $k = 6.23167$, $a = 140.1$ $n = 50$
For t = 91 deg. – 237 deg., $k = 6.30217$, $a = 141.43$ $n = 50$
For t = 238 deg. – 400 deg., $k = 6.27756$, $a = 140.8$ $n = 50$

[1]Associate Professor Mechanical Engineering, University of Illinois, Urbana, Ill.

3 The derivative $\frac{dp}{dt}$ takes the simple form

$$\frac{dp}{dt} = \frac{pna}{T(T-a)} \qquad [2]$$

from which follows

$$T\frac{dp}{dT} = \frac{pna}{T-a} \qquad [3]$$

This relation is important since the product $T\frac{dp}{dT}$ appears in the Clapeyron-Clausius formula for steam volume. Table 1 shows a comparison of the values of p and $\frac{dp}{dt}$ calculated from Formulæ 1 and 2, respectively, with the values obtained by Professor Heck.

4 It will be seen that the maximum difference between the corresponding values of p is about 0.1 per cent. The maximum difference between the corresponding values of the derivative is greater.

COMPARISON OF VALUES OF p AND $\frac{dp}{dt}$

t	p Beck's value	p Bertrand's formula	dp/dt Heck	dp/dt Bertrand's formula
32	0.0886	0.0885	0.003575	0.003588
50	0.1780	0.1781	0.00663	0.006625
75	0.4289	0.4288	0.01425	0.014245
100	0.9462	0.9455	0.02849	0.028575
125	1.9382	1.9387.	0.05295	0.052921
150	3.7141	3.7144	0.0921	0.092041
175	6.714	6.711	0.1519	0.15165
200	11.527	11.523	0.2384	0.23843
225	18.913	18.921	0.3591	0.35985
250	29.820	29.834	0.5210	0.52039
275	45.40	45.382	0.7334	0.73248
300	66.98	66.94	1.0035	1.0027
325	96.11	96.07	1.339	1.3390
350	134.51	134.50	1.746	1.7489
375	184.07	184.18	2.231	2.2394
400	246.93	247.20	2.807	2.8167

Evidently this is due to the fact that the values of $\frac{dp}{dt}$ were obtained by Professor Heck (and also by Professor Peabody) by a method involving some approximation; and it is likely that the values calculated from Formula 2 are the more reliable.

5 A great advantage of Bertrand's formula lies in the ease with which it may be used in calculation. This is shown by the following scheme, which is copied from calculations recently made for the engineering experiment station of the University of Illinois:

$t =$	311	312	313	314
$T =$	770.58	771.58	772.58	773.58
$T-a =$	629.78	630.78	631.78	632.78
$\log T =$	2.8868177	2.8873810	2.8879435	2.8885052
$\log T-a =$	2.7991889	2.7998779	2.8005659	2.8012527
$\log \frac{T}{T-a} =$	0.0876288	0.0875031	0.0873776	0.0872525
$50 \log \frac{T}{T-a} =$	4.381440	4.375155	4.368880	4.362625
$k =$	6.27756	....	....	....
$\log p = k - 50 \log \frac{T}{T-a} =$	1.896120	1.902405	1.908680	1.914935
$p =$	78.726	79.874	81.036	82.212
$\log na =$	3.8475727	....	....	....
$\log pna =$	5.7436927	5.7499777	5.7562527	5.7625077
$\log \frac{pna}{T-a} = \log T \frac{dp}{dt} =$	2.9445038	2.9500998	2.9556768	2.9612550

6 In conclusion, it may be stated that Bertrand's formula seems to have a wide range of applicability, and with proper choice of constants can probably be used equally well for nearly all saturated vapors.

THE AUTHOR. The preparation of this paper was an early step in a special study of the properties of steam upon which the writer has been engaged for some months; the object of this investigation has been to formulate more precisely our knowledge of the subject up to the limit of accurate experiment, at about 400 deg. fahr., and to extend the relations into the higher ranges, even up to the critical temperature. It has been found that Thiesen's formula, by modification of the two coefficients or constants, can be made to fit the Holborn-Henning table exactly from 100 deg. to 200 deg. cent., and that this modified equation holds good up to about 450 deg. fahr. Above that point the curve begins to rise more rapidly, and even crosses the original Thiesen curve. Below 212 deg., the relation can be very exactly extended by means of a small adjustment from the Thiesen values.

2 The formula presented by Professor Goodenough fits the data very well; but for precise calculation it has the disadvantage that at

each point of change in constants there will be a break in the curve of the derivative $\frac{dp}{dt}$. The Thiesen formula has a forbiddingly complex expression for its derivative, and would be much less convenient for regular use, if the calculation of $\frac{dp}{dt}$ were an operation to be undertaken frequently.

3 The writer has tried formula K, which is recommended by Dr. Moss. In the very high range it runs low in p, dropping beneath the modified Thiesen formula, which itself is not high enough for the more reliable data.

4 The values of $\frac{dp}{dt}$ in Table 1, found simply by the method of differences, are not precise enough for close work with Clapeyron's equation, the irregularities resulting from the use of them running as high as nearly one-fifth of one per cent.

No. 1246

A NEW DEPARTURE IN FLEXIBLE STAY-BOLTS

By H. V. Wille, Philadelphia, Pa.

Member of the Society

There is practically no literature on the subject of stay-bolts, and this is particularly true of flexible stay-bolts. The increasing size and pressure of boilers make this subject of vital importance to railroads and to those responsible for the management of that type of boiler in which the firebox is stayed by a large number of bolts.

2 The boiler of the consolidation locomotive, now the prevailing type in freight service, contains about 1000 bolts less than 8 in. long and about 300 of greater length. The large types of Mallet compound locomotives now meeting with much favor have a much larger number, there being 1250 short and 300 long bolts in locomotives recently constructed.

3 In recent years some form of flexible stay-bolt, that is, one having a movable joint, has been very extensively used in the breaking zone of locomotive boilers, but their high cost and the difficulty of applying them, their rigidity from rust and scale, and the fact that their use throws an additional service on the adjacent bolts because of lost motion, has militated against their more general use.

4 It is well known that stay-bolts fail, not because of the tensional loads upon them, but from flexural stresses induced by the vibration resulting from the greater expansion of the firebox sheets than of the outside sheets, but notwithstanding the general acceptance of this theory, engineers have designed stay-bolts solely with respect to the tensional loads. It is quite general practice, it is true, to recess the bolts below the base of the thread, and this has effected a slight reduction in the fiber stress, but practically no effort has been made to design a bolt to meet the flexural stresses or even to calculate their magnitude. This is surprising in view of the simplicity of the calculations to which the ordinary formulae for flexure apply.

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5 Let

F = fiber stress.
E = modulus of elasticity.
I = moment of inertia.
D = diameter.
N = deflection.
L = length.
W = load.

We then have

$$W = \frac{2\,F\,I}{D\,L} \quad \text{.......... (1)}$$

$$N = \frac{W\,L^3}{3\,E\,I} \quad \text{.......... (2)}$$

Substituting

$$N = \frac{2\,F\,L^2}{3\,E\,D} \quad \text{.......... (3)}$$

$$F = \frac{3\,E\,D\,N}{2\,L^2}\,^* \quad \text{.......... (4)}$$

This formula shows that the stress increases in direct proportion to the diameter and decreases as the square of the distance between the sheets.

6 The application of the formula to service conditions gives the following stresses:

Conditions: Bolt spacing, 4 in. centers.
Assumed expansion, 4/100 in.
Length of bolt, 6 in.

Type	Diameter of Bolt	Flexural Stress
Iron	1⅛ in.	51,500
Iron	1 "	45,000
Iron	⅞ "	39,400
Spring steel	1 in. ends 7/8 in. stem	19,700

7 Iron is universally employed in the manufacture of these bolts and it is not good practice to exceed a fiber stress of 12,000 lb. per

* Testing of Stay-bolt Iron, H. V. Wille, A. S. T. M., vol. 4, 1904.

square inch. It is apparent that stay-bolts in the zone which meets the expansion of the sheets are stressed above the elastic limit and must necessarily fail from fatigue. Fractures always originate at the outside sheet at the point where the bending moment due to the movement of the furnace sheets is greatest.

8 The fractures are in detail, usually starting from the base of a thread and gradually extending inward. Manufacturers of stay-bolt material have endeavored to minimize failures and to meet the unusual conditions of an iron stressed beyond its elastic limit by the supply of specially piled iron arranged with a view to breaking up the extension of the initial fracture. For this reason iron piled with

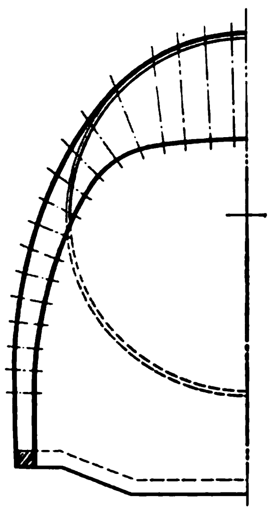


FIG. 1 SECTION OF FIREBOX SHOWING STAY-BOLTS

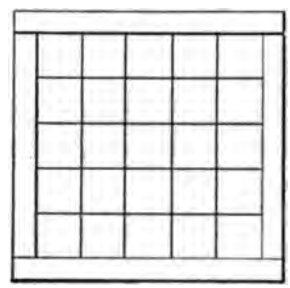


FIG. 2 FAGGOTT PILING FOR IRON FOR STAY-BOLTS

a central section of small bars and an envelop of flat plates has met with much success for this class of service. In a further effort to secure an iron specially adapted to this class of work various forms of shock, vibratory and fatigue tests have been imposed. No design has yet been produced however which permits the employment of material of elastic limit sufficiently high to resist the flexural stresses, although a large class of material particularly adapted to the purpose is available.

9 It is obvious that the remedy does not lie in the use of a slow-breaking material but in the employment of material of sufficiently high elastic limit to meet the conditions of service. It is also possi-

ble to reduce the diameter of the bolt greatly by the use of such a material, thus proportionately reducing the fiber stress in flexure.

10 Stay-bolt material however must possess sufficient ductility to enable the ends to be readily hammered over to make a steam-tight joint and to afford additional security against pulling through the sheets. To meet these conditions the bolt illustrated in Fig. 3 has been designed. The stem is of the same grade of steel as that used in the manufacture of springs. It is oil-tempered and will safely stand a fiber stress of 100,000 lb. per square inch. Its high elastic limit makes it possible to reduce the diameter to $\frac{1}{2}$ or $\frac{7}{16}$ in. or even less. The ends are of soft steel, and it is thus possible to apply and head up the bolt in the usual manner.

11 The employment of a stem of the diameter indicated reduces the fiber stress in flexure to less than one-half that in the ordinary type of bolt and it is of material capable of being stressed to a high degree. It has hitherto been impossible to employ in stay-bolts any of the steels containing chromium, nickel, vanadium or other metaloids

Fig. 3 Flexible Spring Steel Stay Bolt

possessing properties expecially adapted to this class of work, but these steels can readily be used in the stem of the bolt described.

12 The stem of the bolt can be flexibly secured to the end in one of the customary ways, but the flexibility of the bolt does not depend upon a flexible connection. A type of bolt with a relatively inflexible connection, usually one in which the stem screwed into the ends with a running fit, met with the most favorable consideration. Such a bolt is flexible as a spring is flexible, in that it can be deflected to meet the requirements of service without exceeding the elastic limit. In fact the stem may be of a number of pieces, either of plates or small rods, thus increasing its flexibility.

13 The actual breaking strength of the bolt sizes ordinarily employed is shown in the following statement. These bolts were recessed to the base of the thread and tested in the same form as that in which they are employed in service. For comparison the approximate weights of the usual length of bolt are also given. These

weights are for bolts over the entire length, including the squared ends for screwing the bolts into the sheets.

ACTUAL BREAKING STRENGTH OF STAY-BOLTS

Type	Nominal Diameter	Actual Breaking	Weight	Vibrations
Iron.................	1 in.	32,500	20 oz.	6,000
Iron.................	$\frac{7}{8}$ "	24,500	15 "	5,200
Spring steel stem.....	1 in. ends $\frac{7}{16}$ in. stem	32,000	10 "	500,000

14 The vibrating test was made by clamping one end of the bolt in a machine and revolving the other end through a radius of $\frac{3}{32}$ in., the specimen being 6 in. long from the end of the right head to the center of the rotating head. A tensional load of 4000 lb. was also applied to the bolts. The best grades of iron bolts break on being subjected to from 5000 to 6000 rotations, whereas the spring steel bolts were

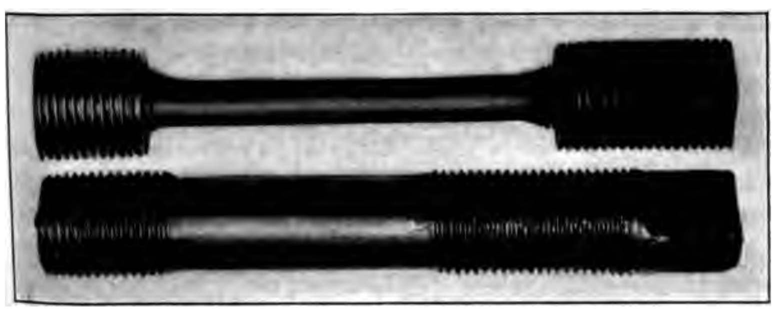


FIG. 4 SPRING FLEXIBLE AND REGULAR IRON BOLTS OF SAME TENSILE STRENGTH

vibrated 500,000 times without failure, and on some of them the test was continued without failure to 1,000,000 vibrations. These tests demonstrated that the bolt is not stressed beyond the elastic limit under these severe conditions and that the probability of its failure in less severe conditions is very remote.

15 The extent of the expansion which can take place in the firebox of a boiler can readily be calculated.

Distance between stay-bolts, 4 in.
Temperature of inside sheet, 400 deg. fahr.
Temperature of outside sheet, 100 deg. fahr.
Coefficient of expansion, 0.0000066.

Then the expansion between two bolts will equal: $0.0000066 \times (400 - 100) \times 4 = 0.0079$, and each bolt will deflect 0.00395 in. It has been shown that this amount of deflection will stress the usual type of bolt beyond the elastic limit. In practice however one bolt may hold rigidly, throwing the entire deflection on the adjacent bolt, or neither bolt may deflect and the sheet will then buckle. Under this condition the neutral axis will assume the form ABC and the length AB will equal 2.00395 in. and the sheet will buckle to an extent, $BD = \sqrt{2.00395^2 - 2^2} = 0.125$ in. It is obvious that the repetition of a force sufficient to buckle a sheet $\frac{1}{8}$ in. must ultimately lead to a

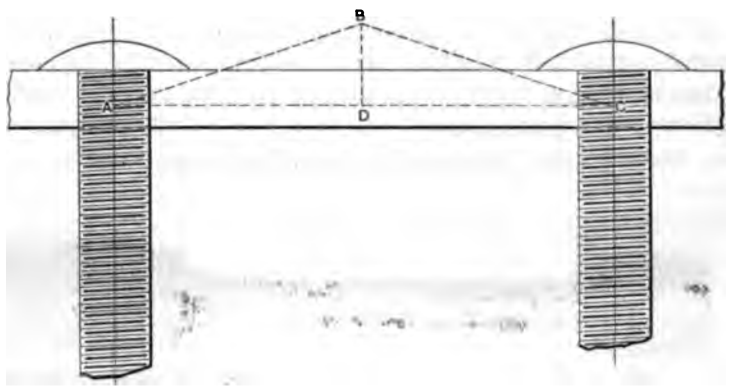


FIG. 5 SHOWING MANNER IN WHICH PLATES BUCKLE WITH RIGID STAYS

crack in the furnace sheets. If, however, the bolt deflects, allowing the sheet to expand normally, the latter will be relieved of these extraneous loads.

16 A bolt of sufficient flexibility to deflect under the forces following expansion, and of material which will not be stressed beyond the elastic limit in resisting these forces, will greatly assist in reducing the cost of boiler maintenance by eliminating broken stay-bolts and reducing the stresses in the furnace plates. If in addition the bolt has a smaller diameter the life of the furnace plates should be further increased, as such a bolt will interpose less obstruction to the circulation of the water in the water legs.

DISCUSSION

William Elmer. The decrease in the diameter of a staybolt, from 15/16 in. or 1 in., which I believe is the present practice, to a small diameter, as 7/16 in., even if the tensile strength of the material is increased, brings to mind at once the possibility of twisting off these small bolts in their mechanical application to the boiler. The writer hopes Mr. Wille will say something about this.

W. E. Hall. Any one who has had the care of the locomotive type of boiler appreciates that staybolt maintenance is a source of intense anxiety, and that the fact that more disastrous results are not forthcoming reflects great credit on the vigilance of the motive power departments of our railroads. This result, however, is accomplished only at high maintenance cost, which, fortunately, has always been of secondary consideration.

2 Flexibility and length, of which Mr. Wille speaks, no doubt have considerable influence on the breakage of these bolts. It would be interesting to have details showing the construction of the bolts, and just how they were held, in his vibratory tests.

3 It should be noted, however, that his construction calls for a studbolt of larger diameter than the ordinary staybolt, and in addition this studbolt must project into the leg of the boiler to give sufficient length of thread-contact of the staybolt proper in the studbolt; in other words, he increases the diameter (the studbolt), introduces two threaded surfaces to the strain of flexure, and shortens the staybolt an amount equal to the projection of the studbolt into the leg of the fire-box. Is this not contrary to his own deductions? Assuming that this construction would decrease, but not eliminate, broken staybolts, would this construction facilitate their detection, difficult under the best conditions, or make it more difficult? The reduced length might be relieved by making the fit of the thread of the bolt in the studbolt looser than that of the studbolt in the sheet, thereby always throwing the point of flexure upon the studbolt at the side of the outer sheet. But this is a risky procedure hardly deserving of consideration in boiler practice.

4 The breakage of staybolts is confined almost exclusively to the upper rows of bolts of the legs of the fire-box. The fracture usually starts from the top, sometimes from the side, not infrequently around the circumference and occasionally from the bottom. The fracture is almost always close to the outer sheet, but a break close to the

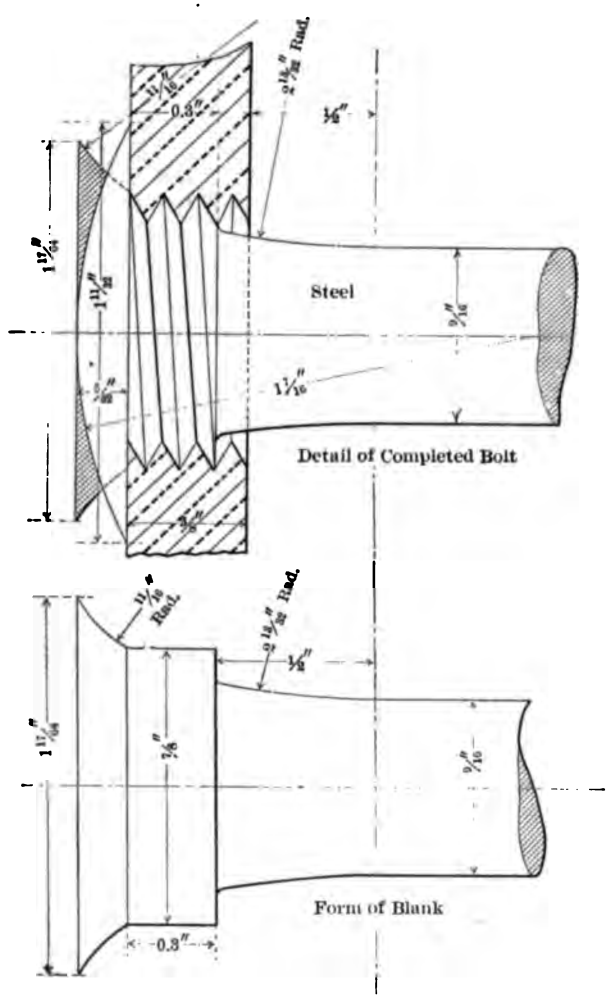


FIG. 1 BLANK AND COMPLETED ENDS OF FLEXIBLE STAYBOLT

inside sheet is not unknown. The fracture is always in detail, barring shamefully defective material or workmanship, or at least up to the point where the remaining area in contact is not sufficient to withstand the strain to which that reduced area of the bolt is subjected. More or less irregularity of the line of fracture is to be expected. Conditions are not always the same for every bolt: all bolts do not fit the sheets alike, there is more or less variation in the upsetting, and the buckling and warping of the inner sheet is not the same for each bolt. These, together with other minor conditions, representing reasonable refinement in practice, preclude uniformity of the fracture.

5 But the important points are, that these bolts always break in *detail* and always at the *root of the thread*. Have we any reason to expect that it would be otherwise? For example, if we wish to break a piece of metal we first grip it in a vise, notch it close to the jaws of the vise and bend it back and forth. We do the same with a stay-

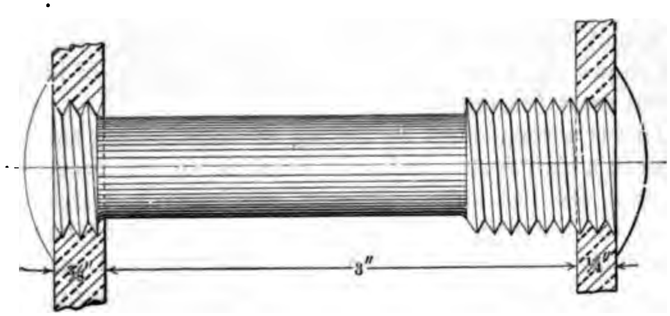


Fig. 2 Flexible Staybolt with End shown in Fig. 1

bolt when we cut the thread with a sharp die (and in this respect the U. S. Standard is inferior to the Whitworth), hold it firmly in the outer sheet and subject it to the bending due to the expansion, contraction and warping of the inner sheet. Cutting the thread produces a constructive defect. This feature is necessary as the bolts are now used and its correction is beyond the scope of mathametics.

6 Fig. 1 and Fig. 2 show designs for bolts made in January 1886. In Fig 1 the object was to make the bolt of iron or steel, and by a gradual reduction from the ends towards the center to make the body of the bolt more flexible; and to make the length of the end such that, after making sufficient allowance for upsetting, the thread could not project beyond the face of the inside of the sheet. The flare on the end of the bolt was shown merely as a more ready means of upsetting the end, especially when the bolt is of steel. As the

number of bolts breaking at the face of the inside sheet is so small as to be negligible, and to accommodate for the difference in widths in the leg of the fire box, the bolts could be made of varying lengths on a bolt machine and kept in stock. By this method, if made by upsetting, it would not be necessary to turn the shank of the bolt. In all cases, however, it is preferable that the threaded end in the heavier sheet should not project into the leg of the fire box. This construction permits of better circulation, and of a somewhat longer bolt, and better braces the threaded length against deflection.

ALFRED LOVELL. The durability of staybolts in high-pressure locomotive boilers is a prominent factor in the cost of locomotive maintenance and has an important bearing on the safety of the public. Any innovation intended to increase their durability or reliability is therefore worthy of the most careful demonstration and service trial. The staybolt described by Mr. Wille is of this character, and his paper clearly shows the desirability of providing a staybolt of high elastic limit and great flexibility.

2 This is accomplished by the use of metal having the requisite qualities in a high degree, and yet maintaining in the direction of the bolt's axis, the desirable features of continuity and rigidity, thus avoiding the liability of unequal tension of adjacent bolts. This is undeniably a departure in the right direction, and one that will receive prompt attention in railway mechanical engineering.

3 One other important feature to be considered, however, in adopting a new form or in selecting a new material for staybolts, is ability to resist corrosion under the conditions of use, a feature even more important with staybolts of the proposed high tensile strength and small diameter, than with staybolts of ordinary diameter. For example, the paper gives the actual breaking strength of a 1-in. iron bolt as 32,500 lb., and of a $\frac{7}{16}$-in. spring-steel stembolt as 32,000 lb. If it is assumed that each is reduced $\frac{1}{8}$ in. in diameter by corrosion, this will reduce the breaking-strength of the 1-in. bolt by 23.56 per cent, and of the $\frac{7}{16}$-in. bolt by 48.98 per cent. The breaking-strength of the iron bolt is then 24,843 lb. and that of the spring-steel stembolt is 16,327 lb. Another $\frac{1}{8}$-in. reduction in diameter would bring the strength of the iron bolt to 18,278 lb., and of the spring-steel stembolt to 5878 lb.

4 Formula 4 of Mr. Wille's paper shows that the flexural stress, the ordinary cause of failure, decreases directly as the diameter, yet it is

evident that if the bolts are affected by corrosion and in equal amount, this will eventually reduce the tensile strength of the small bolt to a point below the proper factor of safety, while the larger bolt has still tensile strength to spare.

5 Since the new type of staybolt makes it possible to employ various steels and alloys which cannot be used in the ordinary staybolt, it fortunately provides for a wide range of selection, and quite probably a steel-alloy or mixture may be provided that will resist corrosion equally well or better than the iron ordinarily used.

6 The impurities in the water used also very greatly affect corrosive action, and it is probable that with the new type of bolt it would be advantageous to examine carefully the water of the locality where it is to be used and to select a metal with reference to the character of the water. Information regarding the resistance of various steel alloys to corrosion, in water of ordinary purity, and experiments determining the metal that will give the least corrosion in water having various impurities, are highly desirable, since the new bolt makes possible the use of that metal which combines in the greatest degree flexibility, high elastic limit and resistance to corrosion.

7 A combination of these three qualities, in a staybolt of the type described, will be an innovation much to be desired in locomotive practice.

F. J. Cole. In reading this paper the following occurred to me:

a Have any of these staybolts been applied to locomotive boilers?

b What is their approximate cost?

c With 1-in. ends is there any danger of overstraining the $\frac{3}{8}$-in. diam. spring-steel center in screwing the bolt to place? It would seem that the torsional stress on the comparatively small central part in screwing the bolts to place when they fit tightly in the sheets would be excessive.

d In upsetting the soft ends, does the $\frac{3}{8}$-in. diam. center afford sufficient resistance to avoid injury to the threads from hammering? It is customary in upsetting the ends to use a hammer or heavy weight at the other end of the bolt to afford the necessary resistance and to prevent injury to the threads.

e In Par. 7 Mr. Wille speaks of 12,000 lb. fiber stress being good practice. I presume he has in mind the combined stress due to tension and bending. Current locomotive practice is represented fairly well by 5500 to 6000 lb. for tension alone.

The Author. Mr. Hall has fully described the general characteristics of a staybolt broken in service. The able experiments of Boushinger and Wohler have shown that a detail fracture occurs in parts which have been stressed by flexure above the elastic limit. If the parts be threaded the stress may be localized at the base of the thread which will increase the possibility of failure of parts stressed at or near the elastic limit, but if the parts be not stressed above the "fatigue" elastic limit of the material, it will not fail irrespective of whether it be threaded or of plain section.

2 The bolt described is made of two kinds of material:

a The stem of high-carbon tempered steel.
b The two ends of soft steel.

This arrangement permits the use of a material having such a high elastic limit and provides an ample factor of safety and still allows the soft ends to be rivetted over to make a steam-tight joint. The bolt referred to by Mr. Hall does not fulfil these conditions, as if it were made of steel having a sufficiently high elastic limit it would be impossible properly to rivet the ends.

3 The ends of the composite bolts are not larger than a normal bolt, those in service being made 1 in. in diameter when used in new boilers and $\frac{15}{16}$ in. and $1\frac{1}{16}$ in. when used to replace $\frac{7}{8}$-in. and 1-in. bolts respectively in old boilers. The stems are not made a tight fit into the ends but are threaded so that they can be tightly screwed into the ends by hand thus affording some flexibility in the joint. It is not necessary however to thread the stems as they can be upset and the ends closed around them, making a ball joint, thus having all the advantages with none of the disadvantages of the usual type of flexible bolt.

4 The bolts have been in service for two years in a district notorious for the trouble experienced with corrosion and the bolts are still intact. Mr. Lovell suggests the possibility of providing special steels to resist corrosion. The high-carbon tempered steel being dense and hard is an admirable material to resist corrosion; furthermore as the stems are oil-tempered they have a coat of enamel which prevented the slightest amount of pitting even after two years' service.

5 There are a large number of these bolts in service in locomotive

boilers and no difficulty has been experienced either from twisting of the stem in screwing into the boiler or from injury to threads in rivetting.

6 The cost of material entering into the construction of the spring flexible bolt is very much less than that of the usual type of staybolt since the weight is much less and the materials are the relatively cheap commercial grades of steel instead of the high-priced irons usually employed in the manufacture of staybolts. The labor cost to manufacture is however somewhat higher.

7 There have been about 3000 of these bolts put into service and no danger has been experienced of overstrainin the $\frac{3}{8}$-inch tem in screwing the bolt in the boiler; nor has any difficulty been experienced because of insufficient resistance to avoid injury to the ehread when the bolts are hammered up.

No. 1247

THE HUDSON-FULTON CELEBRATION

In keeping with the celebration of the discovery of the Hudson River and the successful application of steam to navigation, the House Committee of the Society appointed a subcommittee, Edward Van Winkle, Chairman, to prepare an exhibit of models, drawings, letters, books and other items related to early steam navigation, of interest to the general public as well as the engineer.

The exhibit was the only participation of any engineering organization, as such, in the celebration, and much credit is due Mr. Van Winkle for the time and attention devoted to its preparation. The exhibit was held in the rooms of the Society and was open from 9 a. m. to 5 p. m. every week day during September and October 1910. A list of the exhibits follows:

Model of Fulton's boat, the Clermont, as she was at the time of her maiden trip. After making several trips during the fall of 1807, the Clermont was docked at Browne's shipyard and fitted up for regular passenger traffic. Loaned by the Smithsonian Institution.

Model of Stevens' boat, the Phoenix, at the time of making her notable New York-Philadelphia trip—the first ocean voyage made by a steam vessel. Loaned by the Smithsonian Institution.

Model of a steamboat built by John Fitch in 1786, having a peculiar arrangement of oars dipping into the water something like a canoe paddle. Despite its clumsy appearance the boat made a trip of 20 miles on the Delaware River. Loaned by the Smithsonian Institution.

Model (9-ft.) of the Deutschland, showing the remarkable development in steam navigation during the last one hundred years. Loaned by the Hamburg-American line.

Portrait of Fulton painted by himself while a pupil of Benjamin West. Presented to the Society by Mrs. R. Anna Cary.

Fulton dining table, of mahogany, 8½ ft. long and 5 ft. wide. Presented to the Society by Thomas Egleston in 1891, who received it from his brother, George Egleston, into whose hands it had passed from Mrs. Egleston's House, to whom Robert Fulton gave it.

Two autograph drawings by Fulton, one of them a brush drawing of a high-level canal, presented by Cornelia J. Carll, showing his artistic ability as well as his mechanical genius. The other drawing is of the Sound steamer Fulton, built in 1813, and was presented by Louisa Lee Schuyler.

Drawing of the Fulton reproduced on a bronze tablet made in connection with the Fulton monument erected by the Society over Fulton's grave in Trinity churchyard. The tablet also gives a short description of the Clermont.

ROBERT FULTON

FROM A PAINTING BY HIMSELF, PRESENTED TO THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS BY MRS. R. ANNA CARY, NOVEMBER 30, 1897

Photograph of a letter by Fulton describing his experimental boat on the Seine, with copies of other letters by Fulton and his friends, presented by Henry Harrison Suplee, member of the Committee on Society History.

Hudson River Guide, published in 1850, describing the various points of interest on the Hudson and giving the time of sailing of the Hudson River boats of that period. Loaned by H. J. Gelien.

Oil painting of James Watt, a copy of a portrait by deBreda, now in the possession of John Scott of Hawkhill, Greenock, presented by past members of the Council.

Oil portrait of Capt. John Ericsson, painted by Ballin of Stockholm, showing the designer of the Monitor at the age of 59. A bust of Ericsson, by Knee-land, was presented to the Society by James Mapes Dodge, Past-President, Am.Soc.M.E.

Solid silver model of the Half Moon, loaned by Tiffany and Company.

Model of the Monitor, presented by Thomas F. Rowland, who built the Monitor at the Continental Iron Works.

Models, exhibited by Ericsson at the Centennial Exposition, loaned by the United Engineering Society.

Early books on steam navigation, from the Library of the Society.

Copy of a letter from Fulton to Boulton & Watt in 1810, ordering an engine for another boat.

Copies of letters by Fulton's workmen.

Drawings showing the comparative size of the White Star S. S. Olympic, Clermont and Half Moon; of New York's early water works, corner of Centre and Reade Streets; a water works note issue in 1774: all loaned by Daniel Arthur.

Color print of the Half Moon.

Fac-simile of Rules and Regulations for passengers in the Clermont.

Sketch of James Watt discovering the condensation of steam.

Photographs of a letter written by Fulton to Boulton & Watt, describing the Clermont and ordering an engine for another steamboat. Contributed by the Smithsonian Institution.

Print of the original oil painting by W. F. Halsoll of the battle between the Monitor and the Merrimac in the collection of the late Thomas Fitch Rowland, the builder of the Monitor.

Bills, receipts, etc., written by John Fitch; stock certificate issued by John Rumsey; copy of New York Herald of 1815 containing items of steamboats; copy of Washington Gazette of 1821 attacking Fulton: all loaned by Dr. C. S. Bullock of Stratford, Conn.

Silver Hudson-Fulton Medal, loaned by Dr. Geo. F. Kunz.

Descriptive Guide to the Grounds, Buildings and Collections of the New York Botanical Garden.

List of Prints, Books, Manuscripts, etc., relating to Henry Hudson, the Hudson River, Robert Fulton and Steam Navigation, at the Lenox Branch of the New York Public Library.

The Indians of Manhattan Island and Vicinity, by Alanson Skinner of the department of anthropology of the American Museum of Natural History.

The Wild Animals of Hudson's Day and the Zoölogical Park of our Day, by W. T. Hornaday, Sc.D., Director of the New York Zoölogical Park.

All of these loaned by Geo. F. Kunz.

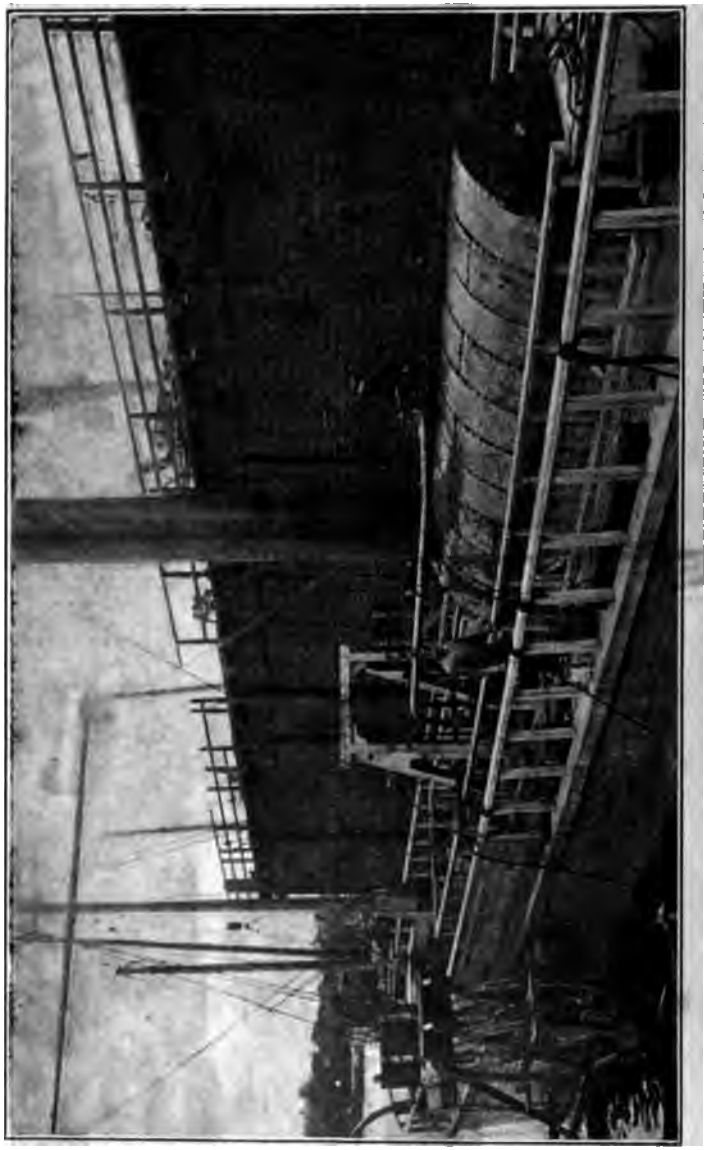
Considerable interest was manifested in this exhibit and it was attended. The total number of visitors who registered was 355, 400 is a fair estimate of the total. The greatest registration for one day was 52 on Monday, September 27. A wide extent of terri was represented, including Connecticut, Delaware, Illinois, M Massachusetts, Maryland, Michigan, Ohio, Pennsylvania, R Island, Tennessee, Washington, D. C., Wisconsin, Canada, J and Switzerland.

On the morning of September 24, in the presence of officers members of the Society and of the Pennsylvania Society, wre were placed on the Fulton Monument erected by the Society in in Trinity Churchyard, New York. The Pennsylvania Society represented by Robert Mazet, Vice-President, and Barr Ferree, retary. J. M. Schroeder, Commissioner from Pennsylvania, was present.

Members of both societies assembled in the lobby of the ch and, preceded by Dr. Manning, rector of Trinity Church, march the Fulton Monument. Dr. Manning offered prayer and wre were placed on the monument, after which the Lord's Prayer recited, followed by the benediction. "Taps" sounded by a Sev Regiment bugler brought the exercises to a close.

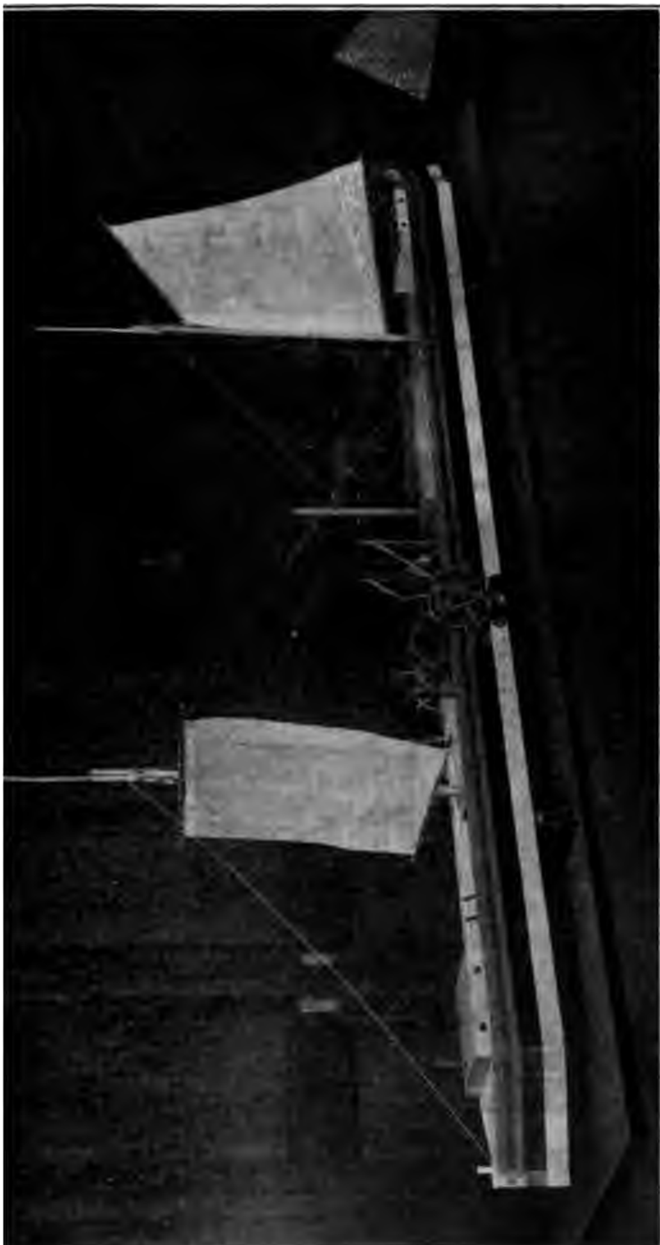
THE CLERMONT II ON HER OFFICIAL TRIAL TRIP

SHE WAS BUILT AT THE YARDS OF THE STATEN ISLAND SHIPBUILDING CO., FROM PLANS DRAWN BY F. E. KIRBY, MEM.AM.SOC.M.E., AND J. W. MILLARD



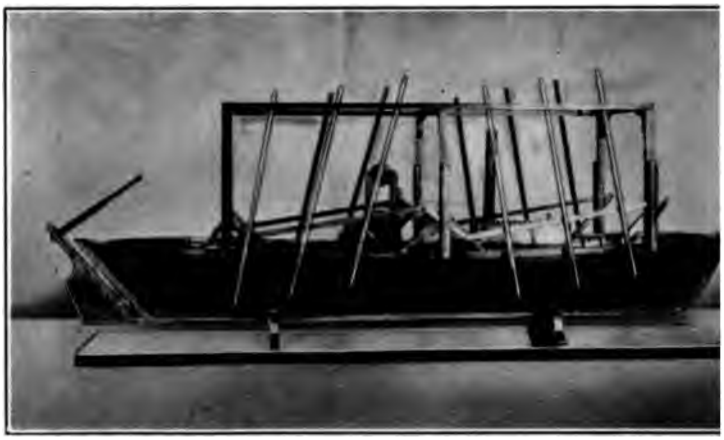
VIEW FROM THE STERN

SHOWING BOILER, STACK, CROSSHEAD GUIDE AND PADDLE WHEEL



MODEL OF THE CLERMONT AS SHE APPEARED ON HER INITIAL TRIP

FROM A PHOTOGRAPH TAKEN IN THE ROOMS OF THE SOCIETY



MODEL OF JOHN FITCH'S BOAT
BUILT IN PHILADELPHIA AND TRIED ON THE DELAWARE RIVER IN 1786

THE PHOENIX, BUILT BY JOHN STEVENS. IN 1809 SHE MADE AN OCEA
FROM NEW YORK TO PHILADELPHIA
FROM PHOTOGRAPHS MADE IN THE ROOMS OF THE SOCIETY

No. 1248

MEETINGS, OCTOBER-DECEMBER

NEW YORK MEETING, OCTOBER 12

At the meeting of the Society held October 12 in the Engineering Societies Building, Prof. R. C. Carpenter presented his paper on The High-Pressure Fire-Service Pumps of Manhattan Borough, City of New York. President Jesse M. Smith presided. The attendance was 192.

Secretary Calvin W. Rice read an invitation from the Institution of Mechanical Engineers to hold a joint session with them from July 26 to July 29, 1910. The President then introduced the following Japanese commissioners visiting the United States to study various industries: Dr. Ryota Hara, doctor of engineering and chief engineer of Yokohama; Rinnosuke Hara, of the Japanese Architectural Society; Junkichi Tanabe, of Tokyo, of the Institute of Japanese Architects; and Narazo Takatsuji, director of a large spinning factory. A telegram of regret was received from Kojiro Matsukata, the leading shipbuilder of Japan.

Those taking part in the discussion of the paper were: Prof. Geo. F. Sever, William M. White, Geo. L. Fowler, John H. Norris, J. R. Bibbins, J. J. Brown, Geo. A. Orrok, Frederick Ray, H. Y. Haden, Thos. J. Gannon, Henry B. Machen, Richard H. Rice, Chas. A. Hague, A. C. Paulsmeier, Prof. W. B. Gregory, Wm. O. Webber and Chas. B. Rearick.

At the close of the discussion Mr. White showed a number of lantern slides giving efficiency curves of various pumps designed by the I. P. Morris Co., Philadelphia, Pa.

Mr. Fowler, described the work of centrifugal pumps in dredging, and exhibited the following lantern slides, as evidence of the great suction capacity of these pumps:

A piece of shaft weighing 70 lb. raised and passed by a 15-in. dredging pump; improvement of New York Harbor, Steamer Reliance.

A piece of tree root raised and passed by a 12-in. pump from 14 ft. of water at Miami, Fla.; Florida East Coast Railway Company improvements.

A piece of pig iron measuring 11½ in. by 4¾ in. by 3½ in. and weighing 35 lb., raised and passed by an 8-in. special cataract wrecking pump from 15 ft. of water from the wreck of a canal boat sunk at Puas Dock, Yonkers, N. Y., by the Baxter Wrecking Company, New York.

ST. LOUIS MEETING, OCTOBER 16

A meeting of The American Society of Mechanical Engineers and the Engineers Club of St. Louis was held at the rooms of the latter organization at 8.15, Saturday evening, October 16, under the direction of William H. Bryan, Chairman, M. L. Holman and E. L. Ohle, Secretary, of the local joint committee.

A letter from President Jesse M. Smith was presented, indicating the sentiment of the Society towards local meetings. This was briefly responded to by President E. E. Wall, of the Engineers Club of St. Louis, who also emphasized his belief in the advantages of coöperation.

Prof. R. C. Carpenter of Cornell then presented in abstract his paper on The High-Pressure Fire-Service Pumps of Manhattan Borough, City of New York, accompanying it by running comments and comparisons.

He was followed by Horace S. Baker, assistant engineer of the City of Chicago, who presented with the aid of illustrations the results of recent study with a view to adopting high-pressure service. E. E. Wall, assistant water commissioner, City of St. Louis, outlined the plan proposed for high-pressure fire-service in St. Louis. He was followed by H. C. Henley, chief inspector, St. Louis fire prevention bureau, and vice-president of the National Fire Protection Association, expressing views of the fire insurance authorities, entirely favorable to the installation of such systems when properly designed and operated. Chas. E. Swingley, chief of the St. Louis fire department, on invitation made a few brief remarks to the effect that such systems were of undoubted advantage in the congested dictricts of large cities, and expressed the hope that something might soon be done along this line in St. Louis. There was further brief discussion by Edw. Flad, Prof. H. W. Hibbard, and H. C. Toensfeldt.

Luncheon was served by the Engineers Club of St. Louis. The attendance was 100.

BOSTON MEETING, OCTOBER 20

On Wednesday evening, October 20, a joint meeting of the Society with the Boston Society of Civil Engineers was held in the latter society's rooms, Tremont Temple, Boston, Mass.

Chas. T. Main, vice-president of the Boston Society of Civil Engineers, presided. Following the routine business of the Society of Civil Engineers, Mr. Main read a letter from Jesse M. Smith, President of The American Society of Mechanical Engineers, regretting his inability to be present at the meeting, and wishing the Boston members success for their coming meetings.

A paper by Cav. Gaetano Lanza, professor, and Lawrence S. Smith, instructor, at the Massachusetts Institute of Technology, on Stresses in Reinforced Concrete Beams, was read by the former. Following the presentation of the paper, a discussion by J. R. Worcester of Boston, Mass., was read. Sanford E. Thompson, Fred S. Hinds, Henry F. Bryant and Geo. F. Swain contributed oral discussions.

The total attendance at the meeting was 180, of whom 60 were members of the Society of Civil Engineers, 50 members of The American Society of Mechanical Engineers and 70 guests.

NEW YORK MEETING, NOVEMBER 9

At the meeting of the Society in New York on November 9, Professor Lanza presented his paper on Stresses in Reinforced Concrete Beams, and Professor Rautenstrauch his paper on Design of Curved Machine Members. The discussion on both papers proved valuable, the lantern slides shown in the discussion of Professor Lanza's paper adding much to its interest. Those participating were Sanford E. Thompson, E. P. Goodrich, Prof. Walter Rautenstrauch, Prof. W. H. Burr, B. H. Davis, of the Lackawanna Railroad, who showed slides of a number of concrete arches in railroad work, C. B. Grady of the New York Edison Company, who showed slides of beams and floor slabs under test, F. B. Gilbreth, who showed slides of the longest concrete beam of a rectangular section ever built in a roof, as well as other beams which had successfully passed through the fire and earthquake of the San Francisco disaster. Contributed discussions by Prof. J. C. Ostrup, E. L. Heidenreich, and C. E. Houghton were also presented. Professor Rautenstrauch's paper was discussed by a number of authorities on machine tool design, as follows: Professor Lanza, Chas. R. Gabriel, George R. Henderson, Professor Burr and Carl G. Barth. Those submitting written discussions were: C. E. Houghton, A. L. Campbell, H. Gansslen, F. I. Ellis, E. J. Loring and John S. Myers.

ST. LOUIS MEETING, NOVEMBER 13

At the meeting of the Society at St. Louis, November 13, with the Engineers Club of St. Louis, a description of the new plant of the Heine Safety Boiler Company of Boston was presented by E. R. Fish, Secretary of the Company, under the title, A Modern Boiler Shop. There was also further discussion of Professor Carpenter's paper on High-Pressure Fire-Service, continued from the October meeting.

BOSTON MEETING, NOVEMBER 17

A successful meeting of the Society was held at Boston in the Lowell Building, Massachusetts Institute of Technology, Wednesday evening, November 17. Two hundred and forty were present at this meeting and the Low-Pressure Steam Turbine was the topic of discussion.

Henry G. Stott of the Interborough Rapid Transit Company gave an interesting account of the difficulties encountered as well as the very fine results obtained from an installation recently made at the 59th Street Station of his company, New York. W. L. R. Emmet, engineer of the lighting department of the General Electric Company, described the low-pressure turbine situation from his viewpoint and pointed out the advantages of this type of prime mover for many mill installations and industrial works in New England. H. E. Longwell, consulting engineer of the Westinghouse Machine Company, and Edward L. Clark, manager of their Boston office, both spoke on the work that company is doing in this field. Max Rotter, turbine engineer of the Allis-Chalmers Company, pointed out in a humorous way a number of situations where the low-pressure turbine was not a desirable proposition. Professor Miller of the Massachusetts Institute of Technology also discussed the subject.

ST. LOUIS MEETING, DECEMBER 11

A meeting was held with the Engineers Club of St. Louis on Saturday evening, December 11, at the rooms of the latter society. The meeting was called to order by William H. Bryan, member of the Meetings Committee of the Society and chairman of the joint committee of the two societies at St. Louis. Prof. E. L. Ohle acted as secretary. There were present fifty-five members and guests.

The paper of the evening was by G. R. Parker of the General Electric Company, on The Relation of the Steam Turbine to Modern Central Station Practice, in which the underlying principles of modern steam turbines were discussed, together with the design of various prominent types on the market, and the developments made in recent years in improving capacity and efficiency. Attention was called to the large turbine capacity which may now be obtained within limited floor space; to the question of low-pressure turbines and their availability in supplementing standard reciprocating engines, increasing both their capacity and economy; also to the work already done in this direction at the plant of the Union Electric Light & Power Company in St. Louis, and to prospective work along similar lines in the same plant. The address was illustrated by lantern slides.

Discussion followed by Chairman Bryan, Prof. H. W. Hibbard, L. R. Day, E. R. Smith and Prof. E. L. Ohle.

On Saturday afternoon an excursion was made to the Ashley Street plant of the Union Electric Light & Power Company, for the inspection of the apparatus and equipment, on the invitation of John Hunter, chief engineer.

BOSTON MEETING, DECEMBER 17

On Friday evening, December 17, a goodly number of engineers of Boston and vicinity gathered on invitation of the local members of The American Society of Mechanical Engineers to discuss the Effect of Superheated Steam on Cast Iron. The meeting was called to order by Prof. Ira N. Hollis.

The committee which has been in charge of the meetings, consisting of Messrs. Hollis, Moultrop, Miller, Mann and Libbey, was continued.

The papers on the subject for the evening were then presented by their authors, Prof. Edward F. Miller of Boston, Arthur S. Mann of Schenectady, and Prof. Ira N. Hollis of Boston, and were discussed by B. R. T. Collins, George A. Orrok, Chas. H. Bigelow, W. K. Mitchell, John Primrose, L. B. Nutting, Wm. E. Snyder and others. The general purport of the discussion was rather reassuring to the users of cast-iron pipe and fittings, and to those who are interested in the extension of the use of superheated steam, in indicating that superheated steam *per se* has no injurious effect upon cast iron fittings, but that if the pipe lines are properly designed for the greater ranges of temperature, the fittings made adequate to the pressure and fluctuations in temperature avoided, the use of superheated steam introduces no piping difficulties which can not be easily overcome.

THE ANNUAL MEETING

The thirtieth annual meeting of The American Society of Mechanical Engineers was held in the Engineering Societies Building December 7 to 10, with an attendance of 628 members and 435 guests. For the first time at such a meeting the arrangements for the entertainment features were entirely in the hands of the local committee, the members in New York and vicinity acting as hosts, this method of handling an important part of the annual meeting being fully justified by the results. A feature of the meeting was an afternoon trip on Wednesday, through the new Pennsylvania Terminal, which brought out a large body of members and guests. The attendance at the reception on Thursday evening, held in the ball-room of the Hotel Astor, was nearly 600.

PROGRAM

OPENING SESSION

Tuesday, December 7, 8.30 p.m., Auditorium

THE PRESIDENT'S ADDRESS

THE PROFESSION OF ENGINEERING, by Jesse M. Smith

ELECTION OF OFFICERS

Report of Tellers of Election of Officers and introduction of the President-elect.

RECEPTION

The President and President-elect, with their ladies, received the members and guests in the rooms of the Society. Music and refreshments followed the reception.

BUSINESS MEETING

Wednesday, December 8, 9.30 a.m., Auditorium

Annual business meeting. Reports of the Council, Tellers of Election of membership, standing and special committees and Gas Power Section. Amendments to the Constitution. New business.

Luncheon was served to members and guests.

Wednesday afternoon

Excursion to points of engineering interest. Hosea Webster, Chairman Sub-Committee on Excursions.

LECTURE

Wednesday, 8.15 p.m., Auditorium

THE ERA OF FARM MACHINERY, L. W. Ellis, of the Bureau of Plant Industry of the United States Department of Agriculture at Washington, D. C. Illustrated by lantern slides.

PROFESSIONAL SESSIONS

Thursday, December 9, 9.30 a.m., Auditorium

MEASUREMENT OF THE FLOW OF FLUIDS

TESTS ON A VENTURI METER FOR BOILER FEED, Chas. M. Allen.

Discussed by F. N. Connet, Clemens Herschel, Dr. Sanford A. Moss, Prof. L. S. Marks.

EFFICIENCY TESTS OF STEAM NOZZLES, F. H. Sibley and T. S. Kemble.

Discussed by A. F. Nagle, A. R. Dodge, Prof. C. C Thomas, J. A. Moyer.

THE PITOT TUBE AS A STEAM METER, Geo. F. Gebhardt.

Discussed by Walter Ferris, A. R. Dodge, Prof. W. B. Gregory.

AN ELECTRIC GAS METER, C. C. Thomas.

Discussed by Prof. W. D. Ennis, E. D. Dreyfus, A. R. Dodge, Prof. L. S. Marks.

Luncheon was served to members and guests at the conclusion of the session.

Thursday, 2 p.m., Auditorium

STEAM ENGINEERING

TAN BARK AS A BOILER FUEL, David M. Myers.

Discussed by A. A. Cary, Prof. Wm. Kent, Prof. L. P. Breckenridge.

COOLING TOWERS FOR STEAM AND GAS POWER PLANTS, J. R. Bibbins.

Discussed by Geo. J. Foran, W. D. Ennis, H. E. Longwell, B. H. Coffey, E. D. Dreyfus, F. J. Bryant, Carl G. deLaval.

GOVERNING ROLLING MILL ENGINES, W. P. Caine.

Discussed by H. C. Ord, James Tribe.

AN EXPERIENCE WITH LEAKY VERTICAL FIRE TUBE BOILERS, F. W. Dean.

Discussed by R. P. Bolton, Prof. Wm. Kent, J. C. Parker, O. Woolson, A. A. Cary, Prof. A. M. Greene, Jr., E. D. Meier, D. Myers.

THE BEST FORM OF LONGITUDINAL JOINT FOR BOILERS, F. Dean.

Discussed by R. P. Bolton, Carl G. Barth, E. D. Meier, Prof. M. Greene, Jr., W. A. Jones, Prof. S. W. Robinson, Geo. I. Rockwood, Sherwood F. Jeter.

Thursday, 2 p.m., Lecture Hall

GAS POWER SECTION

Business meeting and election of officers.

TESTING SUCTION GAS PRODUCERS WITH A KOERTING EJECTOR, C. M. Garland, A. P. Kratz.

Discussed by Prof. R. H. Fernald, G. M. S. Tait, H. H. Suplee, L. B. Lent, S. C. Smith, W. B. Chapman, Edw. N. Trump.

BITUMINOUS GAS PRODUCERS, J. R. Bibbins.

Discussed by, G. M. S. Tait, Prof. R. H. Fernald, W. B. Chapman, H. M. Latham, H. H. Suplee, Edw. N. Trump, H. B. Langer, S. C. Smith, Prof. Walter Rautenstrauch, G. D. Conlee.

RECEPTION

Thursday, 9 p.m., Hotel Astor

The Members of New York and vicinity received the membership of the Society, their ladies and guests, at the Hotel Astor. Dancing and refreshments followed the reception.

PROFESSIONAL SESSION

Friday, December 10, 9.30 a.m.

THE BUCYRUS LOCOMOTIVE PILE DRIVER, Walter Ferris.

Discussed by O. K. Harlan, A. F. Robinson, L. J. Hotchkiss.

LINESHAFT EFFICIENCY, MECHANICAL AND ECONOMIC, Henry Hess.

Discussed by T. F. Salter, Prof. R. C. Carpenter, C. A. Graves, O.K. Harlan, C. J. H. Woodbury, Walter Ferris, Fred J. Miller, A. C. Jackson, C. D. Parker, Oliver B. Zimmerman, Geo. N. Van Derhoef.

PUMP VALVES AND VALVE AREAS, A. F. Nagle.

Discussed by Prof. Wm. Kent, A. B. Carhart, Prof. R. C. Carpenter, E. H. Foster, Chas. A. Hague, I. H. Reynolds, F. W. Salmon.

A REPORT ON CAST-IRON TEST BARS, A. F. Nagle.

Discussed by A. A. Cary, T. M. Phetteplace, Prof. W. B. Gregory, Geo. M. Peek.

COMMITTEES OF THE ANNUAL MEETING

MEETINGS COMMITTEE

Willis E. Hall, *Chairman*

William H. Bryan	Charles E. Lucke
L. R. Pomeroy	H. de B. Parsons

LOCAL COMMITTEE

William D. Hoxie, *Chairman* F. A. Scheffler, *Secretary*

Wm. L. Abbott	L. P. Breckenridge	George J. Foran
H. P. Ahrnke	Wm. H. Bryan	E. H. Foster
Louis Alberger	R. C. Carpenter	H. A. Foster
L. P. Alford	H. R. Cobleigh	Geo. L. Fowler
G. H. Barbour	F. H. Colvin	R. E. Fox
G. M. Basford	W. C. Dickerman	F. L. R. Francisco
Edgar H. Berry	Robert M. Dixon	John R. Freeman
Francis Blossom	F. L. DuBosque	H. L. Gantt
William H. Boehm	Frank E. Eberhardt	Fred J. Gubelman
Reginald P. Bolton	Harrington Emerson	Willis E. Hall
Geo. M. Bond	A. Falkenau	F. A. Halsey
G. I. Bouton	W. H. Fletcher	Geo. F. Hardy

Henry S. Hayward
G. R. Henderson
F. V. Henshaw
M. L. Holman
W. R. Hulbert
Alex. C. Humphreys
William F. Hunt
F. R. Hutton
F. E. Idell
H. S. Isham
E. B. Katte
R. S. Kent
Walter C. Kerr
Chas. Kirchhoff
J. W. Lieb, Jr.
Henry S. Loud
Fred R. Low
Chas. E. Lucke
R. C. McKinney
F. E. Matthews
E. D. Meier
Fred J. Miller
B. M. Mitchell
Chas. A. Moore
I. E. Moultrop
D. M. Myers
W. W. Nichols
J. H. Norris
H. de B. Parsons
E. H. Peabody
L. R. Pomeroy
H. O. Pond
H. F. J. Porter
W. P. Pressinger
Calvin W. Rice
A. L. Riker
Fred. E. Rogers
H. W. Rowley
W. J. Sando
E. F. Schnuck
Jesse M. Smith
Leo H. Snyder
Albert Spies
E. G. Spilsbury
J. E. Starr
Theo. Stebbins
A. F. Stillman
F. H. Stillman
H. G. Stott
H. H. Suplee
Ambrose Swasey
B. V. Swenson
F. H. Taylor
F. W. Taylor
Stevenson Taylor
Edw. Van Winkle
G. T. Voorhees
A. M. Waitt
F. A. Waldron
C. M. Wales
Arthur West
F. M. Whyte
W. H. Wiley
A. L. Williston
Ira H. Woolson

Sub-Committee on Finance

C. A. Moore, *Chairman*

W. L. Clark
W. C. Dickerman
Alex. C. Hump
W. C. Kerr

EXCURSION COMMITTEE

Hosea Webster, *Chairman*

George J. Foran
Percy C. Idell
Alfred F. Masu
Frederick A. Sc

BUREAU OF INFORMATION

F. E. Idell, *Chairman*

Charles C. Phelps
James V. V. C

ENTERTAINMENT COMMITTEE

Dr. D. S. Jacobus, *Chairman*

Sub-Committee on President's Reception

Tuesday Evening

Col. E. D. Meier, *Chairman*

W. C. Dickerman	Geo. L. Fowler	W. W. Nichols
Bernard V. Swenson	Willis E. Hall	H. F. J. Porter
Francis Blossom	Geo. F. Hardy	Albert Spies
Edward Van Winkle	Alex. C. Humphreys	E. G. Spilsbury
Louis Alberger	F. R. Hutton	John E. Starr
G. M. Basford	Chas. Kirchhoff	F. H. Stillman
Harrington Emerson	J. W. Lieb, Jr.	H. H. Suplee
Dudley Farrand	R. C. McKinney	Stevenson Taylor
W. H. Fletcher	Fred J. Miller	A. M. Waitt
C. H. Foster	B. M. Mitchell	Ira H. Woolson
	Charles A. Moore	

Sub-Committee on Reception and Dance

Thursday Evening

Prof. Arthur L. Williston, *Chairman*

Edgar H. Berry	Harrington Emerson	H. O. Pond
Wm. H. Boehm	E. H. Foster	L. H. Snyder
A. P. Boller, Jr.	Henry S. Hayward	Theodore Stebbins
Reginald P. Bolton	F. V. Henshaw	A. F. Stillman
G. I. Bouton	F. E. Matthews	J. Stewart Thomson
H. R. Cobleigh	David M. Myers	Edw. Van Winkle
Frank E. Eberhardt	J. H. Norris	F. A. Waldron
	E. H. Peabody	

Assignments for the Reception of Members

Tuesday

Afternoon	Evening
F. A. Halsey, *Chairman*	Chas. Whiting Baker, *Chairman*
H. P. Ahrnke	W. P. Pressinger
H. A. Foster	Fred. E. Rogers
H. S. Isham	H. M. Rowley

Wednesday

Morning	Afternoon	Evening
Albert Spies, *Chairman*	G. A. Orrok, *Chairman*	H. G. Stott, *Chairman*
Percy Allan	Geo. B. Caldwell	F. H. Colvin
C. G. de Laval	A. Falkenau	R. E. Fox, Jr.
F. H. Taylor	W. R. Hulbert	J. P. Sparrow

Thursday

Morning	Afternoon
F. L. Du Bosque, *Chairman*	James T. Whittlesey, *Chairman*
L. P. Alford	John J. Boyd
Geo. H. Barbour	N. B. Payne
Anson W. Burchard	R. P. Bolton

Friday Morning

Fred R. Low, *Chairman*
G. R. Henderson
E. B. Katte
Geo. Dinkel

LADIES' RECEPTION COMMITTEE

Mrs. Herbert Gray Torrey, *Chairman*

Mrs. H. C. Abell	Mrs. G. S. Humphrey	Mrs. Calvin W. Rice
Mrs. Charles W. Baker	Mrs. A. C. Humphreys	Mrs. E. N. Sanderson
Mrs. G. H. Barbour	Mrs. C. W. Hunt	Miss Marion R. Scheffler
Mrs. A. R. Baylis	Mrs. F. R. Hutton	Mrs. Horace See
Mrs. E. H. Berry	Mrs. F. E. Idell	Mrs. Jesse M. Smith
Mrs. Wm. H. Boehm	Mrs. P. C. Idell	Mrs. J. P. Sneddon
Mrs. R. P. Bolton	Mrs. D. S. Jacobus	Miss Jean Sneddon
Mrs. G. I. Bouton	Mrs. J. E. Jones	Mrs. H. H. Suplee
Mrs. Edward Ciardi	Mrs. J. A. Kinkead	Mrs. Stevenson Taylor
Mrs. J. VanV. Colwell	Mrs. G. L. Knight	Mrs. Edward Van Winkle
Mrs. H. Emerson	Mrs. J. W. Lieb, Jr.	Mrs. S. E. Whitaker
Mrs. G. L. Fowler	Mrs. H. S. Loud	Dr. Lucy O. Wight
Mrs. F. J. Gubelman	Mrs. Fred R. Low	Mrs. Wm. H. Wiley
Mrs. F. A. Hall	The Misses Meier	Mrs. A. L. Williston
Mrs. N. H. Hiller	Mrs. C. W. Obert	Mrs. Jas. Edw. Wilson
Mrs. H. F. Holloway	Mrs. G. A. Orrok	Mrs. Ira H. Woolson
	Miss Eugenie Price	

Chairmen Committees for the Several Days

Tuesday Afternoon	Mrs. John W. Lieb, Jr.
Wednesday Morning	Mrs. James Edward Wilson
Wednesday Afternoon	Mrs. Ira H. Woolson
Thursday Morning	Mrs. J. P. Sneddon
Thursday Afternoon	Dr. Lucy O. Wight
Friday Morning	Mrs. C. W. Hunt Mrs. F. A. Hall

Executive Committee

Mrs. Herbert Gray Torrey, *Chairman*

Mrs. F. A. Hall
Mrs. G. S. Humphrey
Mrs. C. W. Hunt
Mrs. J. E. Jones
Mrs. John W. Lieb, Jr.
Mrs. J. P. Sneddon
Dr. Lucy O. Wight
Mrs. James Edward Wilson
Mrs. Ira H. Woolson

LADIES' GUIDES

Mrs. G. S. Humphrey, Mrs. J. E. Jones } *Chairmen*

Mrs. Edward Ciardi
Mrs. John W. Lieb, Jr.
Miss Jean Sneddon
Mrs. J. P. Sneddon
Mrs. S. E. Whitaker
Dr. Lucy O. Wight

ACCOUNT OF THE ANNUAL MEETING

OPENING SESSION, TUESDAY EVENING

Vice-President Fred J. Miller called the session in the auditorium to order and presented President Jesse M. Smith, who delivered his address on The Profession of Engineering in which he dealt with the need of coöperation among engineers, looking toward the maintenance of high standards in engineering practice.

Following the address, Theodore Stebbins, chairman of the Tellers of Election, presented to the President the report on the election of officers and the following were thereupon declared elected: For president, George Westinghouse; for vice-presidents, Charles Whiting Baker, W. F. M. Goss, E. D. Meier; for managers, J. Sellers Bancroft, James Hartness, H. G. Reist; for treasurer, William H. Wiley.

President Smith then called on Past-Presidents Worcester R. Warner, Geo. W. Melville and Samuel T. Wellman to escort President-elect George Westinghouse to the platform.

After his notification of election and introduction to the members, the president-elect spoke as follows:

When Mr. Warner, the Chairman of your Nominating Committee, after first writing on the subject, came to Lenox to ask me to accept the nomination for president of this great Society, I had already decided that it would be impossible for me to have the privilege of accepting; but after he had explained to me the desires of his associates and had represented to me that it was the unanimous wish of all of the members of your Nominating Committee to honor me at this particular time, and in so doing to express an appreciation of my efforts and accomplishments in the engineering field, I with much hesitation consented to accept the nomination and promised if elected to do everything in my power.

Whether two mistakes have been made—one in yielding to the persuasive words of Mr. Warner, and the other in my election as your president—the forthcoming year will determine. I trust I may be able to fulfil your expectations by adding something to the worldwide reputation of The American Society of Mechanical Engineers.

With these remarks, I now accept with feelings of deep gratitude the honor which the members of the Society have tonight unanimously conferred upon me.

There never was a time in the history of the world when honest, wise and conservative action is more strongly demanded of us and of all men than now, if we have any desire to preserve the right to comfortably carry on our various affairs.

I thank you, and I ask your coöperation in my efforts to perform my duties as your president.

The meeting was then adjourned to the rooms of the Society where the members and guests were introduced by Secretary Calvin W. Rice, to the President-elect and Mrs. Westinghouse, who were assisted in receiving by President Jesse M. Smith and Mrs. Smith, and Honorary Secretary F. R. Hutton and Mrs. Hutton.

WEDNESDAY EVENING LECTURE

As already stated the lecture on Wednesday evening was on the Era of Farm Machinery, by L. W. Ellis, of the Bureau of Plant Industry of the United States Department of Agriculture at Washington, D. C. The lecture was illustrated by lantern slides. Mr. Ellis first gave an idea of agricultural progress, by describing some of the most striking mechanical achievements found on Western farms of the present day. He first described early farm implements and told briefly of the transition from hand to machine methods. In 1800 wheat was sown broadcast by hand, after the ground had been plowed with a heavy, clumsy, wooden plow, requiring as many as eight oxen to pull it. Sickles cut the grain, and it was bound by hand. During the succeeding winter it was threshed out either by a flail or by driving animals over it as it lay in heaps. It was finally winnowed by hand.

Corn cultivation was by the hoe, or a rude shovel plow. The stalks were cut and the ears husked out by hand. Shelling was done by scraping the ears against the handle of a frying pan, a bushel in one hundred minutes.

Hay was cut with a scythe and was pitched by hand from ground to cart, and cart to haymow. Baling and shipping were practically unknown. Hand methods prevailed in the dairy, the stable, the cotton fields, the potato patch, in fact in every phase of production.

From 1855 to 1894 the human labor consumed in producing a bushel of corn by the best available methods declined from four hours and thirty minutes to forty-one minutes, and for shelling it from one hundred minutes to one minute. In 1830, three hours and three minutes of human labor were required to raise and thresh a bushel of wheat; in 1896 ten minutes. Eleven hours were required to cut and cure a ton of hay in 1860, and but one hour and thirty-nine minutes in 1894.

Power corn shellers now used have a capacity of from one hundred to eight hundred bushels per day. The cobs are carried to a pile and the shelled corn delivered into sacks or wagons. The fuel value of the cobs pays the cost of shelling.

Though hand methods still prevail in some sections, the mower is now practically the universal means of cutting the hay crop. This is a modification of the early reaping machines with such factors eliminated as are not necessary for cutting the grass. The steel self-dump rake, the side-delivery rake and the hay loader, the stacker, and the baling press are other developments for hay harvesting.

In the extreme West there has been developed the combined harvester which seems to represent the greatest possible saving of human labor. This machine, drawn by from twenty to forty horses, under control of a single driver, cuts, threshes, recleans, and delivers into sacks the grain from forty to fifty acres per day. Two men are required for sewing the sacks. The straw, including all weed seeds, is distributed over the ground as the team proceeds. On level land the horses may be replaced by the steam engine, which furnishes power sufficient to cut a swath up to forty feet in width and to cover from seventy-five to one hundred and twenty-five acres per day.

For general farm work the internal-combustion tractor may be said to be rapidly supplanting the steam engine, which, however, has a great field of usefulness in sections where it is desired to bring large areas rapidly under cultivation. In older sections, in order to compete successfully with the horse, tractors must bring the cost of operation close to the cost with horses and at the same time be capable of a great variety of work. The internal-combustion tractor meets these conditions better than the steam engine, and is being introduced at a rate estimated anywhere from two thousand to five thousand per year.

The automobile is rapidly finding a place in the business management of the farm. It takes from the heavy draft horse the necessity for long, exhausting trips to town on light errands.

In general, machinery has reduced the cost of producing farm products. It has improved the quality of products by condensing crop operations within the period when the most favorable conditions prevail. By increasing the acre effectiveness of a man it has reduced the labor necessary to produce the nation's food supply, leaving it free to assist in development along other lines. At the same time it has thrown upon the cities the burden of providing work for an ever-increasing army of non-producers. It has increased the investment necessary for the proper organization of a farm, this and the price of land making it more difficult for a person of small capital to engage in farming.

As a nation we have occupied nearly all of our naturally productive area and are confronted with the necessity of providing food for an increasing population with a constant acreage. In the past, machinery has encouraged extensive rather than intensive farming. Henceforth the reverse should be true. If he who makes two blades of grass grow where one grew before, is a public benefactor, then none the less is he a public servant who puts into the farmer's hands the machinery for making such a course attractive.

BUSINESS MEETING

The business session on Wednesday morning was called to order by President Jesse M. Smith. Secretary Calvin W. Rice read the annual report of the Council. The Secretary then read the report of the Tellers of Election of members, including 166 applicants for membership and 21 for advance in grade.

The next in order was the consideration of the proposed amendments to the Constitution. The first amendment relates to C 10 on associate membership, which reads as follows:

C 10 An Associate shall be 26 years of age or over. He must either have the other qualifications of a member or be so connected with engineering as to be competent to take charge of engineering work, or to coöperate with engineers.

The proposed amendment reads as follows:

An associate member shall be thirty years of age or over; he must have been so connected with some branch of engineering, or science, or the arts, or industries, that the Council will consider him qualified to coöperate with engineers in the advancement of professional knowledge.

Another amendment relates to the clause on Junior Membership which now reads as follows:

C 11 A Junior shall be 21 years of age or over. He must have had such engineering experience as will enable him to fill a responsible subordinate position in engineering work, or he must be a graduate of an engineering school.

The following addition is proposed by the Committee on Constitution and By-Laws:

A person who is over 30 years of age can not enter the Society as a Junior.

Both these amendments have been approved by the Committee on Membership. It therefore remains for the members to vote on them by letter ballot.

A third proposed amendment to the Constitution relates to the formation of an additional standing committee. This was presented at the Washington meeting in the form of a resolution, as follows:

Resolved, That we recommend to the Council the appointment of a Public Relations Committee, to investigate, consider and report on the methods whereby the Society may more directly coöperate with the public on engineering matters and on the general policy which should control such coöperation.

It was moved and seconded that this also be referred to the members for letter ballot.

Dr. D. S. Jacobus, Chairman of the Committee on Power Tests, then made a verbal report. This committee was appointed to revise all the codes relating to power tests, some of which did not agree with others, or were not up to date. It had been decided to blend the whole into one report rather than present a series of reports, as on engine testing, boiler testing, etc. The first part of the report will deal with tests in general, calibration of apparatus, units, etc., while the second part will be subdivided for the various classes of machines and apparatus.

Dr. Jacobus also made a verbal report for the Joint Committee on a Standard Tonnage Basis for Refrigeration. This committee had made a preliminary report in 1904 and suggested certain units for measuring the refrigerating capacity of the machinery. They had also suggested a standard set of conditions under which a machine should be tested to obtain the refrigerating capacity of that machine. Later on, the work of the committee was extended, and they were asked to recommend a method of testing the machines. A preliminary report was also prepared on this portion of the work and had been before the Society.

Though the committee had received some favorable discussion on the report they felt that it was not a complete piece of work, and they wished that some one would give the committee additional light on how the report could be made. Furthermore, there were many places in the report where the committee could not make any definite recommendations, because they did not have enough data at hand.

A résumé of the work that has been done by the Committee on Refrigeration was prepared and sent to the Congress of Refrigerating Industries, held in Paris in the fall of 1908, with the request that it be discussed. In making this résumé certain questions were asked, on which the committee wished to obtain specific information. This was done in a semi-official way, and after taking up the matter with the Secretary of this Society, Dr. Jacobus, speaking on the behalf of the committee, concluded the communication to the International Committee as follows:

The policy of The American Society of Mechanical Engineers has always been for the advancement of the arts, and whereas it is only natural that it should take pride in participating in advancements, it will never look except with satisfaction upon activities of other bodies, even in the subjects on which it has worked.

I feel safe in saying, therefore, that any criticism by the members of this organization on the work which has been done in connection with the subject at hand will be gladly received. Criticism leads to the establishment of better and more up-to-date methods, and what The American Society of Mechanical Engineers is after, and what I am sure we are all after, is to work hand in hand for the good of the cause.

I also feel safe in saying that The American Society of Mechanical Engineers will coöperate in every way in the endeavor to establish some standard set of rules which shall conform with the views of such able experts as are gathered in this meeting. It is certainly hoped that the matter presented in this paper will receive a thorough discussion, irrespective of whether those who take part agree or disagree with the findings of the committee.

About the same time, a request was made by the committee that it should be allowed to coöperate with a committee of the American Society of Refrigerating Engineers, so that if this general committee recommended certain units, they would really be used by both societies. A committee of five was appointed by the American Society of Refrigerating Engineers to coöperate with the committee of five of The American Society of Mechanical Engineers. This combined committee had already held one meeting and sent out a circular letter to a number of refrigerating engineers, reviewing the units that had been recommended by the Society, and asking for an opinion regarding these specific units. A great number of replies had been received, showing how much interest there is in the subject. Most of the replies said either that the units were acceptable to those who had read the letter, or that they would leave the selection of the units entirely in the hands of the committee. The committee therefore has a very good working basis, and hopes within a comparatively short time to be able to present the results of its work.

Dr. C. E. Lucke then abstracted the report of the Gas Power Standardization Committee, of which he is chairman. The report was discussed by Dr. D. S. Jacobus, Prof. R. H. Fernald, A. A. Cary, Edwin D. Dreyfus and L. B. Lent.

The report of the Gas Power Plant Operations Committee was presented by F. R. Low in the absence of I. E. Moultrop, chairman of the committee. The report was discussed by Prof. R. H. Fernald, Edwin D. Dreyfus, and Arthur J. Wood.

THURSDAY MORNING SESSION

The Thursday morning session was devoted to papers on the measurement of the flow of fluids.

The first paper presented was on Tests on a Venturi Meter for Boiler Feed, by Prof. C. M. Allen, of Worcester Polytechnic Institute. The object of these tests with the venturi meter was to determine how well adapted it would be for use in measuring the feed to a boiler, in view of the variety of conditions under which it might have to operate such as the methods of pumping the water through the meter, the different temperatures of the water pumped, various and fluctuating pressures and velocities of flow, etc. The results obtained indicate that such occurrences have practically no effect on the satisfactory performance of the meter. Though there are limits to the satisfactory operation of a meter, the tests indicate that the venturi meter is sufficiently accurate for the majority of commercial or engineering requirements.

The paper was discussed by F. N. Connet and Clemens Herschel, Dr. Sanford A. Moss and Prof. L. S. Marks submitting written discussions.

The next paper, Efficiency Tests of Steam Nozzles, by Prof. F. H. Sibley of the University of Alabama, was read by Prof. C. C. Thomas of the University of Wisconsin. The object of the test was to determine the efficiency of various shaped nozzles with steam flowing from a given initial pressure to a known vacuum; also to determine the effect on the efficiency of changing the angle of divergence. Two methods were tried out for finding this efficiency: (*a*) by the pressure in the nozzle by means of a search tube placed axially in the nozzle; (*b*) by the reaction of the nozzle by suspending it in an air-tight box at the end of a flexible steel tube. The deflection of the tube caused by the reaction of the nozzle was measured by a calibrated spring. The results of the tests indicate: (*a*) that the reaction is affected by a difference in pressure between the muzzle of the nozzle and the medium surrounding the nozzle; (*b*) that the efficiencies of the various nozzles were determined within a probable error of 2 per cent; (*c*) that the efficiency is affected more by the smoothness of finish on the inside of the nozzle than by the exact contour of the nozzle.

A. F. Nagle, A. R. Dodge and Professor Thomas discussed the paper, J. A. Moyer submitting a written discussion.

George F. Gebhardt's paper on The Pitot Tube as a Steam Meter was read by the Secretary in the author's absence. The application of a pitot tube system as described in the paper is an accurate means of determining the *velocity* of steam at any point in a pipe, provided the values of the various influencing factors are known; and for straight lengths of piping with continuous flow, under these conditions, it is an accurate means of determining the *weight* of steam flowing. Under average commercial conditions in which the pressure and quality of the steam fluctuate and an average value must be taken for the density of the self-adjusting water column, only approximate results can be obtained, the extent varying with the degree of fluctuation.

Walter Ferris and A. R. Dodge discussed the paper, a written discussion by Prof. W. B. Gregory being read by the Secretary.

The paper on An Electric Gas Meter was presented by the author, Prof. Carl C. Thomas, of the University of Wisconsin. The paper describes a meter for measuring the rate of flow of gas or air, which can be adapted for use as a steam meter or as a steam calorimeter. The operation of the gas meter depends upon the principle of adding electrically a known quantity of heat to the gas and determining the rate of flow by the rise in temperature of the gas (about 5 deg. fahr.) between inlet and outlet. The adoption of this principle of operation permits the construction of a very accurate and sensitive autographic meter of large capacity containing no moving parts in the gas passage; independent of fluctuations in pressure and temperature of the gas; and capable of measuring gas or air at either high or low pressures or temperatures. The electrical energy required is about 1 kw. per 50,000 cu. ft. hourly capacity, at the pressures ordinarily used in gas mains.

Prof. W. D. Ennis, E. D. Dreyfus and A. R. Dodge discussed the paper, a written discussion from Prof. L. S. Marks being also read.

THURSDAY AFTERNOON—STEAM ENGINEERING

At the Thursday afternoon session Vice-President L. P. Breckenridge presided. Five papers were presented dealing with different phases of steam engineering. The first paper, Tan Bark as a Boiler Fuel, by David M. Myers, described the results obtained by burning spent hemlock tan bark, the average fuel value of which is about 9500 B.t.u. per lb. of dry matter, which is about 35 per cent of its total moist weight in the fireroom. The available heat value per pound as fired is 2665 B.t.u. One ton of air-dry hemlock bark produces boiler fuel equal to 0.42 tons of 13,500 B.t.u. coal. A. A. Cary, Prof. Wm. Kent and Prof. L. P. Breckenridge took part in the discussion.

J. R. Bibbins then presented his paper on Cooling Towers for Steam and Gas Power Plants, which contained a critical study of different types of towers with a description of their distinctive features. The paper also describes a simple inexpensive type of tower employing a lath-mat cooling surface and offers suggestions for a combination of natural-draft and forced-draft types.

The paper was discussed by Geo. J. Foran, W. D. Ennis, H. E. Longwell, B. H. Coffey, E. D. Dreyfus and F. J. Bryant. A written discussion by Carl G. de Laval was read by the Secretary.

W. P. Caine's paper, Governing Rolling Mill Engines, was read by Richard H. Rice. The paper describes and gives indicator cards and speed curves of a Corliss engine driving a three-high mill under two different conditions of governing, (*a*) under the widest range of adjustment of cut-off, (*b*) under a limited range, increasing the economy and making the engine run much more smoothly and safely. A table gives the power required for rolling in the mill and the momentary source of energy, whether from the cylinder or flywheel. A description is also given of the tachometer used to take the speed curves. Written discussions by H. C. Ord and James Tribe were read by the Secretary.

The next paper was that by F. W. Dean on An Experience with Leaky Vertical Fire-Tube Boilers. The author discussed the difficulties experienced with some large vertical boilers, somewhat over 10 ft. in diameter, and containing over 6000 sq. ft. of heating surface. The boilers leaked badly very soon after being started and nothing that was done improved their condition until the water legs were lengthened from 2 ft. to 7 ft. $2\frac{3}{4}$ in., the boilers thus being raised 5 ft. $2\frac{3}{4}$ in. Before they were raised the lower ends of the tubes would cover with very hard clinker and become stopped up. This clinker could be removed only by cutting it off when the boilers were cold. After the boilers were raised, a light clinker that could be blown off formed about the tubes; by removing this by blowing every three or four hours the leaks were stopped and they have never returned.

Those taking part in the discussion were R. P. Bolton, Prof. Wm. Kent, J. C. Parker, O. C. Woolson, A. A. Cary, Prof. A. M. Greene, Jr., E. D. Meier and D. M. Myers. A. Bement submitted a written discussion.

Mr. Dean's second paper, The Best Form of Longitudinal Joint for Boilers, dealt with the defects of the usual form of butt joint used on the longitudinal seams of boilers, in which the inside strap is wider than the outside strap. It gave some history of the joint and discussed some of its defects and suggested a substitution for this form.

The paper was discussed by R. P. Bolton, Carl G. Barth, E. D. Meier, Prof. A. M. Greene, Jr., W. A. Jones, Prof. S. W. Robinson, Geo. I. Rockwood, and Sherwood F. Jeter.

GAS POWER SECTION

The session of the Gas Power Section was held on Thursday afternoon, Chairman F. R. Low presiding. In his address, the Chairman referred briefly to the work of the various committees of the Section and stated that during the year the membership had increased from 247 to 378, a gain of over 50 per cent. Mr. Low also dealt with the development in the gas-power field during the year, mentioning some experiments with gas turbines. Gas-engine design, the use of by-product gases, the development of the bituminous producer, the gasification of peat, and the gas engine in marine work, were also briefly dealt with.

The report of the Tellers of Election, Edw. Van Winkle, Prof. Walter Rautenstrauch and J. V. V. Colwell, was then presented by Prof. Rautenstrauch, the results being as follows: for chairman J. R. Bibbins 107; for member of the Executive Committee, F. R. Low 108.

The report of the Gas Power Plant Operations Committee was then presented by James D. Andrew, and discussed by J. C. Parker, J. N. Norris and H. H. Suplee. Prof. C. H. Benjamin reported verbally for the Literature Committee, outlining the work of the committee in bringing gas-power literature to the attention of the members. H. R. Cobleigh and Professor Rautenstrauch also spoke on the work of this committee, the latter suggesting a plan for better organization of the committee to deal with literature on the subject.

L. B. Lent reported for the Gas Power Installations Committee that two forms had been prepared and sent to manufacturers, and while a good deal of information had been received, not enough was on hand for a complete report. The committee hoped to have the material in shape at an early date.

Prof. W. F. M. Goss then presented the paper on Testing Suction Gas Producers with a Koerting Ejector, by C. M. Garland and A. P. Kratz. The paper describes a method of testing the suction gas producer which is independent of the engine. The engine is blanked off from the producer and a Schutte & Koerting steam ejector is inserted, which draws the gases from the producer and delivers them to a scrubber in which the steam used by the ejector is condensed. The gases then pass to a meter for measuring their volume. Complete data of calculations and results are given in appendices.

The paper was discussed by Prof. R. H. Fernald, G. M. S. Tait, H. H. Suplee, L. B. Lent, S. C. Smith, W. B. Chapman and Edw. N. Trump.

The paper on Bituminous Gas Producers was then presented by the author, J. R. Bibbins. The paper describes a double-zone type of producer and the results obtained in gasifying bituminous coal. Continuous operation was secured with tar-free gas of reasonable heat value and producer efficiency and an over-all plant economy of about one pound of fair bituminous coal per brake horsepower (proportionate economies for poorer grades). The efficiency and general effectiveness of operation of the producer on low-grade fuel, lignites, etc., was practically as high as with the higher grades. The following took part in the discussion: G. M. S. Tait, Prof. R. H. Fernald, W. B. Chapman, H. M. Latham, H. H. Suplee, Edw. N. Trump, H. B. Langer, S. C. Smith, Prof. Walter Rautenstrauch, and G. D. Conlee.

FRIDAY MORNING

The session on Friday morning opened with the paper by Walter Ferris on The Bucyrus Locomotive Pile Driver. This paper describes a new railway pile driver, the leading feature of which is a very powerful propelling apparatus and a large boiler, enabling it to act as a locomotive and haul its own train of tool cars, boarding cars, etc., over the road. A special turntable, consisting of hydraulic lifting apparatus and a large ball-bearing, enables the entire pile driver, including trucks, to be turned end for end or crosswise of the tracks. O. K. Harlan discussed the paper, A. F. Robinson and L. J. Hotchkiss submitting written discussions.

The paper by Henry Hess on Lineshaft Efficiency, Mechanical and Economic, described the test of the relative efficiency of a lineshaft of $2\frac{7}{16}$ in. diameter, making 214 r.p.m., with bearing load due to the weight of the parts plus the tension of the belts subjected to known stress by counterweighting, when running in ring-oiling babbitted bearings and when mounted in ball bearings. The savings in power consequent on this change ranged from 14 to 65 per cent, with 36 and 35 per cent under average conditions of good practice, due to belt tensions of 44 lb. and 57 lb. per inch width of single belt respectively. The paper gives data for determining the power savings that may be expected in various plants, by the use of ball bearings.

Those discussing the paper were T. F. Salter, Prof. R. C. Carpenter, C. A. Graves, O. K. Harlan, C. J. H. Woodbury, Walter Ferris, Fred J. Miller, A. C. Jackson, C. D. Parker and Oliver B. Zimmerman. Geo. N. Van Derhoff submitted a written discussion.

A. F. Nagle's paper on Pump Valves and Valve Areas, called the attention of engineers to the need of reviewing the common notion that "valve-seat area" is synonymous with "velocity of flow." The purpose of specifications for pumping engines is to secure a low velocity of flow through the valves, thus reducing the head required to force water through the pump; but to accomplish this purpose, special and intelligent attention should be given to the springs of the valves, rather than to valve-seat areas. If that be done, valve-seat areas need not be greater than the plunger area for the vertical triple-expansion pumping engines so largely used in city pumps. Prof. Wm. Kent, A. B. Carhart, Prof. R. C. Carpenter and E. H. Foster discussed the paper. Contributed discussions were by Chas. A. Hague, I. H. Reynolds and F. W. Salmon.

Another paper by Mr. Nagle, A Report on Cast-Iron Test Bars, brought out the fact that test pieces, whether cast in separate molds or in the same mold as the main casting, are not perfect indications of the character of the iron in the main casting. The results obtained by the author would indicate a probable variation of 15 per cent where uniformity might be expected. A. A. Cary and T. M. Phetteplace discussed the paper, contributed discussion being by Prof. W. B. Gregory and Geo. M. Peek.

The meeting closed with the following resolutions, offered by Luther D. Burlingame:

Whereas The American Society of Mechanical Engineers at its Annual Meeting, December 1909, desires to express its appreciation to those who have provided opportunities for entertainment and on behalf of the visiting members and their guests thanks for the cordial welcome extended by the local members and their friends of New York and vicinity,

Be it Resolved that the Secretary extend the thanks of the Society and express the appreciation of its members and guests to the local committee for their untiring efforts, to those who have sent invitations to visit technical and engineering works and places of interest, to Mr. Geo. Gibbs, chief engineer of the Pennsylvania Tunnel and Terminal Railroad Co., and to Mr. Walter Kerr, president of the Westinghouse, Church, Kerr & Co., and their associates, for the opportunity to inspect the new Pennsylvania Railroad station; to Dr. B. T. Galloway, chief of the Bureau of Plant Industry, Department of Agriculture, for the very instructive and entertaining paper on The Era of Agricultural Machinery, and especially to those ladies who have so efficiently assisted by extending a generous hospitality to their guests.

EXCURSIONS

As usual at conventions of the Society there were numerous excursions to points of interest in New York and vicinity, which constituted an important feature of the program for the entertainment of visiting members and guests. Invitations for these excursions were generously extended by many firms and individuals, and through the efforts of the Excursion Committee, Hosea Webster, Chairman, trips to various plants and industries were arranged, to the representatives of which the grateful appreciation of the Society has been expressed.

A list of excursions follows:

Pennsylvania Railroad Terminal and Passenger Station: Invitation by George Gibbs, Chief Engineer, Pennsylvania Tunnel Terminal R. R. Co., and member of the Society; Henry R. Worthington Hydraulic Works, Harrison, N. J., by William Schwanhausser, Chief Consulting Engineer of International Steam Pump Co., member of the Society; Harrison Lamp Works of General Electric Co., Harrison, N. J., by George H. Morrison, General Manager; Interborough Rapid Transit Co., central power station at 59th St., New York, by H. G. Stott, Superintendent of Motive Power, Manager of the Society; Edison factories and Edison Laboratory at Orange, N. J., by Frank L. Dyer, President of National Phonograph Co., asso-

ciate member of the Society; De La Vergne Machine Co., New York, by Adolf Bender, President; New York Telephone Co.; Gramercy and Stuyvesant Central Offices, by E. F. Sherwood, Superintendent of Traffic; Crocker-Wheeler Co., Ampere, N. J., by S. S. Wheeler, President, member of the Society; Westinghouse Lamp Co., Bloomfield, N. J., by Walter Carey, General Manager; New York Edison Co., Waterside Stations Nos. 1 and 2, by John W. Lieb, Jr., 3d Vice-President, member of the Society; Astoria Light, Heat & Power Co., Astoria, N. Y., by William H. Bradley, Chief Engineer, Consolidated Gas Co., member of the Society; Brooklyn Rapid Transit Co., Williamsburg Power Station, by C. E. Roehl, Electrical Engineer; Rockland Electric Co., Hillburn, N. Y.; Singer Building, New York, by Singer Mfg. Co.; Trenton Iron Co., Trenton, N. J.; Watson-Stillman Co., Ampere, N. J.; Metropolitan Life Insurance Building, New York.

Every possible courtesy was extended to the visiting parties in each case and in some instances special transportation facilities were provided. At the Edison Laboratory visitors were met by Thomas A. Edison, Hon.Mem.Am.Soc.M.E., who personally explained many points of interest about the plant. The Information Bureau, located in the foyer of the building, under the chairmanship of F. E. Idell, was of material aid in this connection with the trips.

ENTERTAINMENT FEATURES

The Ladies' Reception Committee, composed of ladies resident in and about New York, under the chairmanship of Mrs. Herbert Gray Torrey, contributed much to the pleasure of members and guests of the Society. Tea was served from four until six o'clock on Tuesday, Wednesday and Thursday afternoons during the convention, in the ladies' headquarters, located in the reception rooms of the Society on the eleventh floor. Mrs. George H. Westinghouse was the guest of the committee on Wednesday afternoon.

A number of excursions to shops and hotels were arranged and successfully carried out under the guidance of members of the committee The kindness of Mr. and Mrs. John W. Lieb, Jr., made possible several enjoyable automobile rides through Central Park and Riverside Drive-

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THE ANNUAL REPORT OF THE COUNCIL AND COMMITTEES 1909

REPORT OF THE COUNCIL

The Society entered upon a distinct epoch in its history when the Council approved the recommendation of the Meetings Committee that meetings of the Society be held periodically in cities other than New York, thus satisfying a long-felt desire on the part of the membership, as well as of the Council and the Meetings Committee, to extend as fully as possible the benefits of membership in the Society.

As a result meetings have been successfully held in Boston and St. Louis. In the former place, the attendance has been even larger in some cases than the meetings in New York. The spirit of coöperation has been developed and although these are meetings of the Society, fellowship in the profession has been promoted in each center by the participation in the meetings of the membership of local engineering societies and engineers generally.

Inquiries are constantly being received from other centers for information respecting the holding of meetings, and every encouragement is being rendered and assistance pledged by the Society to make it possible for groups of the members in any locality to hold meetings.

Through the policy of conducting these meetings as meetings of the Society rather than of sections or branches, the solidarity and national character of the Society is at once developed. All meetings are conducted in all places on the same basis with an equally high standard and before publication in The Journal all papers and discussions thereon must be approved by the same committee, viz., the Meetings Committee; and no papers may be read or discussed that are not of a uniformly high grade and suitable and worthy of publication for the benefit of the entire membership.

STUDENT BRANCHES

The number of student branches affiliated with the Society which have been formed in colleges and universities during the past year show the importance of another movement. Seventeen of these branches have been established and the reports of their meetings which have appeared at intervals in The Journal indicate a keen interest on the part of these organizations and show that here is a work that the Society may well foster. The basis of affiliation of these student societies with The American Society of Mechanical Engineers is a broad one, and provides for the maintaining of each branch under its own by-laws subject only to limitations set by the Council of the Society. The Journal is furnished to each member for the nominal sum of $2 a year and, in addition, advance copies of papers to be presented before the Society are supplied gratis for discussion at meetings. Papers for local representation may also be printed and supplied at cost to the affiliated branches. A list of the branches includes: Stevens Institute of Technology, Hoboken, N. J.; Cornell University, Ithaca, N. Y.; Armour Institute of Technology, Chicago, Ill.; Leland Stanford Jr. University, Palo Alto, Cal.; Polytechnic Institute of Brooklyn, Brooklyn, N. Y.; State Agricultural College, Corvallis, Ore.; Purdue University, Lafayette, Ind.; University of Kansas, Lawrence, Kan.; New York University, New York; University of Illinois, Urbana, Ill.; Pennsylvania State College, State College, Pa.; Columbia University, New York; Massachusetts Institute of Technology, Boston, Mass.; University of Cincinnati, Cincinnati, O.; University of Wisconsin, Madison, Wis.; University of Missouri, Columbia, Mo.; University of Nebraska, Lincoln, Neb.

HUDSON-FULTON EXHIBIT

The Society's part in the recent Hudson-Fulton celebration was the preparation of an interesting exhibit of steamboat models, drawings, portraits, books, manuscripts, and other material related to the development of steam navigation. In making this exhibit, the Society had the hearty coöperation of the Smithsonian Institution and of the Hamburg-American line, and was able to place on view models of early and modern steamboats, the American Museum of Natural History loaning show cases for this purpose, and members and friends of the Society also helping to make the exhibit of interest by loaning or presenting manuscripts, books and drawings. The American Society of Mechanical Engineers was the only engineering organization as such taking part in the celebration of engineering achievement.

Representatives of the Society, together with the Pennsylvania Society, on September 24th placed a wreath on the Fulton monument erected by this Society in Trinity churchyard. The Rev. Dr. William T. Manning, Rector of Trinity Church, conducted the service.

A description of the improvements in the decorations and the rearrangement of the rooms of the Society is contained in the Annual Report of the House Committee.

The same report contains also a description of the mahogany desk formerly belonging to Edwin Reynolds, Past-President of the Society, donated to the Society by Mrs. Reynolds.

THURSTON MEMORIAL

As stated in the Transactions of last year, permission was obtained from the Alumni Committee of Sibley College, Cornell University, to place in the rooms of the Society, a bronze replica of the Thurston memorial tablet at Cornell University. Arrangements for its execution were made with the sculptor, H. A. MacNeil, a personal friend of Dr. Thurston, and the tablet is now in place in the entrance hall. The figure is about three-quarter life size and below it is the inscription.

1839 ROBERT HENRY THURSTON 1903

FIRST PRESIDENT

AMERICAN SOCIETY MECHANICAL ENGINEERS

The committee having the matter in charge were: Dr. Alex. C. Humphreys, *Chairman,* Dr. R. C. Carpenter, Charles Wallace Hunt, J. W. Lieb, Jr., Fred J. Miller.

The Society was represented by Honorary Vice-Presidents on the following occasions:

Commencement Exercises of Columbia University, Jesse M. Smith; Inauguration of Richard Cockburn MacLaurin as President of Massachusetts Institute of Technology, Worcester R. Warner and Calvin W. Rice; National Conservation Congress, Seattle, Wash., R. M. Dyer, M. K. Rogers, W. F. Zimmermann; American Mining Congress, Goldfield, Nevada, Dr. J. A. Holmes; International Association for Testing Materials, Chas. B. Dudley; funeral of Edwin Reynolds, E. T. Adams, F. M. Prescott, E. T. Sederholm, W. J. Sando and James Tribe; funeral of F. H. Boyer, G. H. Barrus, F. W. Dean, Gaetano Lanza, G. H. Stoddard, Dr. C. J. H. Woodbury.

The following resignations were accepted during the fiscal year:

W. S. Auchincloss, G. W. Blanchard, C. E. Brown, Chas. J. Carney, R. T. Close, Fred Collins, M. T. Conklin, S. G. Colt, B. J. Dashiell, H. H. Dixon,

W. L. Draper, Saml. W. Dudley, Thomas Farmer, Jr., W. Flint, M. L. Foucard, Alex. Gordon, M. M. Green, E. B. Gutherie, O. V. de Gaigne, E. E. Hanna, W. L. Hedenberg, Jas. Inglis, T. A. Holles, Edmund Kent, C. W. Kettell, C. C King, F. C. Kretschmer, A. G. Linzee, J. W. Loveland, Jas. H. Massie, F. Mackintosh, Alfred Marshall, L. M. Northrup, A. T. Porter, A. S. Pritchard, H. S. Richardson, L. C. Schaeffer, E. L. Ross, L. N. Sullivan, Marshall L. Whitney, R. H. Whitlock.

Membership of the following has lapsed during the fiscal year:

M. L. Abrahams, Chas. B. Bruger, H. M. Coale, H. S. Deck, F. H. Davis, C. M. Einfeldt, Robt. P. Fritch, J. M. Garza Aldape, A. A. Hale, M. J. Hammers, R. R. Harkins, L. E. Harper, B. U. Hills, L. A. Holeman, O. H. Klein, D. H. Lopes, Harry G. Manning, Chas. F. Mantine, E. S. Matthews, W. J. P. Moore, Wm. H. Moulton, A. W. Mellowes, C. W. Marx, F. J. McMahon, E. C. Patterson, F. D. Potter, J. A. Prescott. J. L. Rauch, Fred L. Ray, F. S. Ruttmann, G. T. Simpson, H. W. Stacy, R. L. Shipman, O. P. Sells, Wm. E. Toelle, W. O. Teague, Geo. B. Wilson, H. W. Woodward, Chas. H. Young.

The membership has increased during the fiscal year as here indicated:

GRADE	OCT. 1, 1908	LOSSES				ADDITIONS		NET INCREASE	OCT. 1, 1909
		Transfer	Resignation	Lapsed	Death	Transfer	Election		
Honorary	15	......	....	....	1	...	1	......	15
Members	2357	......	18	10	23	35	142	126	2483
Associates	366	11	5	4	4	11	42	29	395
Juniors	786	35	10	11	4	...	108	48	834
Total	3524	46	33	25	32	46	293	203	3727
Affiliates of Gas Power Section								50	150
Affiliates of Student Branches								194	194

The losses by death reported during the fiscal year number the following:

Honorary Member: Gustav Canet; Members: W. M. Allen, W. H. Bailey, F. H. Boyer, A. J. Caldwell, K. Chickering, D. H. Gildersleeve, H. F. Glenn, Thomas Gray, C. L. Hildreth, W. E. Hill, Robert Hoe, W. S. Huyette, E. H. Jones, J. Landsing, R. B. Lincoln, Alex. Miller, A. W. K. Pierce, F. A. C. Perrine, W. T. Reed, E. Reynolds, R. H. Soule, Geo. W. West, A. R. Wolff; Associates: Thomas H. Briggs, Geo. W. Corbin, G. Eberhardt, E. L. Jennings; Juniors: Albert K. Ashworth, Archibald W. Blair, J. R. Rand, A. E. Wellbaum.

The membership has doubled in the last 11 years. The number of applications favorably reported during the year 1909 was 290 for admission, 45 for transfer.

With the number of men eminent in the profession this is a relatively small increase and on account of the benefits which accrue to membership and the importance of extending the Society's influence the members might very properly bring to the attention of engineers of attainment the desirability of securing membership in the Society.

An amendment to C 45 of the Constitution, involving the appointment of a Standing Committee on Public Relations, to investigate, consider and report on methods whereby the Society may more directly coöperate with the public on engineering matters, and on the general policy which should control such coöperation, was proposed at the Spring Meeting and has been approved.

The Committee on Revision and Extension of the Code for Testing Gas Power Machinery, Chas. E. Lucke, *Chairman*, E. T. Adams, George H. Barrus, D. S. Jacobus and Arthur West, was discharged, and a Committee on Power Tests was appointed by the President, consisting of D. S. Jacobus, *Chairman*, Edward T. Adams, Geo. H. Barrus, L. P. Breckenridge, William Kent, Chas. E. Lucke, Edw. F. Miller, Arthur West and Albert C. Wood. The purpose of this committee is to revise the present testing codes of the Society relating to boilers, pumping engines, locomotives, steam engines in general, internal-combustion engines, and apparatus and fuel therefor; to extend these codes so as to apply to such power-generating apparatus, as is not at present covered, including water-power apparatus, and to bring them into harmony with each other and with the best practice of the day. The committee is empowered to confer with other engineering bodies for the purpose of coöperation.

The Committee on Boiler Code, consisting of J. W. Lieb, Jr. and Fred. W. Taylor, reported a revision of the Standard Code for Boiler Tests as desirable in view of the progress made in the art since the formulation of the code.

A Committee on Standards for Involute Gears, consisting of Wilfred Lewis, *Chairman*, Hugo Bilgram, E. R. Fellows, Chas. R. Gabriel and Gaetano Lanza, was appointed to formulate standards for involute gears and report to the Council.

The following were appointed members of the Research Committee: W. F. M. Goss, *Chairman*, James Christie, R. C. Carpenter, R. H. Rice, Chas. B. Dudley.

The report of George H. Barrus, P. W. Gates and W. F. M. Goss, members of the Government Advisory Board on Fuels and Structural Materials, U. S. Geological Survey, was received and placed on file.

Worcester R. Warner, *Chairman*, Walter M. McFarland, Morgan

Brooks, David Townsend and Francis W. Dean were appointed a Nominating Committee.

The request of a number of members of the Society for the organization of a Machine Shop Section was received and referred for action to the Meetings Committee, with the suggestion that a sub-committee to treat the subject be formed rather than a section of the Society.

The invitation extended to the Society by the Institution of Mechanical Engineers of Great Britain, for a joint meeting in England in 1910 has been accepted and a large number of members have already signified their intention of attending. Arrangements will probably be made for the transportation of the party in a single steamer.

The courtesies of the library and rooms of the Society were extended by the President and Secretary to the Japanese Honorary Commercial Commission and the professional members attended a meeting of the Society.

At a gathering of representatives of the four national engineering societies on April 13, the John Fritz Medal was awarded to Charles T. Porter, Honorary Member of the Society, for his development of the high-speed steam engine.

The Society also took a prominent part in the bringing together in a joint meeting of the four national engineering societies for the discussion of our national resources. This was the first meeting of its kind and it is to be hoped that many other occasions will be offered for coöperation.

FINANCES

The Finance Committee has carefully guided the affairs of the Society so that notwithstanding increased activities the excess of income over expense is $4232.79. Of this amount $3010.77 represents 10 per cent of the reserve fund which for some considerable time in accordance with a resolution of the Council has been transferred annually from the reserve to the income account.

It is a source of satisfaction to report that the Society is now so strong that this transfer will be discontinued.

REPORTS OF STANDING COMMITTEES

REPORT OF THE FINANCE COMMITTEE

The Committee submits the statements of the financial condition of the Society, together with the report of Peirce, Struss & Co., of New York, certified public accountants, who have audited the books and accounts.

PEIRCE, STRUSS & CO.
CERTIFIED PUBLIC ACCOUNTANTS
37 Wall Street, New York

November 8, 1909

MR. ARTHUR M. WAITT,
CHAIRMAN FINANCE COMMITTEE

Dear Sir:

In accordance with your instructions, we have audited the books and accounts of The American Society of Mechanical Engineers for the year ended September 30, 1909.

The results of this examination are presented in three exhibits, attached hereto, as follows:

Exhibit A Balance Sheet, September 30, 1909.

Exhibit B Income and Expenses; based on Cash receipts for year ended September 30, 1909.

Exhibit C Receipts and Disbursements for year ended September 30, 1909.

We beg to present, attached hereto, our certificate to the aforesaid exhibits.

Respectfully submitted,
PEIRCE, STRUSS & CO.
Certified Public Accountants

PEIRCE, STRUSS & CO.
CERTIFIED PUBLIC ACCOUNTANTS
37 Wall Street, New York

November 8, 1909

MR. ARTHUR M. WAITT,
CHAIRMAN FINANCE COMMITTEE

Dear Sir:

Having audited the books and accounts of The American Society of Mechanical Engineers for the year ended September 30, 1909, we hereby certify that the accompanying Balance Sheet is a true exhibit of its financial condition as of September 30, 1909, and that the attached statements of Income and Expenses, and Cash Receipts and Disbursements, are correct.

PEIRCE, STRUSS & CO.
Certified Public Accountants

EXHIBIT A

Balance Sheet, September 30, 1909

ASSETS

Equity in Societies Building (25 to 33 West 39th Street)	$353 346.62	
Equity, one-third cost of land (25 to 33 West 39th Street)	180 000.00	
		$533 34
Library Books	$13 700.60	
Furniture and Fixtures	2 966.96	
		16 66
New York City 3½ % Bonds 1954, Par, $35,000	$30 925.00	
Cash in Bank representing Trust Funds	12 918.39	
		43 84
Stores including plates and finished publications		11 60
Cash in Bank for general purposes	$7 444.83	
Petty Cash on hand	250.00	
		7 69
Accounts Receivable		
Membership dues	$4 924.50	
Initiation fees	285.00	
Sale of publications, advertising, etc.	4 334.55	
		9 54
Advances account of land subscription fund		7 96
Advanced payments		2 21
Total assets		$632 87

LIABILITIES

United Engineering Society (for cost of land)		$81 00
Funds		
Life membership Fund	$35 151.07	
Library Development Fund	4 902.71	
Weeks Legacy Fund	1 957.00	
Land Fund Subscriptions	1 227.88	
Robert H. Thurston Memorial Fund	399.13	
Subscriptions to Annual Meeting	205.60	
		43 84
Current Accounts Payable		11 16
Membership dues paid in advance	$494.50	
Initiation fees paid in advance	50.00	
		54

Initiation fees uncollected	$285.00
Reserve (Initiation fees)	24 596.97
Surplus in property and accounts receivable	471 438.68
Total Liabilities	$632 871.54

EXHIBIT B

INCOME AND EXPENSES BASED ON CASH RECEIPTS FOR YEAR ENDED SEPTEMBER 30, 1909

INCOME

Membership dues, current	$50 273.79	
Membership dues, arrears	2 355.00	
Sales gross receipts	8 847.39	
Advertising receipts	11 997.50	
Interest and Discount	1 234.68	
Reserve Fund, 10%	3 010.77	
		$77 719.18

EXPENSES

Finance Committee Office Administration including Salaries	$19 971.91		
Finance, United Engineering Society Assessments	6 000.00		
Finance, miscellaneous	983.56		
		$26 955.47	
Membership Committee		2 392.36	
Increase of Membership Committee		147.94	
House Committee[1]		1 192.43	
Library Committee		2 699.17	
Meetings Committee			
Annual Meeting	$2 074.24		
Spring Meeting	1 410.52		
Monthly Meetings	2 278.19	5 762.95	
Publication Committee			
Advertising Section The Journal	$7 026.06		
The Journal, except Advertising	13 134.80		
Pocket List	1 599.59		
Revises	523.93		
Transactions, Vol. 30	6 533.87		
Year Book	1 401.30		
History	43.65		
		30 263.20	

[1] From Current Income $1192.43
Reserve Fund 2500.00
Total Expenses 3692.43

Research Committee............................	$0.58	
Committee on Power Test.........................	11.25	
Sales Expenditures..............................	4 060.99	
		$73
Excess of Income over Expenses..............................		4
		$77

EXHIBIT C

Receipts and Disbursements for Year Ended September 30, 1

RECEIPTS

Membership dues..............................	$50 832.70	
Initiation fees.................................	6 460.00	
Membership dues and initiation fees paid in advance..	551.00	
Sales of publications, badges, advertising, etc........	20 833.25	
Subscriptions to Land Fund......................	3 251.00	
Subscriptions to Expense of Annual Meeting........	2 188.00	
Interest..	2 072.24	
John Fritz Medal...............................	123.74	
Cash Exchanges per contra......................	575.92	
	$86 887.85	
Cash in Banks and on hand, September 30, 1908	13 708.98	
		$100

DISBURSEMENTS

Disbursements for general purposes................	$76 167.69	
Interest on Mortgage on land.....................	3 240.00	
Cash Exchanges per Contra.......................	575.92	
	$79 983.61	
Cash in Banks and on hand, September 30, 1909.....	20 613.22	
		$100

The Committee also submits as called for by the By-Laws a d estimate of the probable income and expenditure of the Soci the Fiscal year ensuing. This estimate has been submitted careful consideration of each committee concerned and the F Committee has been assured in each instance that the app tions asked for in the estimate include all needed expenditures t out the work of the different committees as now planned and a ized.

It will be noted that the Society is not being operated for but that practically all of the money received is appropriated development of the Society's various interests, and to enable gi each member a constantly increasing return for his membershi

ESTIMATE, 1909–1910

Current Income		Current Expenses	
Dues, Current	$52000	Finance Committee	$26000
Dues, Arrears	2000	Membership Committee	2400
Reserve Fund, 10 %	2800	Increase Memb. Committee	500
Sales gross receipts	5000	House Committee[1]	1150
Interest	800	Library Committee	2880
Advertising	21000	Meetings Committee	8050
		Publication Committee	34900
	$83600	Research Committee	500
		Executive Committee[2]	600
		Power Tests Committee	500
		Sales Expenditures	3000
			$80480
		Excess of income over expense	3120
			$83600

[1] In addition $3000, to be appropriated from the Reserve Fund for the House Committee for betterments for 1909-1910.

[2] The appropriation for the Executive Committee for the foreign meeting to be $3000, to be divided from Current Income at not less than $600 per year for a term of years until cancelled.

Especial attention of the Council is called to the fact that in connection with entering upon our occupancy of the present refined and dignified headquarters, a large sum was advanced from the Society's working capital, known as the Reserve to the Land and Building Fund from which fund by vote of the Council the interest on the mortgage for the land is paid. Admittedly the Society cannot afford to pay for the present headquarters out of its current income unless the Society is freed from debt; and it was with the understanding that sooner or later this debt would be raised, that the Society was justified and enabled to accept the gift from Mr. Carnegie. During the past year the total receipts to the Land and Building Fund have been practically used up for paying the interest on the mortgage, without decreasing the total of the mortgage to the extent of one dollar.

The Finance Committee observes that it has been the custom, by ruling of the Council, to take 10 per cent of the Reserve Fund each year to be applied to the payment of current expenses; and they recommend to the Council that this custom be discontinued, and that the total payments into the Society of initiation fees, which go to make up the Reserve Fund, shall remain in the Reserve, and that only by special vote of the Council shall money be expended from this Reserve.

The Finance Committee trusts that the time is opportune for the Land and Building Fund Committee to take steps during the coming year to raise a portion if not all of the indebtedness amounting to about $90,000.

It is highly desirable in view of plans for broadening the work of the Society that our income available for such extension of work be increased. The organization of our Society is such that the Finance Committee is charged solely with the responsibility of administering the Financial affairs of the Society as they find them and not to produce revenue. All the remaining activities of the Society are for the expenditure of revenue. The Finance Committee suggests therefore that it would be in keeping with good management if a special committee was appointed to consider the essential feature of the Society's broader life, viz: the income side, and to see that it is increased to provide for the reduction caused by the discontinuance of taking 10 per cent annually from the Reserve for operating expenses and to provide for a broader work in the future.

Respectfully submitted

ARTHUR M. WAITT, *Chairman*
EDWARD F. SCHNUCK
GEORGE J. ROBERTS
ROBERT M. DIXON
WALDO H. MARSHALL
Finance Committee

REPORT OF THE HOUSE COMMITTEE

The House Committee reports that it has endeavored to make the headquarters of the Society more attractive, by a rearrangement of the rooms and by additions to the furnishings.

When the Society entered its new headquarters nearly three years ago, provisional furnishings were purchased sufficient to carry on the business of the Society but with no attempt at decorative features.

The original plans of the rooms provided for a large reception hall which visitors enter from the elevators. In common with the other floors of the building this hall was open to the main stairway.

A partition cutting off this stairway and another partition separating the offices has converted this hall into an excellent reception room.

Sliding doors have been arranged so that the Council Room, the Library and the Secretary's office give the effect of one large and spacious room.

The walls have been retinted, and new rugs cover the floors. Comfortable furniture has been placed in the reception room. There will be portieres between the rooms, draperies at the windows, and more comfortable chairs and divans added to the library and Council chamber.

The Committee has aimed to make the rooms homelike and comfortable, to make a place which the members will use freely for their own convenience and for meeting other members or friends for social or business engagements.

In addition to the large rooms referred to, a small room is especially reserved where members may attend to their correspondence or hold private conferences.

Photographs of the Past-presidents have been placed on the walls of the Library and by order of the Council a similar portrait of each succeeding President will be added as he retires from office. Name-plates have been placed on the portraits, paintings and historical objects, and a very complete catalogue of all these objects of historical interest has been prepared. This catalogue represents the result of long and painstaking research on the part of Mr. Edward Van Winkle of our Committee.

Respectfully submitted,

HENRY S. LOUD, *Chairman*
W. C. DICKERMAN
B. V. SWENSON
FRANCIS BLOSSOM
EDWARD VAN WINKLE

House Committee

REPORT OF THE LIBRARY COMMITTEE

During the past year further steps have been taken in the evolutionary process of administering the libraries of the American Institute of Mining Engineers, the American Institute of Electrical Engineers and that of our own Society, as far as practicable, as a unit.

This process has involved the development of a comprehensive plan whereby the library of each society maintains only books on subjects in which its membership is particularly interested, treating all other publications in its library as duplicates. To the American Institute of Mining Engineers have been assigned the subjects of mining engineering, geology, mineralogy, chemistry, metallurgy and a part of chemical technology. To the American Institute of Electrical

Engineers the subjects of electrical engineering, electricity, physics, mathematics and pure science; and to this Society the subjects of general engineering, railroad engineering, mechanical engineering, civil engineering and a part of chemical technology. This plan has given satisfaction as a temporary working basis enabling each organization to complete or supplement imperfect sets from the collections of the others.

During the year a union card catalogue has been established, covering the libraries of the three Founder Societies, which enables readers to find at a glance all the literature on any subject that may be contained in any of the libraries.

A Library Conference Committee, consisting of the Chairmen of the Library Committees of the three societies, has under consideration a further important step toward the unification of the three libraries, involving the organization of the library of the United Engineering Society, to which the three societies shall bear the same relation as do the Founder Societies in the holding of the United Engineering Societies building and property. Such a plan will enable gifts of books or periodicals not specifically designated for one society to be received and taken care of and it may eventually result in the purchase of books jointly in which the three Societies would have a common ownership. This plan avoids purchases in triplicate or duplicate and concentrates the purchasing power and extension of the library in a way that will be of undoubted advantage to all who may have occasion to consult a comprehensive library of engineering literature, covering all branches of the profession and having available promptly after publication all the important books.

It is probable that these improvements will necessitate the carrying out of the original building plans for the library, providing additional shelving in the library room proper, so that all of the volumes may be readily accessible.

The present status of the Library of The American Society of Mechanical Engineers is as follows:

The following titles have been catalogued to date:

Durfee library	570	vol.
A. S. M. E. library	7237	"
Withdrawal of duplicates (not accessioned)	800	"
Pamphlets	1339	"
Total	9946	"

The additions provided for 1908–1909 and included in the above are as follows:

By gift	168	vol.
By purchase	95	"
By binding of exchanges	197	"
Total accessions	460	"

Respectfully submitted,

J. W. Lieb, Jr., *Chairman*
C. L. Clarke
H. H. Suplee
Ambrose Swasey
Leonard Waldo
Library Committee

Report of the Meetings Committee

To facilitate the work of the present Committee, and it is hoped, of succeeding committees, a record has been made of its policies and decisions, some of the more important of which are given below:

The policy of the Committee shall be:

1 Further condensation of papers by the elimination of all superfluous and irrelevant matter, or matter previously printed, and of such statements of fact as are of common knowledge in the profession.

2 The solicitation and selection of such papers, together with the plan of their presentation at meetings, as may make the Transactions a historical and up-to-date record of the progress of all branches of mechanical engineering.

3 The presentation of a subject, whenever possible, in such a way as best to permit of a general and thorough discussion; and to this end to extend invitations to those, whether members or otherwise, whose experience has been such as to bring out the most valuable discussion of the subject.

4 At the Annual and Semi-Annual Meetings, a reduction, when possible, of the number of professional sessions, and of the number of papers assigned thereto in order that more opportunity may be given for satisfactory discussion and for social intercourse between the members. It is the opinion of the Committee that the professional sessions heretofore have been too crowded.

5 For the sake of uniformity, the adoption of a few rules for the guidance of authors, these to be based on the experience of the Committee and of the editorial department of the Society, and to offer a review of the rules governing similar organizations.

6 The adoption of rules tending towards greater uniformity in the actions of the Commitee; these rules to be such only as concern actions within the jurisdiction of the Committee and subject to such exceptions as in the opinion of the Committee may seem desirable.

During the past year, the Committee has submitted to the Council a number of suggestions relative to changes in some of the methods of conducting such affairs of the Society as relate to the Meetings Committee. All of these, with slight modifications, have been accepted and endorsed by the Council and so far as possible placed in operation.

The selection of a local committee to take charge of all entertainment, apart from the professional sessions, was tried at the last Annual Meeting with satisfactory results, which we believe long-established practice will make even better. This is creating greater interest among the local members, and a feeling of some responsibility for the entertainment of the visiting members, and places the Annual Meeting upon the same basis as the Spring Meeting, thereby eliminating what has been heretofore a somewhat inconsistent situation. The Social and Entertainment Committee will for the first time this year collect and disburse the fund for this purpose, which will be kept separate and apart from the funds of the Society. This phase of the arrangement cannot be otherwise than satisfactory.

The resolution of the Committee submitted to the Council, relative to meetings in mid-season in cities other than New York, was put into operation immediately upon approval by the Council. In the opinion of the Committee, this movement is progressing very satisfactorily and seems to be assuming a natural, healthy growth. Successful meetings were held at Boston, April 16, June 11, October. 20, and November 17; and at St. Louis, April 10, May 15, October 16, and November 13. This movement, as was desired and anticipated, is bringing before the Society much valuable material in the form of papers and especially of discussion that would otherwise be inaccessible to the members. It has resulted in an exchange of papers, which promises to become more extensive in the future.

The Council's amendment to the Committee's resolution, "subject to the approval of the Council," we find from experience to be cumbersome. To facilitate these meetings, the Committee must act promptly upon request from members residing in places other than New York. With the appropriations for these meetings decided upon the Committee urges that the Council modify its instructions to the effect that the Committee may have full authority in compliance with the original resolution submitted by the Committee to the Council.

The Committee's interpretation that B 21 did not include the vouchering of bills covering the expenditures of the appropriations for its work, has been confirmed by the minutes of the Council of a few years ago, when the details of such expenditures were placed in the Secretary's hands as business manager. The rules governing office procedure have, however, been changed to define more clearly this interpretation, resulting in some simplification of the work of the accounting department.

Last spring a number of members of the Society requested a meeting or conference on the subject of Smoke Abatement. This petition and the action of the Committee were referred to the Council on May 28, 1909. This request was for a National Conference with the elimination of the engineering features as far as possible. After due consideration the Committee declined to take favorable action.

Subsequent to the above, the Committee received a second petition asking for a National Conference, but along strictly engineering lines. In the absence of precedent relative to such a Conference, the Committee referred the question to the Council. The Committee has not received, but would gladly receive and carefully consider, a paper on the subject of Smoke Abatement, if presented along strictly engineering lines.

We believe the best interests of the Society make necessary a close working arrangement between the Research and Meetings Committees.

A plan was inaugurated early in the year which it is thought will bring before the Society more new material than has been heretofore available. This is accomplished by correspondence with those interested in original research.

The usual number of meetings were held by the Society during the past year, all of which are now on record. The Committee begs to express its appreciation for the assistance and coöperation during the year of the officers and the several departments of the Society.

WILLIS E. HALL, *Chairman*
WILLIAM H. BRYAN
L. R. POMEROY
CHARLES E. LUCKE
H. DEB. PARSONS

Meetings Committee

REPORT OF THE MEMBERSHIP COMMITTEE

During the current year the Membership Committee has held seven meetings, at which a total of 361 applications for membership have been considered with the following results:

Applications void and withdrawn	11
Applications deferred	11
Recommended for membership	33(

There were two ballots during the year on which the applicant recommended by the Committee were voted for. These were at th

Washington meeting	14
New York meeting	18
Total	33

In addition to the most careful consideration which the Secretar and the Membership Committee can give to the applications for mem bership, the coöperation of the whole voting membership is neede in order to maintain the high standing of the Society. In severa instances during the year action by certain members in giving infor mation to the Committee has caused reconsideration of application with the result that they have been indefinitely deferred.

A member should not agree to act as proposer or seconder for a applicant unless he actually knows from his own personal observa tion enough of the latter and his work to be able to answer favorabl all the questions on the reference blank regarding him.

The Committee has endeavored to maintain under the By-Law the standard of qualifications of applicants for whom they have rec ommended to be voted.

The work of the Committee has been greatly facilitated and expe dited by the complete and admirable way in which the cases hav been arranged by the Secretary and his staff for presentation to them

Respectfully submitted,

HENRY D. HIBBARD, *Chairman*
CHARLES R. RICHARDS
FRANCIS H. STILLMAN
GEORGE T. FORAN
HOSEA WEBSTER

Membership Committee

REPORT OF THE PUBLICATION COMMITTEE

The Publication Committee submits herewith the annual report o its work and of the activities under its control for the past year.

The Committee has held frequent meetings and has earnestly en deavored not only to maintain the high standard for the publication of the Society which has previously been set, but also wherever pos sible to raise the standard to a new level. In its work upon Volume 30 of the Transactions which contains the record of the spring and winter meetings of 1908, the Committee has given careful study to the available papers with a view of selecting for that volume only those of greatest value for permanent record. After due consideration several papers have been omitted and others have been edited or revised with the approval of the authors. Discussions also have been edited and in some cases considerably condensed in order to separate material of permanent value from that which had but a temporary or passing interest.

In compliance with the Resolutions passed by the Council in April 1909, the Publication Committee has undertaken the general supervision of The Journal in addition to its other duties, and has adopted the following general plan for the conduct of this work:

As a general policy, The Journal should be regarded as the newspaper of the Society and reports of committees, reports of meetings, professional papers of the Society as a whole or of sections, book reviews, Society items, etc., should be published as requested by committees in their official capacity when approved by this Committee, without charging to the committees or activities concerned any expense for publication. The Journal has its own expense account and the appropriation for The Journal should be sufficient to cover editing and publication of this material.

No papers, whether for the meetings of the Society as a whole, or for sections, technical, student or geographical, are to be published except as formally authorized by the Meetings Committee.

Material from standing committees offered officially will, in general, be published in the form which these committees desire.

Reports of meetings of the Society and of sections, except when containing strictly professional papers and discussions will, in general, be published in condensed form.

All matter presented at meetings other than the professional papers provided by the Meetings Committee, including all discussions, will be edited under the direction of the Publication Committee. As a general policy, discussion will be condensed, commercial matter removed, with a view to presenting only engineering data, opinions based on experience, historical notes and similar material of value for permanent record in Transactions.

The advertising section of The Journal which began with the number of September 1908, has proven successful. The income from this source has increased steadily until at the present time there is a gross annual income from it of $21,000; and through the action of the Council this increased income may be applied to the improving of the quality of, and to the development of The Journal. Plans for such development are under consideration, and it is the purpose of the Committee to make improvements as rapidly as conditions may warrant.

But the most effective work upon The Journal and that which will be of greatest benefit to our membership at large is the careful preparation for publication of the professional material presented at the regular meetings of the Society, and at the meetings of the different sections. In this great fund of material there is always some that is unimportant and irrelevant, and much more that could be made of greater value by skilful editing or by condensation. During the past year the Committee has done much in this direction that has resulted in the improved quality of our paper, and also in a considerable economy of money, and the papers now appearing in The Journal are suitable, with little or no alteration, for publication in the Transactions.

In addition to the volume of the Transactions and The Journal the Committee has issued the annual Year Book of the Society and the Pocket List of Members.

Respectfully submitted,

A. L. WILLISTON, *Chairman*
D. S. JACOBUS
H. F. J. PORTER
H. W. SPANGLER
G. I. ROCKWOOD

Publication Committee

REPORT OF THE RESEARCH COMMITTEE

The Research Committee was formally notified of their appointment under date of April 7, 1909, and at the suggestion of the President, the members were requested to meet during the Spring meeting of the Society at Washington. Notice was given a short time in advance of the meeting, and only Prof. R. C. Carpenter and R. H. Rice were present. These members, however, together with the President of the Society, Jesse M. Smith, and Charles W. Hunt, Past-President, and originator of the suggestion that a Research Committee be appointed, engaged in an informal conference.

A second meeting was called for Wednesday, June 23, 1909, to be held in New York. There were in attendance the President, Jesse

M. Smith, R. H. Rice, James Christie, W. F. M. Goss, and the Secretary, Calvin W. Rice. Dr. Goss was chosen Chairman. The Secretary of the Society was recognized as the secretary of the Committee. The minutes of the informal meeting held in May were read for the information of the members. After a considerable discussion as to the scope of the work of the Committee, it was agreed that the Committee should have information concerning the laboratories of the various colleges, and other public institutions in America, in which work of engineering research is proceeding, and to this end the Secretary was directed to develop a process which would result in the establishment of such a record in the office of the Society.

It was agreed that the Committee should consider the question of safety valve efficiency. Arrangements were made for gathering in existing information upon this general subject, and steps were taken which will, it is believed, result in a satisfactory outline from which actual work may proceed. Several other subjects for research, referred to the Committee by the Council, were laid on the table for future consideration.

Respectfully submitted

W. F. M. Goss, *Chairman*
James Christie
R. C. Carpenter
Richard H. Rice
Charles B. Dudley

Research Committee

No. 1250

THE PROFESSION OF ENGINEERING

PRESIDENTIAL ADDRESS 1909

By Jesse M. Smith, New York

President of the Society

Great engineering works existed in many parts of the world long before Columbus discovered America. We have but to consider the ruins left by the Incas in South America and the Aztecs in Mexico to realize the great work done on this continent in engineering. In Asia the great wall of China, the temples of Japan, China, Babylonia and Assyria bear record of the presence of the engineer.

2 In Africa, the vast pyramids of Egypt and the temples on the Nile are evidences that great engineers existed long before the Christian era. We marvel still when contemplating the pile of immense blocks of stone forming the pyramids and try to imagine what form of apparatus could have been used in placing those great stones one upon the other.

3 In Europe the Greeks and Romans did marvelous work in roads, bridges, aqueducts, and various mechanical structures which the modern engineer may well ponder upon and admire. While we read much in history of the emperors and kings who reigned when these great engineering works were produced, we learn little of the men who produced them, men whom we now call engineers.

4 While engineers have existed for thousands of years it is only within a comparatively recent time that they have begun to form themselves into societies for their mutual education and the advancement of the profession of engineering.

5 In England, as early as 1771, Smeaton and his contemporaries came together to form the Smeatonian Society of Engineers, which, therefore, according to the calculations of a noted English engineer, is five years older than the United States. The Institution of Civil Engineers of Great Britain came into existence in 1818, and was

An address delivered at the Annual Meeting, New York, (December 1909) of The American Society of Mechanical Engineers.

followed by its sister society, the Institution of Mechanical Engineers, in 1847. La Société des Ingénieurs Civils de France was founded in 1848. Der Verein Deutscher Ingenieure was organized in 1856.

6 In this country the Boston Society of Civil Engineers began its work in 1848. Our elder sister among national societies, the American Society of Civil Engineers, was organized in 1852. The next member of the family, the American Institute of Mining Engineers, was born in 1871. Our own Society came into existence in 1880, and our younger and very vigorous sister, the American Institute of Electrical Engineers, came along in 1884.

7 Each of these four national societies, the American Society of Civil Engineers, the American Institute of Mining Engineers, The American Society of Mechanical Engineers and the American Institute of Electrical Engineers, has grown greatly since its organization, and each continues to thrive. During the process of upbuilding of these four great national societies, several other national societies of specialists in engineering and many local societies of engineers have been formed, and all of these are also active and thriving.

8 The four greater national societies have an aggregate membership at this time of over 19,000 members. Twelve national societies of engineering specialists contain more than 13,000 members. Twenty-three local engineering societies in different cities of the United States count over 8,600 in their membership.

9 What does this great army of over 40,000 engineers, organized into many different societies, all for purely professional purposes, mean? It means that the engineering profession is making itself felt in this country of ours, that it proposes to take a prominent place in the great activities by which the country is being developed, that it will take its place in public affairs, that it is coing into its own.

10 The national societies are not antagonistic to each other; on the contrary, they support and give confidence to each other. The national societies of specialists are not at war with the other national societies; they supplement them.

11 The local societies are not in opposition to the national societies; they extend their influence; they are the outposts of the great army. The specialists do not interfere with each other. We are all specialists to a greater or less extent; but we are all *engineers.*

12 In the legal profession, some men practice in the criminal courts; others devote themselves to titles in real estate; others are in corporation law; others in patent causes; they all squabble with each other in their practice; but when they meet in their bar associations they are all lawyers; they stand by each other and their profession; they are a power in the world.

13 The medical profession is made up of surgeons, oculists, aurists, general practitioners, specialists of the skin, the heart, the lungs and every other part of the human anatomy; but when they come together in their general medical associations they are all doctors; they also stand by each other and their profession; they also are a power in the world.

14 In the engineering profession why may not the men who practice in steam engineering, in machine construction, in hydraulics, in railroad, bridge, mining, electrical and chemical engineering, in metallurgy, refrigeration, heating and every other specialty in engineering, come together, stand by each other and their profession, become known as *engineers* and be a power in the world?

15 When, in 1889, the Institution of Civil Engineers of Great Britain invited the four national American societies of civil, mining mechanical and electrical engineers to visit it in London, there was inaugurated a spirit of friendship and coöperation in the engineering profession which has grown stronger and stronger as the years have passed. Following the visit in London, La Société des Ingénieurs Civils de France, in the same year, invited the American societies to Paris.

16 Those who were fortunate enough to participate in those memorable demonstrations of hospitality cannot fail to realize how greatly the seed of coöperation sown in that year has fructified.

17 In 1900 this Society was again invited by the Institution of Civil Engineers and the Institution of Mechanical Engineers to visit them in England, and again invited by the French society to visit it in Paris. Thus the spirit of coöperation was still further advanced by these remarkable meetings. On both occasions the sister societies abroad were untiring in the entertainment of the American engineers.

18 The year 1904 was made memorable by the acceptance of an invitation extended by this Society to the Institution of Mechanical Engineers of Great Britain to hold a joint meeting in Chicago. Thus the spirit of coöperation and good friendship was again strengthened and extended.

19 Now the Institution of Mechanical Engineers of Great Britain has expressed the desire still further to promote this friendly spirit by inviting this Society to a joint meeting in England in July 1910. The Council of our Society has accepted this very cordial invitation of the Institution in the spirit of good will in which it was extended. It remains for the membership of The American Society of Mechani-

cal Engineers to respond to this spirit and to go to England next year with its best talent and its best men.

20 The helpful coöperation in professional work which has already been established with our sister societies over the seas is also becoming manifest in our own country. The four national societies of civil, mining, mechanical and electrical engineers on March 24, 1909, held in this auditorium a joint meeting on the Conservation of the National Resources, which did much to bring engineers close together and into coöperative relation.

21 Our Society invited the Boston Society of Civil Engineers to join in the monthly meetings of the Society recently held in Boston. The Engineers' Club of St. Louis in like manner was asked to join with us in the Society's monthly meetings recently held in St. Louis. In both cases the invitations have been accepted in the best spirit of coöperation.

22 The engineering societies of the country may be likened to the members of a large and harmonious family, each member independent to do its own special work in its own way, each member ready to help each of the others, each residing in its own home, but all ever ready to stand by each other, to work for the common good, to advance and dignify the profession of engineering.

23 A striking example of the "getting together" of the engineering societies is found in this building which is the home of our Society. It is also the home of our sister societies, the American Institute of Mining Engineers and the American Institute of Electrical Engineers.

24 Under the same roof are grouped together fifteen other societies of engineering and allied arts. Twenty-five thousand engineers practicing in all the specialties of engineering may call this building their professional home. We are living together here in peace and harmony. We have brought our books together into a single library open to the profession and to the public, where every one is welcome.

25 Our meetings are held in the same auditorium and lecture halls; the doors stand open that all who wish may enter. Our professional brethren of every society of every country are welcome here. The large hall at the entrance to the building is a foyer where all engineers may come together on the same plane, where they may unite to strengthen each other and to sustain and advance the profession of which they form a part.

26 The spirit of coöperation which now exists must be fostered, strengthened, made enduring, to the end that as great solidarity will

exist in the engineering profession as exists in any of the other great learned professions.

27 Numbers in membership are, of course, important in the societies which represent the engineering profession, but a high standard of membership is of much greater importance.

28 With a considerable number of high-grade technical schools throughout the country all striving with each other to raise the standards of engineering education ever higher and higher; and with the graduates from these institutions taking, from year to year, a larger and more responsible part in the great activities of the country, there is no lack of material from which to form a membership in the engineering societies which will be worthy of the profession.

29 In the Institution of Civil Engineers, as well as in the Institution of Mechanical Engineers of Great Britain, we are informed, no person is admitted into the lower grade of membership unless he can pass a satisfactory examination as to the fundamental principles of engineering, conducted by an examining board of the Institution. The rules laid down by this examining board form the standard by which the applicants to membership are measured. If the technical schools in Great Britain maintain an equally high standard in granting their degrees in engineering, then the degree may be accepted in lieu of an examination. In other words, the engineering institutions in Great Britain establish the standard for the degrees granted by the technical schools. A promotion from a lower to a higher grade of membership is made only upon a showing of sufficient experience in engineering to satisfy the rules laid down by the Institution.

30 In The American Society of Mechanical Engineers, a person may enter the Society as a Junior upon the presentation of a degree in engineering from a technical school. But this Society has not, up to the present, established a standard by which to measure that degree. I believe the standard for such a degree in engineering should be established by the Society, and that it should be as high as that of the best schools of engineering in this country. It will follow that the schools having a lower standard will soon be brought up to the higher standard.

31 Promotion to higher grades of membership in our Society is only made upon a showing of engineering experience satisfactory to our Membership Committee. This committee is maintaining a high standard of membership, and I believe that acting under the influence of the membership and the Council of the Society, it will not allow that standard to fall, but rather cause it to rise.

32 If we are to have a profession of engineering, as distinguished from the trade of engineer, we must have a broad education befitting men of a learned profession, as distinguished from a narrower education sufficient for men of a trade.

33 President Lowell of Harvard in his recent remarkable inaugural address, gave this as his conclusion: "The best type of liberal education in our complex modern world aims at producing men who know a little of everything and *something well.*" If that conclusion be true of a liberal education leading to the learned profession of the law or medicine or theology, why is it not also true of a scientific education leading to the learned profession of engineering?

34 If preponderance be given to one part of President Lowell's conclusion over the other part, certainly knowing "a little of everything" leads to superficiality; while just as surely knowing but one thing well leads to narrowness. There would seem to be a happy mean between these two extremes in the education of the engineer.

35 The engineer capable of being at the head of the larger engineering works must know something of many things, several things well and one thing profoundly.

36 The engineer, president of a great railway system, for example, must know *something* of the alignment and gradients of the permanent way, its construction and maintenance; *something* of the proper location of sidings and stations; *something* of the system of signals, of the various kinds of cars, of the quality of water for the locomotives, of the heating and lighting of cars, and many other things. He must know *well* that the bridges have been designed for safety and endurance and that they have been properly constructed. He must know *well* that the tunnels are safely protected against external pressure and falling rocks. He must know *well* that the locomotives for drawing the high-speed trains, as well as those for the heavy freight trains, are of the very best design and capable of performing their duty with efficiency, economy and endurance. He must know *well* how to manage the traffic and keep the accounts. He must know *profoundly* how to coördinate all the different parts of this complex organization so that each part will perform its proper and full function, to the end that passengers and freight will be carried safely, surely, quickly and cheaply, and also that dividends will be paid to the shareholders.

37 The engineer knowing something of many things, several things well and one thing profoundly, is still one-sided if all this knowledge is confined strictly to his profession. He will be a much

broader man and a better engineer, if in his leisure hours he can turn his thoughts entirely away from his professional work and toward those things in nature and art which give that rest and renewal of the professional mind necessary to continued work.

38 Engineers have known for many years that the profession of engineering is a learned profession; the rest of the world is rapidly arriving at the same conclusion.

39 When in April 1907, this building was dedicated "To the advancement of Engineering Arts and Sciences," President Hadley of Yale, where the learned professions have been taught for nearly 200 years, said:

> The men who did more than anything else to make the nineteenth century different from the other centuries that went before it, were its engineers.
>
> Down to the close of the eighteenth century the thinking of the country was dominated by its theologians, its jurists, and its physicians.
>
> These were by tradition the learned professions, the callings in which profound thought was needed, the occupations where successful men were venerated for their brains.
>
> It was reserved for the nineteenth century to recognize the dominance of abstract thought in a new field—the field of constructive effort—and to revere the trained scientific expert for what he had done in these lines.
>
> Engineering, which a hundred years ago was but a subordinate branch of the military art, has become, in the years which have since elapsed, a dominant factor in the intelligent practice of every art where power is to be applied with economy and intelligence.

It is encouraging to engineers to have their profession recognized as a "learned profession" by so great an authority as the president of Yale University.

40 Enthusiasm and devotion to his profession are characteristic of the engineer, and from my observation these begin with the student in engineering and extend throughout his life. President Wilson of Princeton, in an address at Harvard not long since, dwelt upon "the chasm that has opened between college studies and college life. The instructors believe that the object of the college is study, many students fancy that it is mainly enjoyment, and the confusion of aims breeds irretrievable waste of opportunity." These conditions, I believe, exist to a much smaller extent in the technical schools where engineers are taught, than in the general colleges, where a liberal education is obtained.

41 Enthusiastic love of work, for his profession's sake, resides in the heart of the engineer who becomes great. The man who merely works for wages, and without enthusiasm, does not rise; he remains a paid servant, and poorly paid at that.

42 Where enthusiasm exists, love of work exists; success follows. Our individual enthusiasm is quickened by the study of the work of our brother engineers.

43 What engineer while being whisked through the tunnels which connect Manhattan Island with the lands surrounding it, can fail to rejoice in his profession as he contemplates the work of the civil engineers, the mining engineers, the mechanical engineers, the electrical engineers, which, joined together, supplemented each other to produce success in those marvelous undertakings? The highest knowledge and skill in each of the four branches of the engineering profession were called for, and were forthcoming, in the consumma-tion of this great work. It is not a question which engineers did the most toward the success of this problem in transportation; they all did their best; they all did well; each contributed a necessary part to the success; they were all engineers working for the advancement of the profession of engineering.

44 Will not every true engineer feel his enthusiasm in his profession quicken, as he watches the great vessels of trade and the great vessels of war sweep out to sea, and stops to consider how much brains, and long experience, and hard work of many men are concentrated in each one of them?

45 We marvel still, our enthusiasm is inspired, as we see ponderous steam locomotives and mysterious electric locomotives competing in the hauling of trains, ever heavier and heavier, ever faster and faster, and both succeeding.

46 The automobile in its present highly developed and thoroughly practical form is the result of enthusiastic work of many engineers, principally within the last fifteen years.

47 The enthusiasm of the engineer is never satisfied. Having conquered the highway with the automobile driven by the internal-combustion gas engine, he now proposes to conquer the air with the aëroplane driven by the same kind of an engine in improved form.

48 The American Society of Mechanical Engineers has before it a future of usefulness to its members and influence in the profession, which is unlimited. It only requires that we stand by our tradition of increasing the membership with men of high quality as engineers; that the members maintain enthusiastic devotion to good professional work; that they coöperate with each other in the broadest and most friendly spirit to produce that solidarity of membership and devotion to high ideals, which will compel the world to class the profession of engineering with the other learned professions.

No. 1251

THE HIGH-PRESSURE FIRE-SERVICE PUMPS OF MANHATTAN BOROUGH, CITY OF NEW YORK

DESCRIPTION OF PUMPS AND PUMPING SYSTEM WITH RESULTS OF TESTS

By Prof. R. C. Carpenter, Ithaca, N. Y.
Member of the Society

The object of this paper is to present a concise description of the high-pressure pumping system installed for fire service in the city of New York and the results of a test of the pumping machinery.

2 The system protects the district extending north from City Hall to Twenty-fifth Street, and east, approximately, from the North River to Second Avenue. It comprises about 55 miles of extra heavy cast-iron main, from 12-in. to 24-in. in diameter, with 8-in. hydrant branches; and two pumping stations so located that they never can be in the center of a conflagration. At the present time the pumping stations have a combined capacity of over 30,000 gal. per min. delivered at a pressure exceeding 300 lb. per sq. in.

THE SOURCE OF WATER SUPPLY

3 The supply of water is ordinarily obtained from the water mains of the city, which deliver Croton water to the stations at a pressure of from 14 lb. to 40 lb. per sq. in., depending upon the demand for water in that district. Both of the pumping stations are located close to tidal water and connections are made so that sea water can be obtained in case of difficulty with the Croton supply.

4 The advantage of the Croton water over salt water is that it is less likely to injure goods, and as the amount required for fire purposes is only a small percentage of that consumed for the daily supply of the city its use for fire protection makes no material difference from financial or insurance standpoints. As this is a matter of considerable importance data upon the quantity needed are given in the next paragraph.

Presented at monthly meetings, New York and St. Louis (October 1909), of The American Society of Mechanical Engineers.

WATER REQUIRED FOR FIRE PURPOSES

5 The general impression that an enormous quantity of water is required for fire purposes is erroneous as shown by figures furnished to Chief Engineer I. M. de Varona by the fire department for the Boroughs of Manhattan and Brooklyn, years 1900, 1901, 1902, 1903 and 1904. These give the average quantity of water used for fire protection during these years in the Borough of Manhattan as 74,010,803 gal. per year, of which 31,056,928 gal. was river water. The daily average use of Croton water, therefore, for the above five years was 117,000 gal.

6 For the Borough of Brooklyn the average for five years was 43,705,568 gal. of which 19,010,928 gal. was river water; daily average, 67,000 gal.

7 During these five years the greatest quantity used in the Borough of Manhattan was 99,000,000 gal. in 1901, which included 69,500,000 gal. of river water, leaving 29,500,000 gal. for Croton water, and Mr. de Varona states (Report of the Department of Water Supply, Gas and Electricity): "Even if this quantity be made 100,000,000 gal. per year, by comparing it with the average daily consumption of about 300,000,000 gal. it will be seen that the total amount used for fire purposes would be only about one-third of the amount used for all purposes in 24 hr., forming, therefore, an insignificant percentage of the total consumption. The quantity needed for fire purposes (one-tenth of one per cent) may therefore be entirely neglected as a factor in determining the water supply of the city.

8 "The capacity of each of the pumping stations will be for the present 15,000 gal. per min. or 43,000,000 gal. per day for the two stations. By the installation of three additional units in each station, for which provision is made, this capacity can be increased in round numbers to 69,000,000 gal. per day.

9 "The two stations, with the motors and pumps as installed, have a total capacity in excess of that of all the fire engines in the Boroughs of Manhattan, the Bronx and Brooklyn working under normal conditions. This comparison assumes the engines to work on one line of 2½-in. hose, say 500 ft. long, under a pressure of, say 200 lb., and with the capacities as printed in the official blank forms of the reports of the fire department. It should furthermore be remembered that provision is made for the installation of still another pumping station."

MOTIVE POWER

10 The power for driving the pumps is transmitted electrically from several of the electric power and lighting systems located on Manhattan Island. As the stations of systems are widely separated and any or all of them are available for motive power the system of electric transmission was considered more reliable in the case of a large and general conflagration than power plants maintained directly at the pumping stations. Each station is provided with two independent sets of transmission lines located as far as possible beyond danger or injury in case of a great conflagration.

11 The cost of erecting and maintaining an independent power plant would have entailed a greater annual charge than the cost of the electric current; consequently the present arrangement is advantageous from a financial standpoint.

12 In addition to the charge per kilowatt for the current delivered there is a charge aggregating $90,000 per year for reserving the first right of use for the necessary generating machinery for this purpose. The total cost of maintenance of the system is estimated at $170,000 a year, which amount it is believed will be saved many times over by a reduction in insurance premiums now paid in the protected district.

13 The electric current is supplied at a pressure of 6600 volts from the following stations of the New York Edison Company, having the capacity indicated: 53 Duane Street, 7600 kw.; 115 East 12th Street, 1700 kw.; 45 West 26th Street, 400 kw.; 140th Street and Ryder Ave., 4000 kw.; Waterside Stations No. 1 and No. 2, 196,700 kw. In addition there are feeders extending to the Brooklyn Edison Company stations which can be called on in case of an emergency demand.

14 The pumping stations are connected to 18 sub-stations, equipped with rotary converters and storage batteries, aggregating a capacity of 124,000 ampere hours at 135 volts, an enormous reserve.

15 Each station is connected with the main stations of the Edison Company by two 250,000 c.m. three-phase cables laid in ducts, and two independent reserve feeders extend to the sub-station system of the Edison Company. With all these precautions, interruption of the power supply would seem a physical impossibility.

THE DISTRIBUTION SYSTEM

16 The following information upon the distribution system is taken largely from the department report of Chief Engineer de Varona

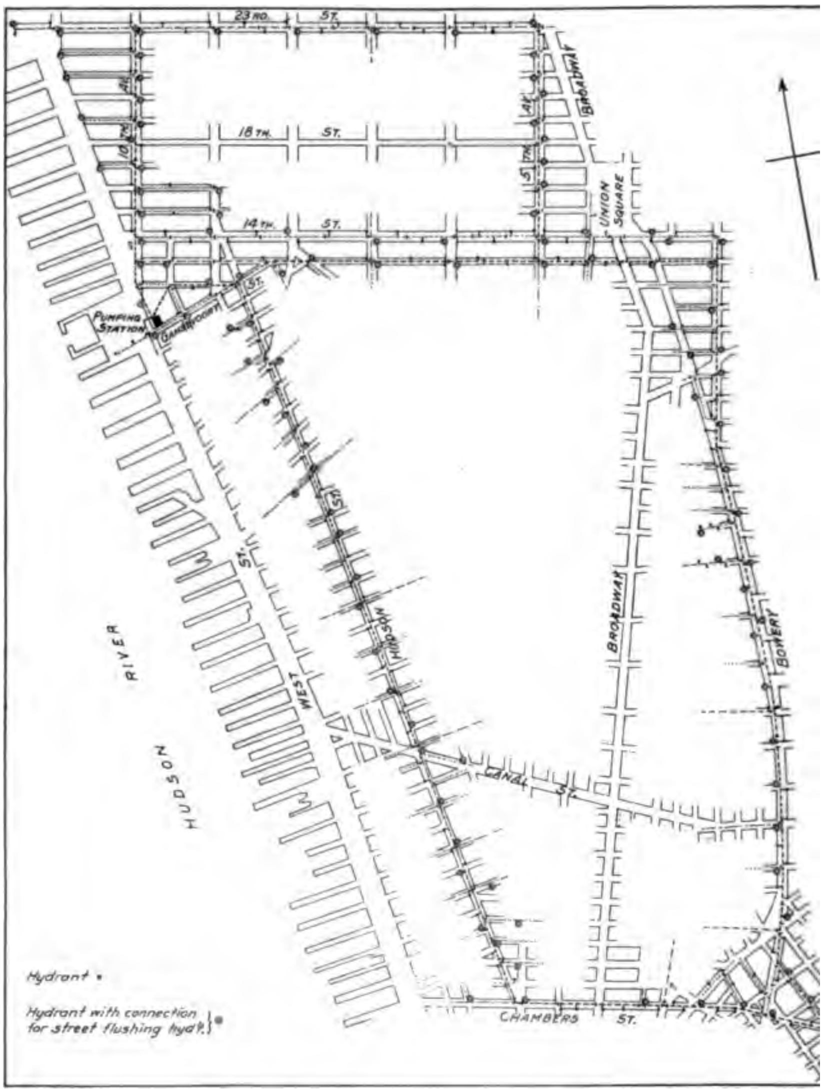

FIG. 1 SHOWING LOCATION OF STATIONS AND AREAS COVERED BY HIGH-PRESSURE PUMPING SYSTEM

THE AREA INDICATED IS SERVED BY A SYSTEM OF MAINS RANGING FROM 24 IN. TO 12 IN. I[N] DIAMETER WITH 8-IN. HYDRANT CONNECTIONS for 1905. Fig. 1 shows the system to be bounded by mains laid on the north through Twenty-third Street; on the east, through Broadway to Fourteenth Street, through Fourteenth Street to Third Avenue, down Third Avenue to the Bowery, down the Bowery to Chambers Street; through Chambers Street on the south to West Street; and on the west through West Street.

17 The area actually protected is considerably greater than this as hose can be extended over a zone 600 ft. wide beyond the limits of the mains.

18 This district was selected as that in which the fire losses were the greatest and which most urgently needed fire protection. Plans have been prepared for the extension of the system southerly to the Battery, easterly as far as the East River, and, if necessary, northerly as far as Fifty-ninth Street, by the simple extension of the mains and probably the erection of a third pumping station.

19 The pipes, castings and hydrants were tested at a pressure of 450 lb. The specified allowance for leakage in a 10-min. test was at the rate of 4 gal. in 24 hr. for each lineal foot of pipe joint, equivalent to a leakage of 487,000 gal. for the whole system in 24 hr., which is somewhat over one per cent of the total specified pumping capacity now installed. The actual leakage on test was at the rate of 264,000 gal. per day or about six-tenths of one per cent of the pumping capacity. Considering the difficulties of construction and the high pressure, the results attained were remarkable and reflect great credit on the engineer in charge.

20 There are sufficient hydrants so that if a block were on fire 60 streams of 500 gal. per min. each, or the full capacity of both stations, could be concentrated on a block with a length of hose not exceeding 400 ft. to 500 ft., assuming the use of 3-in. hose and 1¼-in. nozzles.

21 The layout of the mains at the stations both for suction and delivery is on the loop system; that is, the supply can be taken from either one of two mains, and the discharge is through either one or both of two mains. With this system even the breakdown of one of the discharge mains at the station would only slightly reduce the pressure at the fire and would not affect the capacity of the station, as the pumps would be capable of forcing their full discharge through the short length of a single 24-in. main if made necessary by such an accident.

22 The mains are of cast-iron, bell and spigot pipe, of the thicknesses given in the following table:

Size of Pipe Inches	Thickness Inches	Unit Tensile Strain with 300 lb. pressure	Factor of Safety
24	1⅞	1920	0.4
20	1½	2000	10.0
16	1¼	1920	10.4
12	1	1800	11.1
8*	⅞	1371	14.6

* Used only for hydrant branches.

SUPPLY PIPING

23 At the South Street Station the fresh water supply is derived from two 30-in. lines, one connected at Chestnut Street to the 36-in. line on Madison Street, and the other connected at Pike Street to the 36-in. line on Division Street. These two main feeders, to which the two 30-in. lines are connected, increase to 48 in. in diameter and extend independently and directly to the Central Park Reservoir and are also reinforced by connections with the main feeders in this section of the city.

24 An auxiliary salt-water supply, consisting of two 36-in. pipes about 140 ft. long, brings the salt water from the East River to a suction chamber located directly in front of the pumping station. This suction is so constructed that the pipes are always below mean low water, thus insuring a supply at all times and avoiding the possibility of a break in the suction caused by air getting into the suction lines. On the river end of this suction there are constructed heavy bulkhead screens and in the suction chamber are two sets of bronze screens which are readily accessible for cleaning. From the suction chamber there are taken two 30-in. flanged mains to the duplicate set of mains in the pumping station proper. The vacuum in these 30-in. pipes is maintained by automatic electric vacuum pumps located on the pump room floor of the station.

25 At the Gansevoort Street Station the fresh-water supply is derived from two 30-in. mains, one connected at Twelfth Street to the 48-in. line on Fifth Avenue, which runs direct to Central Park Reservoir, and the other connected to the 36-in. line on Ninth Avenue at Little West 12th Street, which increases to a 48-in. line and runs also direct to the Central Park Reservoir. These two main feeders, in addition to having their supplies direct from Central Park Reservoir, are also reinforced by connections with the main feeders in this section of the city.

26 The salt-water suction lines for this station are practically identical with those for the South Street Station except that the 36-in. lines from the North River to the station are 650 ft. long.

PUMPING STATIONS

27 The two stations, known as the Gansevoort pumping station, located near Gansevoort Market on the North River, and the South Street station, located on the corner of Oliver and South Streets near the East River, are identical in construction and equipment. The buildings are of simple design, of steel fire-proof construction, with concrete foundations. The Gansevoort Street building, which is typical of both, is one story high with basement, 63 ft. 8 in. by 97 ft. 4 in. Each station is large enough for eight pumping units.

MACHINERY

28 There are now five units in each station consisting of Allis-Chalmers five-stage centrifugal pumps driven by Allis-Chalmers induction motors and the necessary auxiliary machinery. The motors and pumps are alike and their parts are interchangeable.

29 The pumps each have a specified capacity of 3000 gal. per min. of sea-water, working with a suction lift of 20 ft. and a delivery pressure of 300 lb. per sq. in. The actual capacity as indicated by a 24-hr. test was about 30 per cent in excess of that specified. The original specifications contemplated the use of six-stage pumps, with the expectation that sea-water would be used at each fire. Because of the facts already referred to (Par. 4), that the relative amount of water required for fire purposes is insignificant and that sea-water may do considerably more damage to goods than fresh water, a change in the specifications was agreed to, whereby the pumps should work at best efficiency when receiving water from the Croton mains at a pressure on the intake side varying from 15 lb. to 40 lb. per sq. in.

30 To meet this new condition the pumps were all built with five stages. All the sea connections and priming machinery as originally contemplated were installed, so that sea-water can be pumped into the mains whenever desired. The effect of the change is merely to reduce the pressure head slightly in case sea-water is used.

ARRANGEMENT OF MACHINERY

31 The floor-plans of the buildings and general layout of machinery, piping, switchboards, etc., are shown in Fig 2. As will be seen

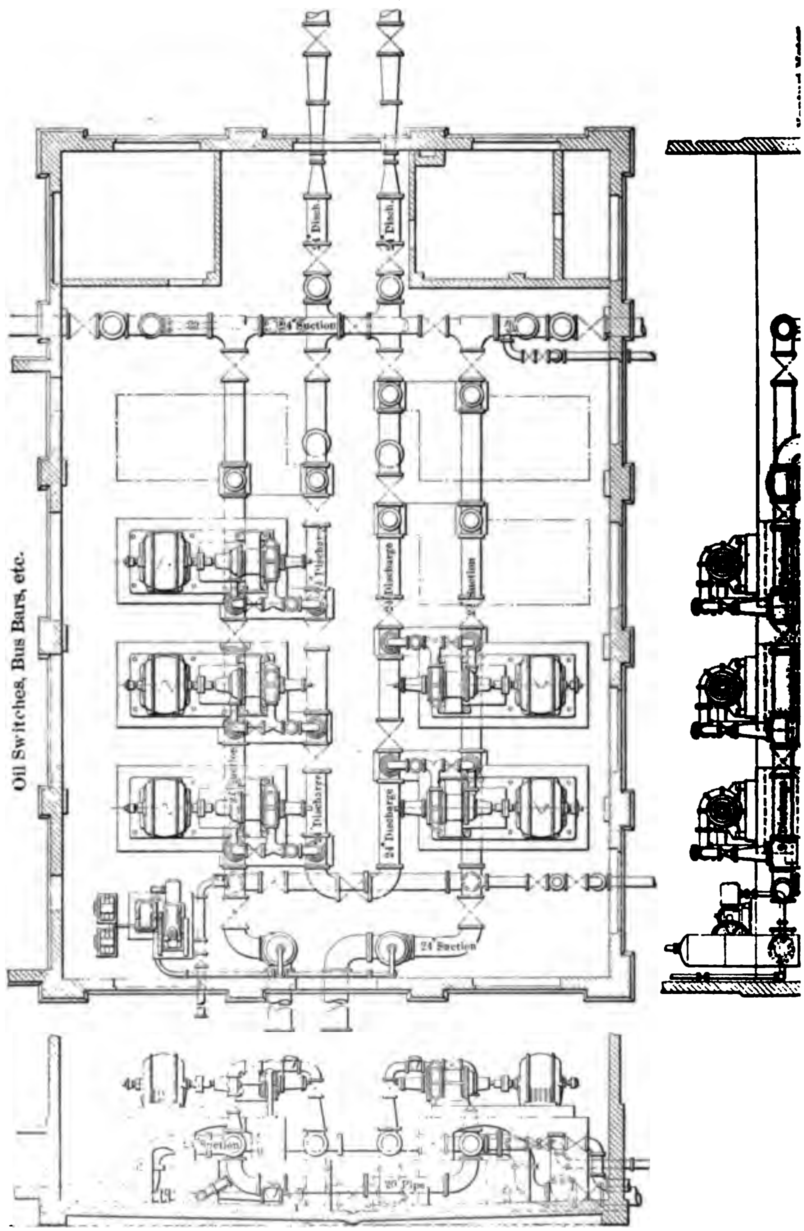

Fig. 2 Plan and Elevation Showing Arrangement of Hydraulic a Electrical Apparatus in Pumping Stations

ce is provided for three additional units. Working detail plans of machinery were furnished by the contractor. The arrangement wn in Fig. 2 is the same for both stations, the only difference being t the switchboard and office in the South Street station are on erent sides of the building as compared with the Gansevoort et station.

2 The motors and pumps, with suction and delivery branches, located on the main floor of the pump room. The switchboard switchboard apparatus are placed in an enclosed two-story and ment gallery.

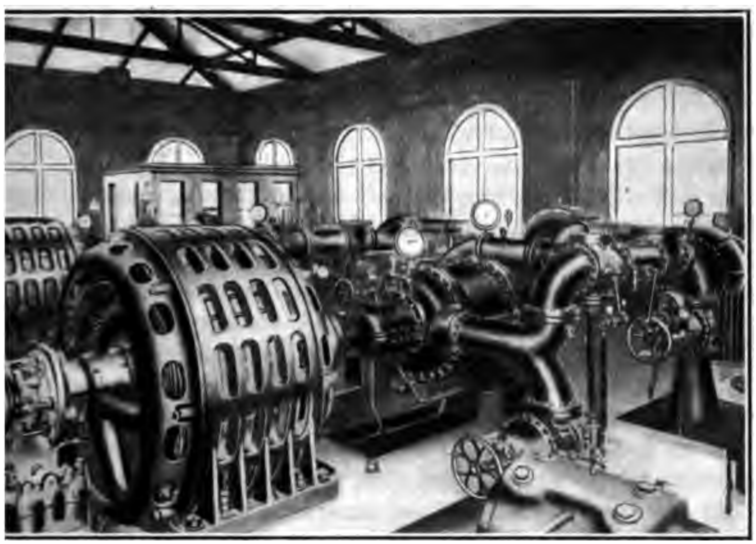

Fig. 3 Interior View of Station

3 The four high-tension feeders and all other wires entering the lding are brought in through the gallery basement. All terminal rk on the entering wires is located in the basement. On the first or of the gallery, which is approximately on the same level as the mp-room floor, are placed the oil switches, with their controlling d protective devices, fire-proof cells and compartments.

34 The operating switchboard is conveniently located in the closing wall of the gallery, and is so placed as to allow a man nding on the pump-room floor to perform all the operations neces-ry for controlling the apparatus in the station. The bus bars,

with their fireproof compartments, are placed on the second floor of the gallery.

MOTORS FOR CENTRIFUGAL PUMPS

35 The motors are of the constant-speed, wound-rotor induction type, 3-phase, 25-cycle, 6300-volt to 6600-volt, designed to operate at about 740 r.p.m. Each pump is direct-connected to its motor by a flexible coupling which takes care of any variation from alignment. In starting, an iron grid resistance is connected in the secondary circuit and gradually cut out by means of a handwheel on the motor switchboard panel. When the resistance is all cut out the rotor is automatically short-circuited and operated by specially constructed solenoids through a small switch mounted directly on the shaft of the handwheel above referred to. An interlocking arrangement prevents the operator from closing the switch connecting the motor to the line while the motor is short-circuited.

36 The specifications required the motors to have sufficient starting torque to attain full speed between 30 sec. and 45 sec. after starting, with a current not exceeding 150 per cent of that used when the motor is working under full speed. Each motor was required to develop not less than 800 b.h.p. when using current of 6300 volts, 25 cycles, and under these conditions to have an efficiency not less than 92 per cent, a power factor not less than 93 per cent, and a motor slip not in excess of 2 per cent. At three-quarters load the efficiency was not to be less than 92 per cent and the slip not to exceed 1.5 per cent. It was specified that the temperature of the motors should not rise more than 40 per cent on a 24-hr. test at full load, when measured by a thermometer, the air in the room being 25 deg. cent.

37 Prof. Geo. F. Sever of Columbia University tested two of the motors in the shops of the contractor and found them to meet the specifications and to have a full-load efficiency of 93.2 per cent. The other motors were inspected and found to be alike and were assumed to have the same efficiency. The motors were also tested for temperature rise at the time of the official test to be described later.

MOTORS FOR AUXILIARIES

38 Direct-current motors of 240 volts are provided to operate the various gate valves in the station and the piston pumps employed for maintaining a vacuum on the salt-water suction lines.

PUMPS

39 As previously stated the pumps were finally constructed with five stages, each to give a pressure of somewhat over 60 lb. per sq. in., making the combined pressure of the five stages about 300 lb. per sq. in. above the intake pressure, which is the maximum working pressure of the stations at normal speed of 740 r.p.m. This type of pump is the simplest now on the market for pumping water either against a high head or low head, and this simplicity was the deciding factor which led to the selection of this style of machinery.

40 The pumps are water-balanced by a piston connected to the last impeller and upon which the water pressure acts, but should any additional end-thrust occur, it would be taken up by the ball bearing provided in the outboard bearing. This ball bearing consists of two rings of 1½-in. diameter steel balls and is water-cooled. The balancing piston is fitted very loosely in order to keep the friction losses small, and as a result a considerable amount of water leaks past it into a chamber at the end of the pump, which is provided with a discharge pipe and valve leading into the suction. By adjusting the valve in this pipe the difference of pressures on the piston can be regulated as desired. The bearings are of the ring-oiled type and are separated from the pump casing by packing glands which prevent foreign matter from entering the bearings. The impellers are of bronze and the shaft of forged steel. All parts of the runners and diffusion vanes are thoroughly lubricated by oil cups on the base of the pumps. A feature is the wide base, shown in Fig. 4, which allows the pump barrel to set low, giving stability.

41 Each combined unit is equipped with automatic and hand control. The pumps are kept primed for instant service and the simple operation of a switch on the main switchboard starts the machine and gives full pressure in about 30 sec.

PRESSURE-REGULATING VALVES

42 A combined regulating and relief valve is interposed between the discharge pipe and the suction pipe of each pump, and set to regulate the discharge of each pump to any predetermined pressure.

43. When the volume of the water discharged by the pump is in excess of that forced into the system, this valve acts as a relief valve and by-passes this excess into the suction to the pump, the pressure on the main distribution system remaining at the predetermined

point. When no water is forced into the distribution system all of th water discharged from the pump is then by-passed into the suction

44 The pressure-regulating valves were made by the Ross Valv Mfg. Co., of Troy, N. Y., and much of the practical success of th station has been due to the accuracy with which they maintain any desired pressure.

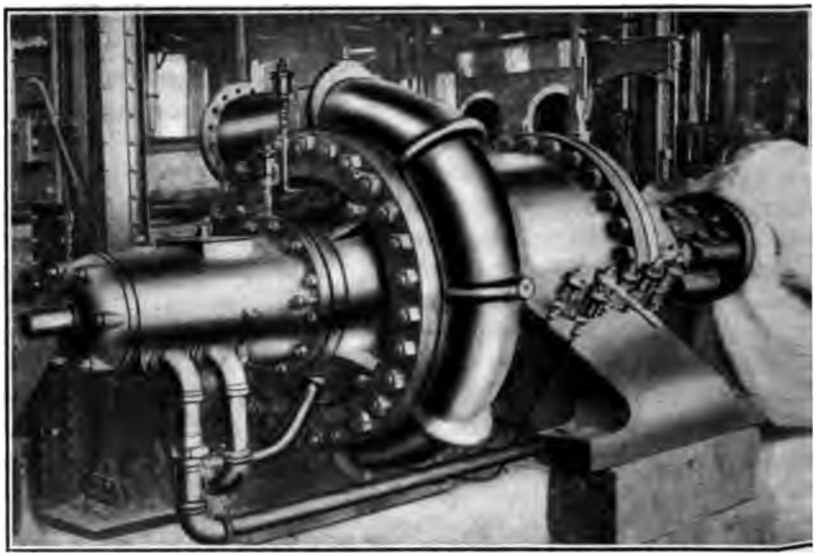

FIG. 4 MULTISTAGE PUMP, CAPACITY 3000 GAL. PER MIN.; MAXIMUM HEAD 300 LB. PER SQ. IN.

PRIMING APPARATUS FOR SALT-WATER SUCTION LINES

45 The priming apparatus in each station consists of three motor-driven vacuum pumps, each arranged to maintain automatically a vacuum of 26 in. in the suction lines. These pumps are of the piston single-action type, one having a displacement capacity of 300 cu. ft. per min. for a piston speed of 200 ft. per min. and each of the others a displacement capacity of 50 cu. ft. with a piston speed of 160 ft. per min.

46 An air-collecting chamber is connected to each of the salt-water suction lines and equipped with water-gage glass and vacuum gage. The air-suction piping between the air chambers and the air pumps is provided with a vertical loop sufficiently high to prevent

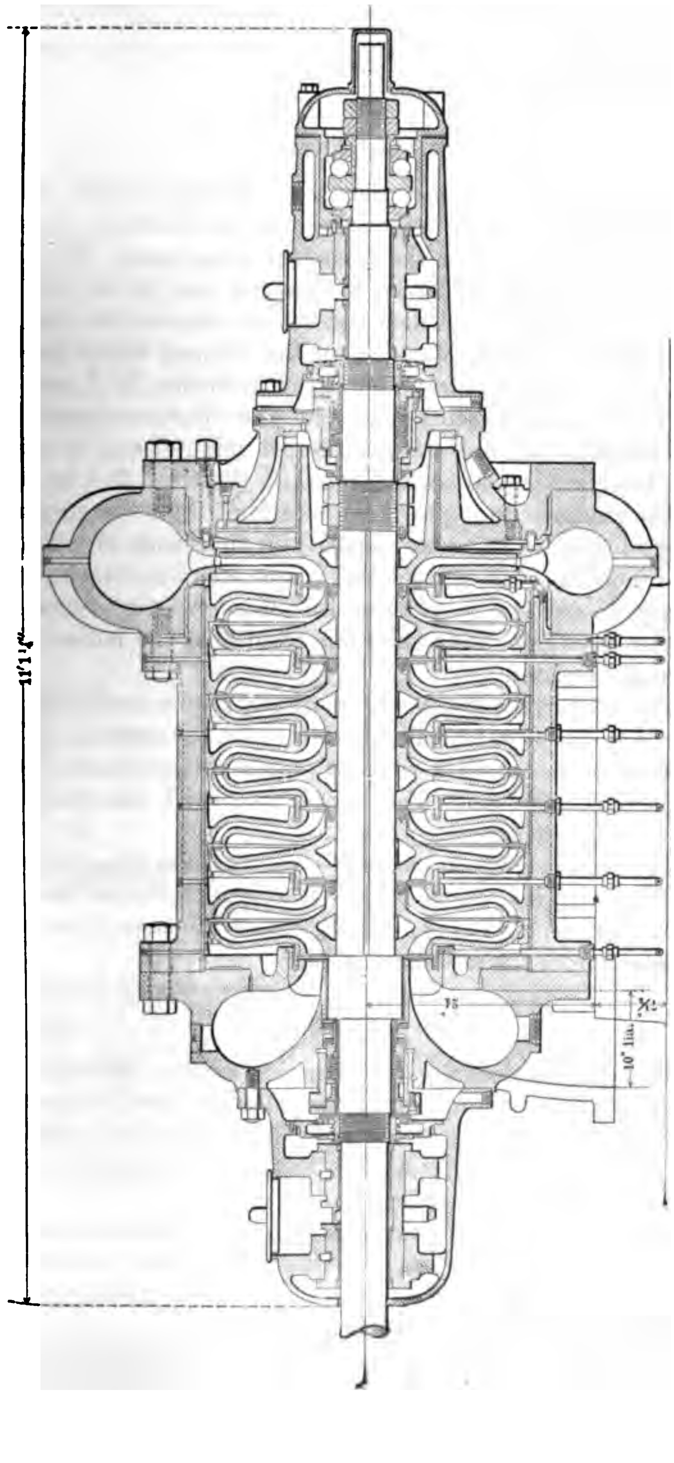

FIG. 5 CROSS-SECTION OF FIVE-STAGE, HIGH-LIFT CENTRIFUGAL PUMP

water being carried over to the pumps. The air pumps are interconnected to each air chamber.

VENTURI METERS

47 Venturi meters for measuring the discharge of water from the station and from one main to the other were set by the contractor on each discharge main and on the cross-connecting main. The meters of the discharge main are 24 in. in diameter and on the cross-over main 12 in. in diameter. These meters were standardized under the direction of F. N. Connet, Manager of the Venturi Meter Sales Department of the Builders Iron Foundry, Providence, R. I., and were provided with dial-indicating gages and also chart-recorders graduated to indicate the flow in gallons per minute; and in addition with an integrating meter which registers the total flow in gallons.

48 The readings during the test were taken by a mercury manometer, graduated to show the capacity in thousands of gallons per minute. For this purpose a Venturi manometer was attached with a temporary connection to each of the 24-in. Venturi tubes. The manometer gave essentially the same reading as the indicating dial on the main register.

49 The Venturi manometer is practically a tube partly filled with mercury, one side of which communicates with the upstream pressure chamber of the meter tube, while the other communicates with the throat-pressure chamber. The connections with the manometer are indicated in the diagram, Fig. 6.

50 The sketch shows a 24-in. high-pressure meter tube, its register-indicator-recorder and manometer. The instruments and meter tube are drawn to scale, but in the pumping station the meter tube is about 75 ft. distant from the instruments.

TESTS OF MACHINERY

51 The specifications for the pumping system provided for an endurance test of each motor and pump lasting 24 hr. without stop, during which time the capacity and efficiency of the pumps and motors were to be determined. The tests were to be in charge of an expert appointed by the commission.

52 The specifications provided for making the test with sea water, but this was later changed to a test with Croton water under the conditions of actual use. In view of this change the contractor increased the efficiency guarantee from 70 to 71 per cent.

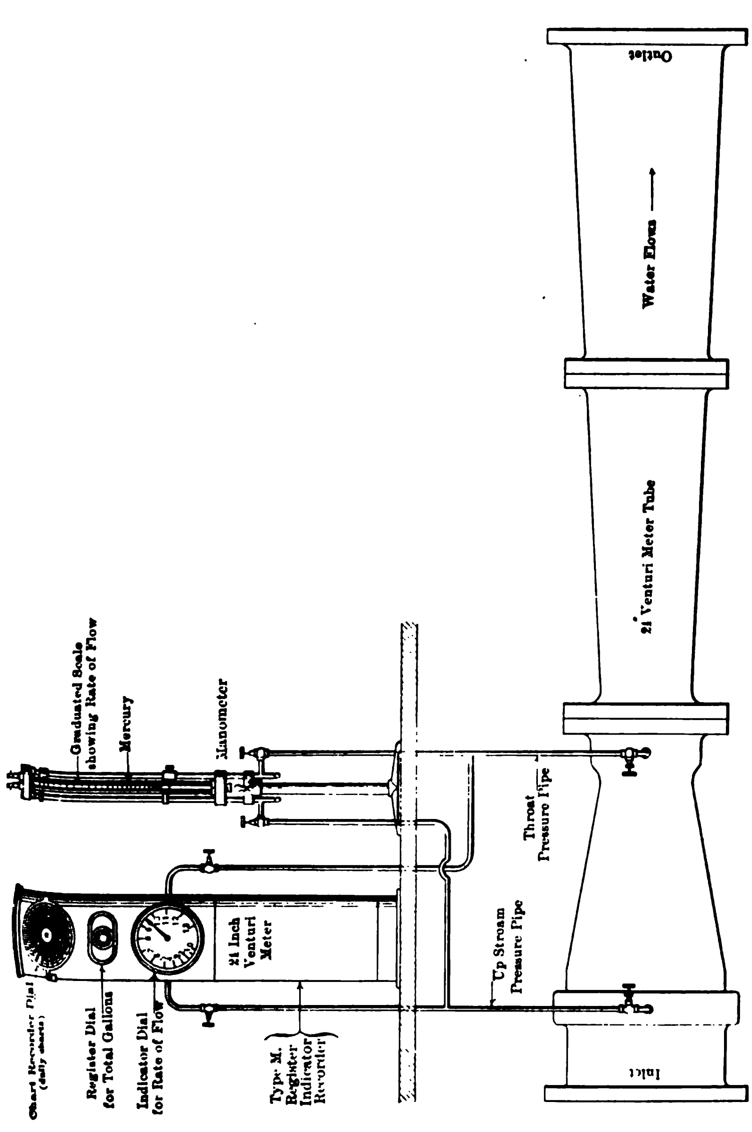

FIG. 6 ARRANGEMENT OF MANOMETER AND ITS CONNECTIONS TO THE VENTURI METER

53 The original specifications called for a capacity of 3000 gal. of sea water per minute against a discharge pressure of 300 lb. per sq. in. and a suction lift not exceeding 20 ft. The total increment of pressure is equivalent to 308.66 lb. from the intake to the delivery side. The Croton pressure varies at the stations in different parts of the day from about 40 lb. to 13 lb. per sq. in. and is affected by the amount of water being drawn from the mains. Consequently, to meet the requirements, the delivery pressure would need to be 308.66 lb. in excess of the intake pressure. There is also a further correction from the fact that sea water is heavier than fresh water and this correction under maximum conditions might amount to 2.5 per cent.

54 The specifications further provided that the brake horse-power developed by the motors under test should be computed from the electrical energy supplied to them, corrected for the efficiency of the motors as determined by the test. They further provided that if the aggregate of all stops exceeded one hour for any motor the test for capacity for such motor was to be run over again for a period of 24 hr.

55 The specifications also provided that the pumping capacity of the apparatus and the efficiency of the pumps should be based on the minimum rate of pumping during any eight consecutive hours of the endurance test, during which none of the motors were stopped.

56 The discharge of the pumps was determined by the reading of the Venturi meters, one of which was located in each discharge line. These readings were under the direction of F. N. Connet, and were checked by observers representing the contractors and also the city.

57 The modified specifications also required that the efficiency of *each pump* should be not less than 70 per cent and its capacity not less than 3000 gal. of sea water when lifted to a pressure equivalent to 308.66 lb. To determine whether the requirement was met, a separate test of each pump was required.

58 The efficiency of the pumps was computed by dividing the horse-power output of the pumps by the horse-power input as received from the motors. The horse-power input was computed as follows:

$$\text{h.p. input} = \frac{\text{total watts}}{746} \times \text{efficiency of motors (93.2 per cent)}$$

The horse-power output was computed as follows: h.p. output

$$= \frac{\text{wt. per gal. } (8.34) \times 2.31 \text{ head in pounds} \times \text{no. of gal. per min.}}{33{,}000}$$

SOUTH STREET STATION TEST

59 The test of the South Street Station was begun at 12:30 p.m. on September 2, 1908, after about 2 hr. of preliminary running for the purpose of adjusting the delivery pressure; it was continued without interruption for 24 hr. With the exception of a short stop of motor No. 2 which was shut down from 2:11 to 2:41 a.m., September 3, to remedy a slight defect in the insulation of the field coils, no pump was stopped. During the time No. 2 was stopped the pressure on the delivery mains fell to about 300 lb.; during the remainder of the test the pressure was maintained at or above the contract requirement, as will be noted by consulting the last column of Table 1.

60 The average results for each hour for the 24-hr. test of all four motors are given in Table 2. The smallest delivery for eight consecutive hours occurred at the last part of the test, when the average capacity, as shown by the readings, was 18,447 gal. per min., and the average efficiency was 72.2 per cent. During this time the average pressure pumped against was 314.5 lb., or an excess of about 6 lb. over contract requirement.

61 It will be noted from the last column of Table 2 that there is considerable variation in the efficiency; that during the first hour the efficiency was less than 70 per cent, whereas during the third and fourth hours the efficiency exceeded 75 per cent. This variation in efficiency was doubtless caused by variation in the amount of water by-passed from the pressure to the suction side of the pump over the balanced piston and through the bearings, and possibly during the first hour by the discharge of some water through the relief valve which was pumped but not metered. The valves for regulating the differential pressure on the balance pistons were nearly closed during the third, fourth and fifth hours of the South Street Station test, but were opened the normal amount for the remaining portion of the test. The amount of water which for maximum difference of pressure may leak around the balance piston of any pump without passing through the meter could not be accurately determined but was estimated to be in excess of 4 per cent. Hence it appears that slight changes in the opening of the valve controlling the differences of pressure at this piston must materially affect the efficiency. The normal

opening of this valve appears to correspond to an efficiency of about 72.5 per cent.

62 During the test of the South Street Station all the bearings ran cool with the exception of those on No. 6 pump, which heated up during the third and fourth hours but were brought to a normal condition without stopping the pump or reducing its load by the application of lubricants and cooling water.

TABLE 1 HOURLY AVERAGE OF READINGS OF DISCHARGE AND INJECTION GAGES ON PUMPS

South Street Pumping Station, September 2 and 3, 1908

Hour	Pump No. 6		Pump No. 4		Pump No. 2		Pump No. 1		Pump No. 3		Average		Net Pressure Lb.
	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	
12:30– 1:15	335.0	21.8	332.3	22.6	329.8	22.5	331.0	24.0	332.9	23.3	332.2	22.8	309.4
1:30– 2:15	347.2	20.9	347.0	22.2	345.5	22.1	343.8	22.6	346.2	22.1	345.9	22.0	323.0
2:30– 3:15	345.0	20.9	343.8	21.8	342.8	22.1	341.0	22.5	343.4	21.8	343.2	21.8	321.4
3:30– 4:00	344.2	20.9	343.8	21.9	340.3	22.3	339.8	22.6	342.2	21.9	342.1	21.9	320.2
4:30– 5:00	341.7	21.1	341.8	21.9	340.8	22.6	339.3	23.1	342.2	22.4	341.2	22.2	319.0
5:30– 6:00	336.2	22.4	336.3	22.9	335.3	23.8	333.3	25.3	335.7	23.6	335.4	23.6	311.8
6:30– 7:00	337.2	24.4	336.8	24.9	334.8	25.3	333.8	26.3	335.7	24.9	335.7	25.2	310.5
7:30– 8:00	339.7	25.1	337.8	25.9	335.8	26.6	334.8	27.3	337.7	25.9	337.2	26.2	311.0
8:30– 9:00	341.2	26.6	339.3	26.9	338.3	27.3	337.3	28.8	338.7	27.1	339.0	27.3	311.7
9:30–10:00	344.7	27.6	342.8	27.9	341.8	28.6	343.3	29.3	344.7	27.9	343.5	28.3	315.2
10:30–11:00	342.2	28.9	341.3	29.1	340.3	29.3	340.8	30.8	342.7	29.6	341.5	29.5	312.0
11:30–12:00	343.7	30.4	344.3	29.9	343.3	30.3	342.3	32.1	343.7	30.6	343.5	30.7	312.8
12:30– 1:00	345.7	30.6	347.3	30.1	347.3	30.8	345.8	32.6	347.9	31.4	346.8	31.1	315.7
1:30– 2:15	334.0	30.9	334.5	30.6*			332.6	33.1	333.9	31.4	333.7	31.5	302.2
2:30– 3:00	332.4	31.1	331.0	30.9			330.8	32.8	332.2	31.6	331.6	31.6	300.0
3:30– 4:00	349.2	31.4	348.3	31.1	346.3	31.6	347.3	33.3	349.2	31.6	348.1	31.8	316.3
4:30– 5:00	347.2	31.1	346.8	30.9	346.3	31.3	345.8	32.8	349.4	31.4	347.1	31.5	315.6
5:30– 6:00	346.7	28.6	346.3	28.4	345.3	29.3	345.3	30.6	348.2	29.4	346.4	29.3	317.1
6:30– 7:00	342.2	27.6	342.3	25.4	340.3	26.1	340.3	27.1	341.2	26.1	341.3	26.5	314.8
7:30– 8:00	332.7	21.6	332.3	22.1	330.3	22.6	329.3	24.1	331.2	22.6	331.2	22.6	308.6
8:30– 9:00	332.7	20.9	332.3	21.4	331.3	22.1	329.3	22.6	331.2	21.6	331.4	21.7	309.7
9:30–10:00	331.7	20.6	332.3	20.9	328.8	21.8	328.3	22.8	331.2	22.1	330.5	21.6	308.9
10:30–11:00	334.2	21.4	336.8	21.6	332.8	22.3	331.8	23.3	336.2	22.4	334.4	22.2	312.2
11:30–12:30	336.0	21.9	338.0	21.9	333.6	22.6	334.0	23.8	337.7	22.6	335.9	22.6	313.3

Readings corrected for error of gage and to center of pumps.
* Pump No. 2 shut down from 2:11 to 2:41 on account of motor.

63 It will be noted from Table 2 that the average results of the 24-hr. test of the South Street Station exceeded the contract requirements in capacity, pressure head and efficiency.

64 The horsepower delivered by the motors during the test averaged for the 24 hr. about 920 or about 15 per cent above rating, without excessive heating.

TABLE 2 COMPUTATION OF PUMP EFFICIENCIES

South Street Pumping Station, September 2 and 3, 1908

Hour Beginning	r.p.m.	Total kw. per hr.	Total h.p. from motors (input)	Gal. per min.	Net pressure lb.	h.p. delivered	Efficiency per cent
12:30 p.m.		3875	4841.4	18334	309.4	3311.6	68.6
1:30	757.0	3851	4811.4	18634	323.9	3523.6	73.2
2:30	755.0	3829	4784.0	19217	321.4	3605.7	75.4
3:30		3819	4771.5	19220	320.2	3592.8	75.3
4:30	755.0	3811	4761.5	19145	319.0	3565.4	75.0
5:30		3837	4794.0	18995	311.8	3457.6	72.1
6:30	755.0	3818	4770.3	18970	310.5	3438.7	72.2
7:30	755.0	3815	4767.5	18980	311.0	3446.0	72.3
8:30	757.0	3863	4826.4	19020	311.7	3461.1	71.6
9:30	756.0	3868	4830.2	19120	315.2	3518.3	72.8
10:30	757.0	3873	4838.9	19095	312.0	3478.1	72.1
11:30	756.5	3859	4821.4	19120	312.8	3491.6	72.4
12:30 a.m.	757.0	3870	4835.2	19175	315.7	3534.1	73.0
1:30	756.5	3672	4587.8	18790	302.0	3315.0	72.5
2:30	757.0	3667	4581.6	18776	300.0	3288.4	71.5
3:30	757.0	3890	4860.2	19190	316.3	3543.5	72.8
4:30	757.5	3891	4861.4	19160	315.6	3530.2	72.6
5:30	754.7	3861	4823.9	19110	317.1	3337.7	73.2
6:30	757.0	3865	4828.9	19005	314.8	3492.7	72.4
7:30	754.7	3706	4630.4	18710	308.6	3370.8	73.0
8:30	745.6	3659	4571.5	18100	309.7	3272.5	71.5
9:30	745.6	3651	4536.5	17890	308.9	3226.2	71.5
10:30	745.0	3619	4521.6	17795	312.2	3243.4	71.8
11:30	747.0	3618	4519.1	17806	313.3	3256.8	71.8
Average	756.1						72.5

Average efficiency, 1st period of 8 hr. = 73.0 per cent.
Average efficiency 2nd period of 8 hr. = 72.3 per cent.
Average efficiency 3rd period of 8 hr. = 72.5 per cent.

No. of cycles per sec. 12:30 p.m. to 6:30 a.m. = 25.5
No. of cycles per sec. 6:30 a.m. to 12:30 p.m. = 25.0

TABLE 3 TEST OF INDIVIDUAL PUMPS

South Street Station, September 3, 1908

Time	No. of pump	Gal. per min.	Pressure delivery	Lb. per Sq. In. Inj.	Lb. per Sq. In. Net	h.p. output	Efficiency of pump
12:58- 1:14	1	3372	344.4	29.3	315.1	620	74.6
1:22- 1:37	2	3809	336.0	27.9	308.1	683	70.1
1:43- 1:58	3	3495	334.0	28.7	305.3	623	73.2
2:03- 2:18	4	3705	334.5	27.8	306.7	662	76.0
2:24- 2:38	6	3740.7	344.5	28.8	315.7	689	77.0

Immediately following the 24-hr. test for capacity.

65 Immediately after the close of the endurance test of 24 hours, a short test was run on each motor separately, which was continued long enough after uniform results were shown to obtain 12 to 15 readings. This test was run for the purpose of ascertaining whether there were deficiencies in any of the individual motors, and to meet the requirements specified in the printed specifications for the work, viz: that each pump should be free from defects, should have a capacity of 3,000 gal. per min. and an efficiency not less than 70 per cent. The results of these tests, Table 3, show that the individual pumps had an efficiency from 4 per cent to 6 per cent in excess of the average when operated together, and that the capacity for the specified discharge pressure was considerably in excess of the requirement of the specification. It is, I believe, generally the case that individual centrifugal pumps delivering water into a main singly show a greater efficiency by from 4 per cent to 6 per cent than the same pumps delivering together into a single main, due probably to less loss in eddy currents and friction head, etc.

GANSEVOORT STREET STATION TEST

66 The endurance test of the Gansevoort Street Station with all the pumps in operation was begun at 9:45 a.m., September 5, after the pumps had been operated for about 20 min. giving uniform results. The test was continued for 24 hr. The method of testing and the various observers were the same as for the tests at the South Street Station and the results are given in Tables 4 to 6.

67 For the Gansevoort Street Station the efficiency average for 24 hr. was 72.9 per cent, with a variation (excepting the first hour) of less than one-half of 1 per cent. It fell below 70 per cent during the first hour, which was due to the opening of an automatic relief valve on pump No. 2, which discharged some of the water into the suction before it had been metered. For that reason the efficiency during the first hour has not been considered in determining the performance of the pumps.

68 The least capacity during the eight consecutive hours when all the water pumped passed through the meters occurs from 10.45 a.m. to 6.45 p.m. The average capacity during this time is 17,419 gal. The average net pressure in pounds is 324.3 which is nearly 16 lb. in excess of the contract requirements. The average efficiency for the period above is 72.90 per cent.

TABLE 4 HOURLY AVERAGE OF READINGS OF DISCHARGE AND INJECTION GAGES ON PUMPS

GANSEVOORT STREET PUMPING STATION, SEPTEMBER 5 AND 6, 1908

Hour	Pump No. 6		Pump No. 4		Pump No. 2		Pump No. 1		Pump No. 3		Average		Net Pressure Lb.
	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	
9:45–10:30	342.4	24.7	342.6	25.6	344.8	24.9	346.9	25.2	345.1	25.3	338.4	25.1	319.3
10:45–11:30	347.4	24.7	347.1	25.7	345.4	25.2	348.4	25.4	346.4	25.2	346.9	25.2	321.7
11:45–12:15	348.9	24.9	348.9	26.2	347.9	25.7	349.4	25.9	347.9	25.9	348.6	25.7	322.9
12:45– 1:15	351.4	25.7	350.9	27.7	350.1	26.7	352.4	26.9	350.4	26.9	351.0	26.8	324.2
1:45– 2:15	352.4	25.7	352.4	27.4	350.9	26.7	353.4	26.7	351.4	26.9	352.1	26.7	325.4
2:45– 3:15	352.9	25.7	352.9	27.7	351.4	26.9	354.9	27.2	351.4	27.2	352.7	26.9	325.8
3:45– 4:15	353.9	26.4	353.9	28.2	353.4	27.7	355.4	27.4	353.4	27.9	354.0	27.5	326.5
4:45– 5:15	354.4	26.9	354.4	28.7	353.4	27.9	355.9	28.2	353.4	28.7	354.3	28.1	326.2
5:45– 6:15	349.9	27.4	349.9	29.7	350.4	28.7	351.9	28.9	350.9	29.2	350.6	28.8	321.8
6:45– 7:15	349.9	28.4	348.9	30.2	349.4	29.2	351.4	29.2	349.4	29.4	349.8	29.3	320.5
7:45– 8:15	351.9	28.9	350.9	30.7	350.4	29.7	352.4	29.2	349.9	29.9	351.1	29.7	321.4
8:45– 9:15	351.9	29.7	351.9	30.9	350.6	29.7	353.4	29.9	350.9	30.7	351.7	30.2	321.5
9:45–10:15	352.4	29.7	354.4	30.9	351.4	30.2	353.4	30.4	352.9	30.9	352.9	30.4	322.5
10:45–11:15	354.4	30.4	353.9	31.2	353.4	30.2	354.9	30.9	353.4	31.2	354.0	30.8	323.2
11:45–12:15	354.4	31.2	353.4	31.7	352.9	31.2	353.9	31.7	352.9	31.2	353.5	31.4	322.1
12:45– 1:15	353.9	31.2	352.4	31.7	352.9	31.2	354.9	31.7	353.4	31.2	353.5	31.4	322.1
1:45– 2:15	352.4	31.2	351.9	32.2	352.9	31.2	354.4	31.9	353.4	31.2	353.0	31.5	321.5
2:45– 3:15	350.4	31.7	349.4	32.2	349.4	31.4	351.9	32.2	350.4	31.7	350.3	31.8	318.5
3:45– 4:15	350.9	32.2	350.4	32.7	348.9	31.2	351.4	32.7	348.4	31.4	350.0	32.0	318.0
4:45– 5:15	350.9	31.9	350.4	32.4	350.4	31.2	352.9	32.2	349.9	31.2	350.9	31.8	319.1
5:45– 6:15	352.4	31.2	351.9	32.2	352.4	30.7	353.4	31.7	351.9	31.4	352.4	31.4	321.0
6:45– 7:15	350.9	29.9	349.9	31.2	349.4	29.9	351.9	30.7	349.4	30.9	350.3	30.5	319.8
7:45– 8:15	349.4	28.7	349.4	29.7	347.4	29.4	349.9	29.2	347.9	29.4	348.8	29.3	319.5
8:45– 9:45	348.1	27.4	347.7	28.9	346.4	28.0	348.4	28.0	346.4	28.5	347.4	28.2	319.2

Readings corrected for error of gage and to center of pumps.

69 The average capacity for the entire test is 17,867 gal. which was obtained with an average speed of 753.6 r.p.m.

70 Immediately after the completion of the endurance test of 24 hours duration, each pump was tested when operating alone for a period sufficiently long to obtain 12 to 15 readings after they had become practically uniform. These tests gave in every case an efficiency several per cent greater than that obtained when the pumps were all discharging into the same main.

CONCLUSIONS

71 It appears from the endurance test in each station that the capacity, efficiency and pressure exceeded the contract requirements by a large margin, and that during the endurance test no mechanical

TABLE 5 COMPUTATION OF PUMP EFFICIENCIES

GANSEVOORT STREET PUMPING STATION, SEPTEMBER 5 AND 6, 1908

Hour Beginning	r.p.m.	Total kw. per hr.	Total h.p. from motors (input)	Gal. per min.	Net pressure lb.	h.p. delivered	Efficien per cen
9:45 a.m.	740	3671	4586.5	17107	319.3	3188.9	69.5
10:45	749	3589	4484.2	17310	321.7	3251.0	72.5
11:45	750	3591	4486.7	17290	322.9	3259.3	72.9
12:45	752	3591	4486.7	17280	324.2	3270.5	72.9
1:45 p.m.	752	3604	4502.9	17285	325.4	3283.6	72.9
2:45	753	3604	4502.9	17315	325.8	3293.3	73.3
3:45	753	3630	4535.3	17345	326.5	3306.1	72.9
4:45	753	3685	4604.0	17670	326.2	3365.0	73.1
5:45	756	3696	4617.9	17855	321.8	3354.4	72.6
6:45	756	3661	4574.0	17825	320.5	3335.2	73.4
7:45	754	3676	4592.9	17775	321.4	3335.2	73.3
8:45	753	3685	4604.0	17755	321.5	3332.5	72.8
9:45	755	3657	4569.0	17720	322.5	3336.2	72.9
10:45	755	3693	4614.2	17755	323.2	3350.1	72.7
11:45	756	3704	4627.9	17830	322.1	3352.8	72.6
12:45 a.m.	756	3753	4689.0	18195	322.1	3421.4	73.0
1:45	756	3760	4697.8	18310	321.5	3436.6	73.3
2:45	756	3735	4665.5	18315	318.5	3405.5	73.0
3:45	755	3725	4654.0	18290	318.0	3395.5	73.0
4:45	756	3743	4677.6	18315	319.1	3411.9	73.0
5:45	756	3784	4727.9	18330	321.0	3435.0	72.7
6:45	755	3747	4681.6	18315	319.8	3419.4	73.0
7:45	755	3723	4656.5	18255	319.5	3405.0	73.1
8:45	755	3722	4655.3	18189	319.2	3389.5	72.8
Average	753.6						72.9

Average efficiency, 1st period of 8 hr. = 72.9 per cent.
Average efficiency, 2d period of 8 hr. = 73.0 per cent.
Average efficiency, 3d period of 8 hr. = 72.9 per cent.

No. of cycles per sec. 9:45 a.m. to 2:45 p.m. = 25.00
No. of cycles per sec. 2.45 p.m. to 4.45 p.m. = 25.25
No. of cycles per sec. 4:45 p.m. to 6:45 p.m. = 25.50
No. of cycles per sec. 6:45 p.m. to 7:45 p.m. = 25.00
No. of cycles per sec. 7:45 p.m. to 9:45 p.m. = 25.25
No. of cycles per sec. 9:45 p.m. to 9:45 a.m. = 25.50

or electrical defects were observed. During the test of the Sout Street Station one of the pumps was stopped for half an hour repair the motor insulation, while during the test of the Gansevoo Street Station no stop was made. The bearings in both statio were in perfect condition at the end of the test and the temperatu of the motors not sufficiently high to interfere with the continuo operation for a longer period. Apparently the endurance test coul have been continued indefinitely without injuriously overworkir or overloading the pumps and motors.

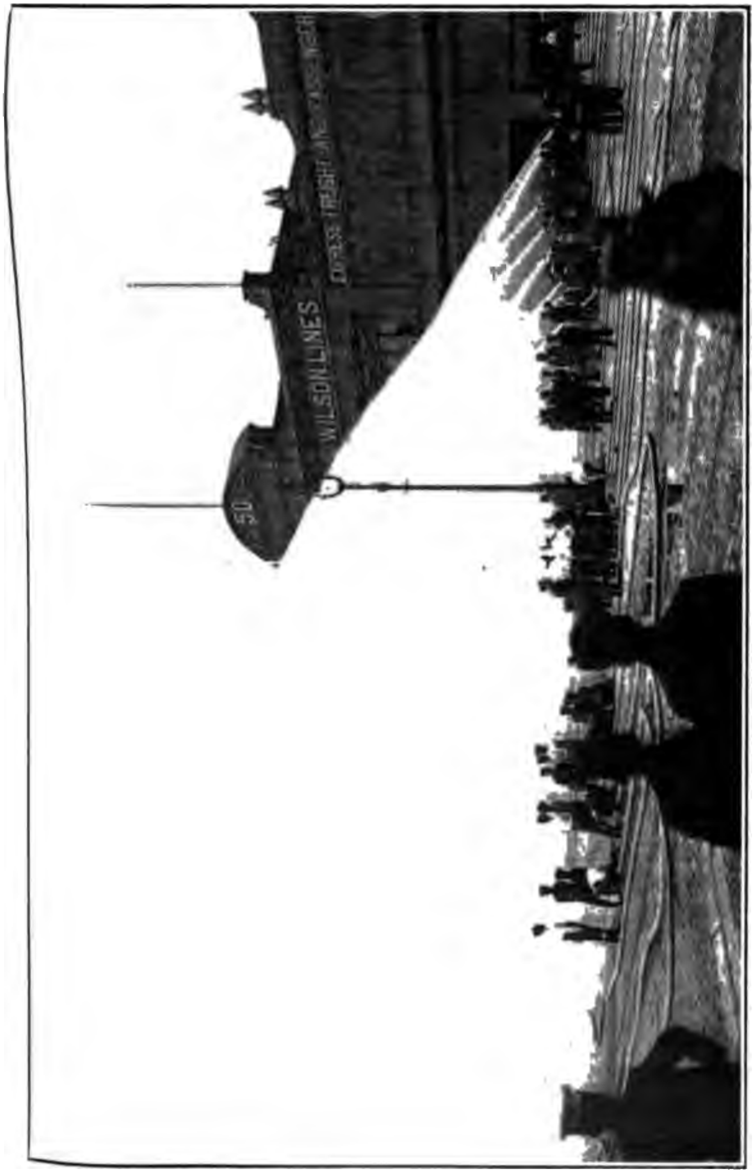

Fig. 7 View Showing Test of Fire System

72 The specifications call for pumping sea water, which mos authorities consider to be approximately 2.5 per cent heavier tha fresh water. The effect of substituting sea water for fresh wate would have been to reduce the capacity of the pump by about 2 per cent for the same horse-power delivered by the motor, withou sensibly affecting the efficiency. Because of the large capacit

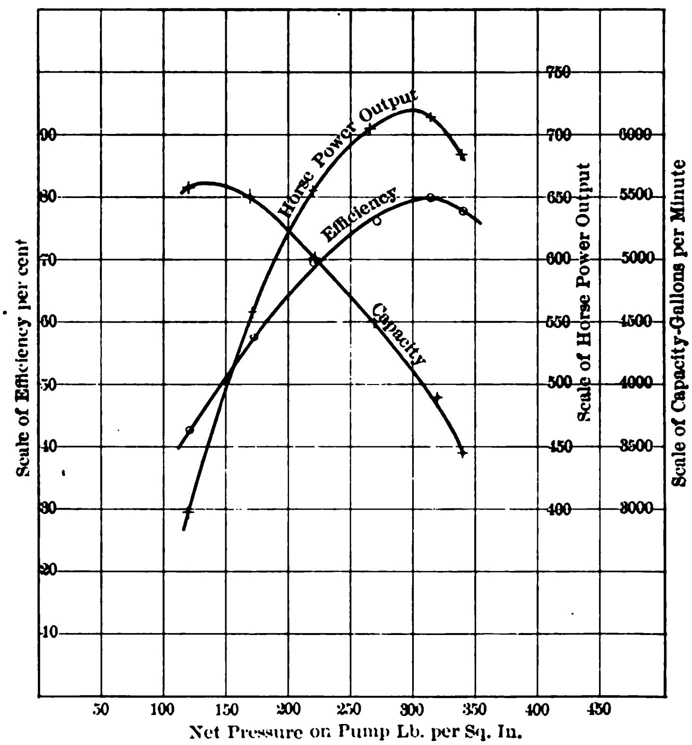

FIG. 8 CHARACTERISTIC CURVES OF THE PUMP FOR VARYING DISCHARGE-PRESSURES

RESULTS OF TESTS

shown by the pump, this does not materially affect the results in relation to the contract requirements.

73 The data and results of the tests at the two stations are given concisely in the tables. The efficiency is given as computed for each hour, and shows a slight variation which probably can be accounted for by changes in the amount of water leaking past the balancing piston. The individual pump tests at the South Street

Station show a variation in efficiency from 70 per cent to 77 per cent, and at the Gansevoort Street Station from 70 per cent to 79 per cent. This variation may have been due to the structure of the pumps but in my opinion is more probably due to variable leakage past the balancing piston or through the relief valves.

74 Pump No. 6 at the Gansevoort Street Station was tested with varying openings of the valve in the discharge pipe. The results are shown in the latter half of Table 6.

TABLE 6 TEST OF INDIVIDUAL PUMPS

GANSEVOORT STREET STATION, SEPTEMBER 6, 1908

Time	No. of pump	Elect. h.p. input	Gal. per min. Hg. Col.	Pressure delivery	LB. PER SQ. IN. Inj.	LB. PER SQ. IN. Net	h.p. output	Efficiency of pump
10:05–10:31	1	916	3800	356.8	35.4	321.4	711	77.6
10:36–10:51	2	877	3800	350.8	35.1	315.7	700	70.8
10:54–11:12	3	920.5	3820	350.4	34.1	316.3	703	78.0
11:17–11:30	4	892	3751.4	352.5	35.6	316.9	695	77.7
11:37–11:53	6	899	3880	350.9	35.2	315.7	714	79.4
11:55–12:03	6	880.3	3457	376.1	36.0	340.1	686	77.9
12:03–12:07	6	929	4500	304.4	34.6	269.8	708	76.1
12:09–12:13	6	946	5070	255.6	33.6	222.0	654	69.4
12:24–12:28	6	952	5500	207.4	33.2	174.2	559	58.7
12:32–12:36	6	927	5588	155.2	33.2	122.0	397	42.8

Immediately following the 24-hr. test for capacity.

PRACTICAL RESULTS FROM THE NEW SYSTEM

75 The high-pressure fire system in New York, which was put officially into service on July 6, 1908, has been successfully operated at many fires, but it had a crucial test on January 7, 8 and 9, 1909, when it was brought into service for five simultaneous fires, three of them of more than the usual extent and activity, and one particularly so. Information upon the results attained with the system and the amount of water consumed was given by Chief Engineer I. M. de Varona and published in the Engineering News of February 11, 1909.

76 The fires occurred at Hudson and Franklin Streets, Hester Street and the Bowery, Houston Street and Broadway, Sixth Avenue and 17th Street, and Houston Street and the Bowery. The situation became so dangerous that every engine south of 37th Street, or 40 engines, were summoned, as well as a force consisting of 12 battalion chiefs and more than 600 men, but there was no need to use a single one of the engines.

77 As the violence of the fires increased, additional pumps were brought into service, so that at one time four pumps and motors were in commission at the South Street Station and three pumps at the

TABLE 7 SPECIFIED CHEMICAL ANALYSIS FOR PUMP MATERIALS

	Nickel steel	Medium steel	Steel forging	Steel casting
Parts of 1 per cent				
Phosphorus not to exceed	0.04	0.10	0.04	0.05
Sulphur not to exceed	0.04		0.04	0.05
Tensile strength at rupture, pounds	100,000	650,000	75,000	65,000
Tensile strength at elastic limit, pounds	65,000	32,500	38,000	32,000
Per cent elongation in 8 in	2	22		
Per cent elongation in 2 in	22		22	18
Contraction of area per cent	32		32	24
Carbon not less than	20 parts of 1%			
Nickel percentage	21 to 24			

Gansevoort Street Station, delivering 35,500 gal. per min. against an average pressure of 225 lb. at the pumps and 205 lb. at the hydrants. During the operation of the pumps 14,095,000 gal. were pumped as recorded by the meters, and the current used was 81,450 kw-hr., the cost of which was $1222.

DISCUSSION AT NEW YORK

Prof. George F. Sever.[1] The electrical features of this installation are of much interest but the reasons for selecting that system which is now in operation should be given. In the discussion of this problem both alternating and direct-current power were considered for the operation of the motor-driven pumps, and alternating-current power was decided upon. The reasons for such selection I have noted herewith:

a Absolute simplicity, which is the key-note of the electrical end of this power installation.

[1] Professor of Electrical Engineering, Columbia University.

Note.—The high-pressure system was designed by I. M. de Varona, Chief Engineer of the Department of Water Supply, Gas and Electricity of New York. It was also constructed under his supervision. The construction of the electrical machinery was supervised by Prof. Geo. F. Sever as Consulting Engineer. The details of construction were in charge of Thomas J. Gannon, John P. Reynolds and Henry B. Machen, assistant engineers of the department. The machinery of each station was designed and erected by the Allis-Chalmers Co. of Milwaukee.

b Commutating apparatus and brushes are entirely absent.

c Induction motors provide very quick starting when it is necessary to operate the station on a fire signal.

d There is less expense for copper in the distribution system to insure continuity of service.

e The induction motor is a less expensive apparatus than the direct-current motor.

f With the induction motor there are absolutely no exposed live circuits in the station, as there might be with a direct-current apparatus. The final decision was for 3-phase service at 6600 volts and 25 cycles. It was decided that it would not be desirable to establish a power house to be operated by the city because it would be a municipal plant.

2 In order to insure continuity of service there is brought to each pumping station an independent feeder from each of the two Waterside stations of the New York Edison Company. There is also brought to each pumping station an independent feeder from the nearest substation of the New York Edison Company, as follows: to the Gansevoort Street station two feeders from the Horatio Street sub-station, and to the South Street station two feeders from the Duane Street station of the company. Hence there are really four independent sources of power supply for each pumping station, assuring practically no possibility of shutdown.

3 The contract for electric power for the Manhattan stations was let to the New York Edison Company. This contract provides for two payments, the first for a reservation of 3250 kw. capacity, of generating, distributing and controlling apparatus, available at either pumping station at an instant's notice, or practically without any notice at all. Thus four pumps can be thrown on with absolutely no notice to the New York Edison Company that they are to be used. For that reservation, and care and maintenance of the whole distributing system, the city pays about $63,000 per year, and the city also pays one and one-half cents per kw-hr. for all high-tension power used in each station.

4 Another stipulation in the contract may be of interest to engineers as it provides for the protection of the city. This stipulation is as follows: "If the contractor, under the terms of this contract, shall fail to maintain and deliver a continuous and uninterrupted supply of electric power when required, the contractors shall and will pay to the city the sum of five hundred dollars per minute for

each minute's interruption or delay of electric power supply after the power has been interrupted or delayed for three consecutive minutes." So, if they cannot deliver power after an interruption of three minutes, immediately a charge of $500 per min. is imposed and is deducted from the bills which the New York Edison Company renders.

5 The operation of both these stations is extremely simple. The handle of the oil switch is turned, throwing the 6600 volts directly on the stator of the motor. By turning a hand wheel, the motor is brought up to speed in less than 33 sec., and in starting the current is not supposed to exceed 150 per cent of the full-load current, which is 64 amperes. As far as I have observed the operation of the stations, there has been absolutely no trouble from the electrical end, no trouble with the feeder system, and none with the motors, and I think the City of New York has two plants which will give it for many years to come absolutely no trouble whatsoever.

Wm. M. White. The paper deals with questions in which I am directly interested. The methods employed in making the tests were probably the best that could have been selected. There is probably no more accurate method of determining the quantity of water delivered by a pump than by the venturi meter, especially when in the hands of an expert who is familiar with its workings. The venturi meter, as Professor Carpenter says, has been used for a number of years; it has been tested in various ways and proved to give accurate results. The power delivered to the pumps can be most carefully measured by electrical instruments.

2 The writer accepts without question the various efficiencies obtained and presented by the author, who states, calling attention to the variation in efficiencies obtained, that the individual observations do not agree as closely as he would like. I do not think Professor Carpenter should offer any apology as the results seem to agree very closely, and certainly are as accurate as are generally obtained on work of this kind. The efficiencies obtained on these pumps, though not the highest that have been obtained, are as high as is usual for similar conditions of head, capacity and speed. The designers of the pumps deserve credit for the performance shown by the pumps.

3 I am at a loss to find a reason for the variation in efficiencies of the pumps, as mentioned in Par. 65, where it is stated that individual pumps delivering water into a main singly show greater efficiency

than the same pumps delivering together into a single main. I assume, of course, that the variation in efficiency refers to the pumps when they are delivering exactly the same quantity against the same head at the same speed, whether working singly or in parallel. In the normal operation of pumps, it would be a fact that when one pump was operating from a suction main to a discharge main, the efficiency of that pump would be different from what it would be when working with another pump from the same suction main and discharging into the same discharge main, because the two pumps would usually be working against a higher head than when a pump was working singly. The increased head on the pumps would mean a decrease of capacity, and the increase of power demanded by two motors instead of one would mean a slight increase in line loss, which would again slightly decrease the speed and slightly change the conditions of operation for two pumps over that which would exist when one pump only was in operation. Of course, under these conditions, the two pumps would show different efficiencies, because the efficiency curve of a pump varies as its capacity and head.

4 I do not believe, however, that this is the condition to which Professor Carpenter refers. I assume that he has corrected for this difference, and has obtained from two pumps working in parallel the same capacities, heads and speeds as though one pump were in operation, and that under this latter condition he finds the difference in efficiency in the two pumps. If this be a fact, it is the most important point brought out from a designer's point of view.

5 I am at this time attempting to duplicate the conditions, to see whether the efficiencies are different under the same conditions of capacity, head and speed, as mentioned by Professor Carpenter.

George L. Fowler. A number of years ago I was associated with Joseph Edwards, who at that time had the contract for excavating the ship channel in New York Harbor, probably one of the first, if not the first, very large hydraulic engineering projects successfully accomplished by the contractor and to the satisfaction of the Government.

2 The ship channel leading from the Narrows down to Sandy Hook and out to sea, is about 15 miles long, and runs almost due south first, turning to nearly due east before reaching Sandy Hook, and passing through Gedney Channel to the sea. Cutting across it is

the Swash Channel, not used by any deep-draft boats. When the work was undertaken New York Harbor was shoal at two points on the Gedney Channel and the ship channel, where the water depth was a little less than 24 ft. The Government had a survey made and an estimate of costs based on material actually removed by the ordinary methods of dredging. Through the open space from Sandy Hook to Coney Island the whole lower bay is subject to all the winds coming in from the Atlantic on the east and across Raritan Bay, so that the water is nearly always rough. Two contractors had attempted the work by ordinary bucket dredging and both had failed.

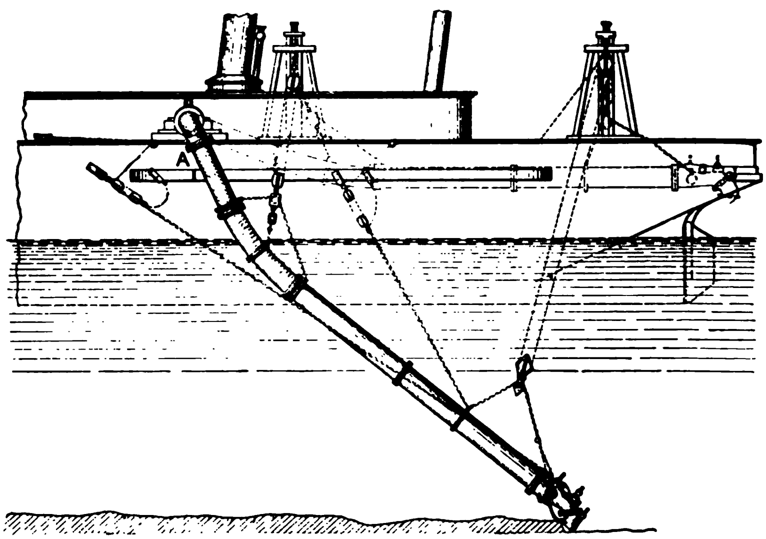

Fig. 1 Hydraulic Dredger for Deepening Ship Channels

3 In the ship channel the material was sand and sedimentary clay, lying over hard sand; in the Gedney Channel it was gravel, shell and sand, for two feet overlying hard shingle. Hydraulic dredging was specially suited for this kind of work, and many kinds of material were removed from the channel besides the ordinary silt.

4 Three sea-going vessels were built for this work by the Joseph Edwards Company: the Reliance, the Advance, and the Mt. Waldo. Fig. 1 shows the general arrangement of the ships. At *A* is the long drag aft, where the pipe goes into the vessel and where the pumps are located, each driven by a 192-h.p. engine at 178 r.p.m. The suction and delivery pipes were 15 in. in diameter, with a shell of 40 in. The

pumps delivered 10,000 gal. per min. at a velocity of 1100 ft. The efficiency was thus between 65 and 70 per cent, although in later tests made by the Government, when nothing but water passed through the pipes, the efficiency rose to as high as 80 per cent.

5 The shoe used is a hook that drags along the bottom, chains being fastened to the vessel for this purpose. The vessel never stopped from morning to night, simply running out to sea, dumping, and coming back again to work.

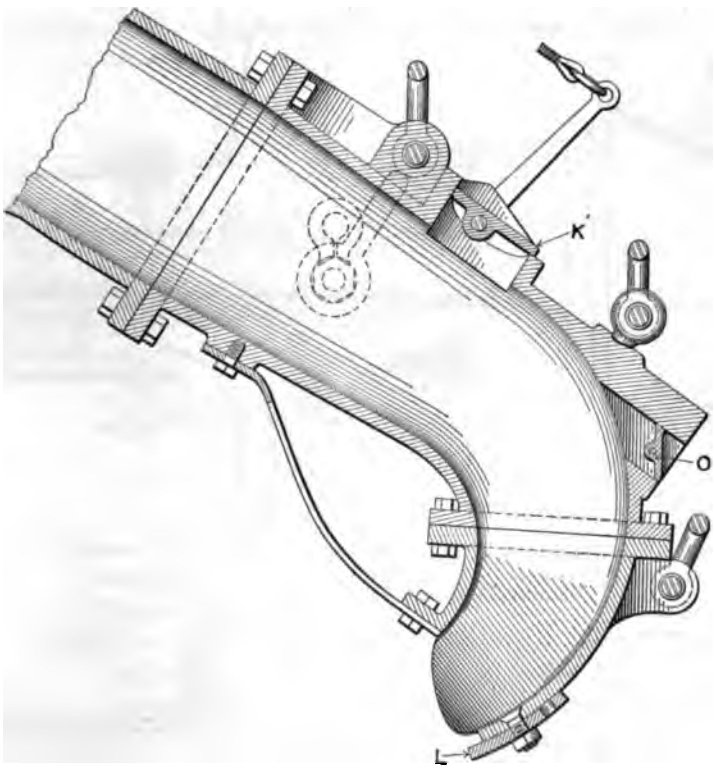

Fig. 2 Detail of End of Suction Line

6 At the point L, Fig. 2, was the heavy shoe that served to dig into the mud and gravel. At O was a butterfly valve, kept open all the time to admit water above the drag to mix with the material raised. At the bottom K was another valve which could be opened in an emergency, in case not enough water was admitted at O.

7 The pump itself was of a plain centrifugal type, 40 in. in diameter, with vanes cut away at the center, as shown in Fig. 3. Because of this arrangement, the material would come in at *C* and out of the

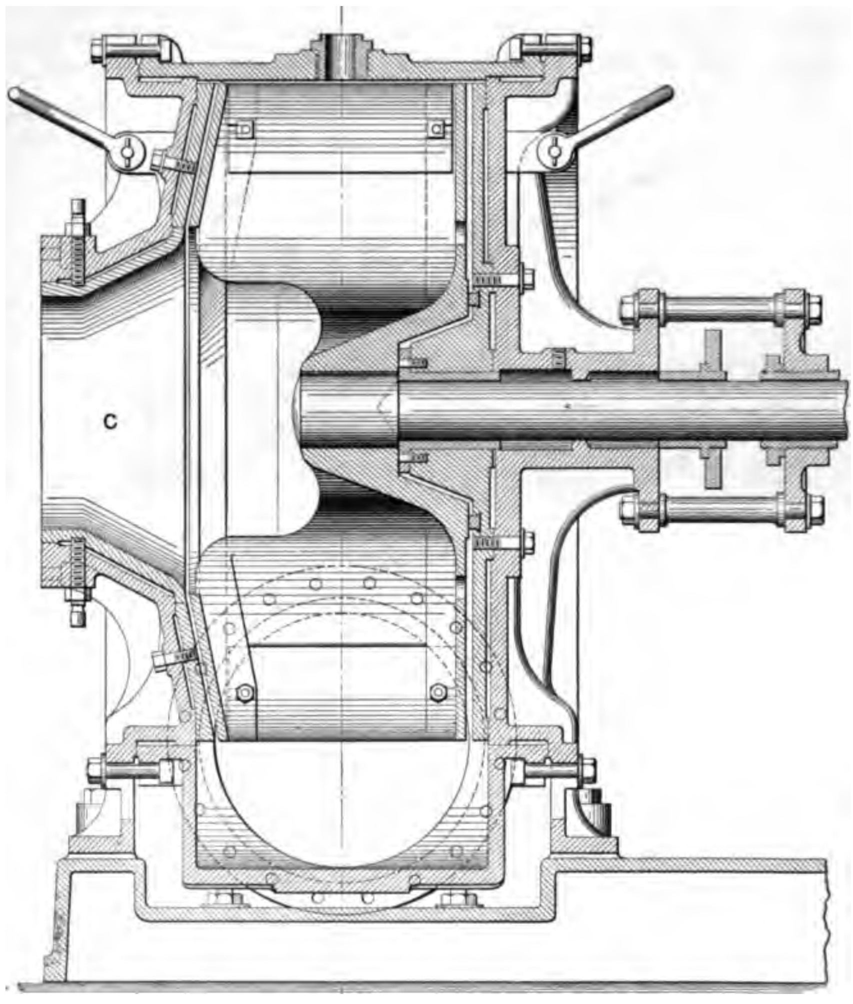

Fig. 3 Sectional View of Centrifugal Pump for Dredging

vanes at the discharge, without damaging the pump when heavy substances were drawn in. The three vanes were made with wings bolted on, and accessible from both sides. The thrust was taken up by the bearing at T (Fig. 4), the nuts marked m being screwed into a head carried by the bars O, bringing the thrust plates at the point i. The reason for threading the nut m was to adjust it to the vanes in proper relative position to the sides of the pump. That is a simple construction maintained ever since, with the exception that ball bearings are now used.

8 Although the pumps were originally intended to take water and other loose material, such as sand and gravel, they proved capable of lifting practically anything that came in their way. The three following specimens are interesting as showing the pumps' lifting power:

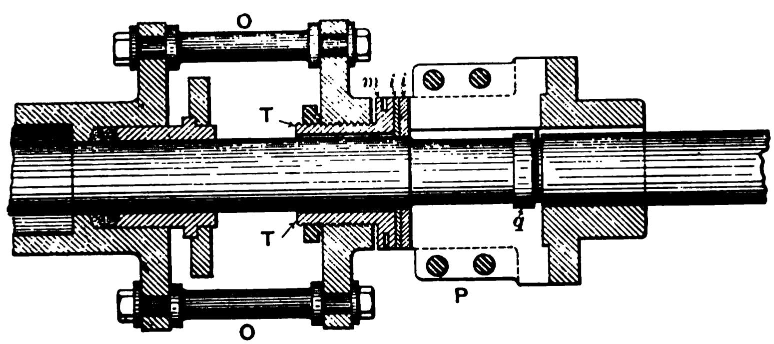

Fig. 4 Detail of Thrust Bearing of Pump

a A piece of shaft weighing 70 lb. raised and passed by a 15-in. dredging pump; improvement of New York Harbor, Steamer Reliance.

b A piece of tree root raised and passed by a 12-in. pump from 14 ft. of water at Miami, Fla.; Florida East Coast Railway Company improvements.

c A piece of pig iron measuring 11½ in. by 4¾ in. by 3½ in. and weighing 35 lb. raised and passed by an 8-in. special cataract wrecking-pump from 15 ft. of water from the wreck of a canal boat sunk at Puas Dock, Yonkers, N. Y.; by Baxter Wrecking Company, New York.

9 For hydraulic dredging, the Government pays by the scow load and gets what is excavated. In ordinary hydraulic dredging, like that in the ship channel, about 15 per cent of the pump discharge was solid matter. About 40 per cent in excess of the amount deposited

in the bins went overboard with the overflow, and was carried out to the flats at the sides by the cross currents, which also carried the loose material stirred up by the drag. The result was that the Government obtained an excavation about 70 per cent in excess of what would have been obtained had all of the material removed from the bottom been caught in the bins. This, of course, greatly reduced the actual cost of the excavation. For example: the last contract made on the ship channel was at the rate of 16¾ cents per yard, while with the allowance indicated, above the actual cost per yard—channel measurement—it was about 11 cents.

10 As for the time of loading, some records indicate that this ship, 157 ft. long and with a capacity of 650 cu. yd., was loaded in 48 min.; there are also records of its being loaded at the rate of 16 cu. yd. per min., of solid matter placed in the bins; and records of its taking out to sea nearly 4000 cu. yd. per day. The vessel was worked in all kinds of weather, even when tackles had to be used to board her; and yet the ship was taking her load steadily. Except in the case of an actual breakdown the work could be carried on for 16 hr. per day.

John H. Norris. In a pumping plant of the character described, this type of equipment seems in the present state of the art the most suitable that could have been selected. I would like, in this connection, to call attention to another type of installation for service of this kind, though not on so large a scale, which appeals to me as being more desirable than the electric-driven centrifugal pumping plant taking its power from a public utilities company.

2 At Coney Island was installed the first plant operated by the City of New York for fire protection by means of water delivered into mains under high pressure, with the idea of taking care of a restricted area where there was great danger from fire.

3 This plant consists of three 150-h.p. three-cylinder, vertical gas engines direct-connected to triplex pumps, each unit capable of pumping 1500 gal. per min. against a pressure of 150 lb. These engines take their fuel from the mains of the local gas company and can be arranged if necessary to run on gasolene. They are installed in a building on city property and are arranged to take their water supply from the city mains or from Coney Island Creek, within 50 ft. of the pumping station. The engines are started with compressed air, and the three units can be started up in less than three minutes. On every occasion they have been found ready for service whenever

the demand was made upon them. The cost of this pumping station was as follows:

Building	$10,000	
Equipment	37,000	
		$47,000
The annual operating expenses are:		
Labor	$13,140.00	
Supplies and Repairs	897.27	
Fuel	150.00	
		$14,187.27

4 By comparing the foregoing figures it will be evident that for service smaller than is required in the City of New York, the gas-engine-operated triplex pump gives an economical equipment that can be allowed to stand idle for any length of time and yet be ready for instant service.

5 New York City pays the New York Edison Company an annual charge of $90,000 for the privilege of calling for sufficient current to operate the equipment at any time. This item capitalized at 5 per cent would pay for a good-sized gas-engine plant.

6 The following data were taken from the capacity tests of the Coney Island units:

Duration of test	14	hr.
Average piston speed of pump	90.3	ft. per. min.
Total head pumped against	156.5	lb.
Average pump horsepower for each unit	142.2	h.p.
Average gas consumed per hour for the 3 units	8914.0	ft.
Average capacity	4512.0	gal. per min.
Slip of pump	3.45	per cent
Average efficiency of pumps	82.00	per cent

J. R. Bibbins. Although Professor Carpenter's paper deals primarily with multistage pumps, I wish to direct attention to the question of motive power, upon which the success or failure of the system practically depends. We have seen excellent examples of two systems diametrically opposed in regard to power supply—the electrical and the gas-driven system. Under certain conditions, both are extremely serviceable. The first high-pressure installation on a large scale, in this country, was the gas-driven system at Philadelphia. Although I have not had an opportunity to follow the results of that station for the past two or three years, the results obtained and published for the first year or so showed that such a system of gas-driven pumps merits every consideration.

2 *First as to the security of power supply:* In Philadelphia the Delaware Avenue station receives its gas supply directly from a 24-in. trunk main running between two very large gas holders, located in different parts of the city. Roughly, the pipe line measures four miles in length, its capacity constituting a considerable reserve in itself, if both the holders were unavailable. There is no intermediary apparatus whatever between the pipe line and the engine; that is, the plant may draw directly on these two large holders of several million cubic feet capacity. This constitutes a very safe and reliable source of motive power which can hardly be paralleled except, perhaps, by the situation in the New York electric service, where there are so many stations to draw from.

3 In this connection, I would like to ask whether it is at present possible to utilize the storage battery capacity in the various sub-stations for reserve service at the high-pressure pumping station. It is stated that the storage batteries are available for reserve in emergencies, such as discontinuance of the main high-tension current supply. I am under the impression that an inverted rotary requires a direct-driven exciter to maintain a definite frequency and prevent racing. Without special controlling apparatus, this inversion would be impossible in the ordinary sub-station equipment. Possibly special provision has been made in the New York systems, in which case, the security of power supply is certainly beyond criticism. In other words, would it be possible to invert the synchronous converters on short notice?

4 *Second, quick starting:* It seems to be a fact that a large part of the minimum time required for the starting of a fire-service station is consumed in the operation of the motor-driven by-pass valves. In Philadelphia these valves are operated from an independent supply, as in New York, and at least fifteen seconds are required to close them; whereas the engines are brought up to speed within half a minute from the time the signal is given, the remaining time being usually consumed in closing this motor-driven valve.

5 The various tests of the Philadelphia plant showed that each of the units could be readily put on the line in well under one minute. It is an interesting fact that the original underwriters' tests specified the time limit as twelve minutes for the starting of the first three units, whereas the whole station can be started in that time, and has been started in seven minutes.

6 During the 36 days of preliminary service trials of the Philadelphia station, out of one hundred alarms given, only four misses were made in getting any of the eleven units started. In not a single instance has the station, as a whole, failed to respond to the service, at least during the period over which my observation extended. This has been accomplished with the regular operating force of three men.

7 *Third, in regard to the cost of service at Philadelphia;* The only data on a large fire available, are those of the fire in the Coates Publishing House, which lasted about nineteen hours. The average cost for pumping was about six cents per thousand gallons, including gas, wages and supplies. The cost of the large East Side service, cited in the paper, is about nine cents for power alone, and I think this does not include the readiness-to-serve factor. On the other hand, it is patent that the cost of service in either the gas or the electrical station is relatively unimportant. The main desideratum is reliability.

8 Finally, I desire to advance an argument for the development of a new type of pump unit, namely, a high-speed gas-driven centrifugal pump. Some time ago, in connection with water-works service, I found great difficulty, even with the present high-speed single-acting gas engine, in matching engine speeds with those required in centrifugal pump work However, for the pressure necessary in water-works practice, about 125 lb., one or two sizes of engines were found to be directly applicable to multistage pumps, with fair proportion of parts and good efficiencies. It seems possible to adopt a modified type of gas engine which would permit the direct connection mentioned.

9 This modification would naturally follow along lines of short stroke and high piston speeds with perhaps four cylinders. The engines at Philadelphia were designed with a piston speed of but 730 ft. per min. with a 22-in. stroke. This might be increased to 1000 ft. per min. without exceeding present-day limits, especially for units designed for occasional service. Such a unit would find immediate application in many industries and would combine the high economy of the gas engine with the simplicity of the centrifugal pump. The efficiencies shown by Professor Carpenter place the centrifugal pump in a position of closest competition with reciprocating pumping units.

J. J. Brown. I recently made a series of tests on three 6-in., 8-stage centrifugal pumps, each designed for 1000 gal. per min. and 560 lb. pressure at 1200 r.p.m. One of these pumps gave an efficiency from wire to water of 71 per cent, or a pump efficiency of 76 per cent.

regret that Professor Carpenter did not give the results of his tests on the New York fire-service pumps at lower capacities. All of the tests were made at capacities considerably in excess of that for which the pumps were designed and they apparently show their best efficiency at approximately 25 per cent over the normal rating. This increased efficiency at excess capacity seems to be apparent in several recent tests made on high-lift centrifugal pumps. The 8-stage machines previously referred to give their best efficiency at 1300 gal., or about 30 per cent over rating.

2 Mr. White has raised a question as to the difference in efficiency between the New York fire-service pumps working in multiple and as separate units. I think this is occasioned by the variation in capacity of the pumps when working together on a common suction and discharge line. I have found it rather difficult to balance two centrifugal pumps on a common discharge, and pitot tube tests indicate in almost every case a considerable difference between the amounts of water handled by the individual units under these conditions.

3 I have in mind one installation on fire service, where the pumps were called upon to deliver against the maximum pressure for which they were designed and it was only with considerable difficulty that we were able to cut in additional units. I think that if venturi meters or pitot tubes had been placed on the discharge of each of the five pumps when they were working in multiple, a difference in capacity of the several units would have been shown, which would account for the difference in efficiency observed when the pumps were working individually and not in multiple.

George A. Orrok. At the time of the award of contract for these fire pumps, the New York Edison Company was obtaining proposals for centrifugal feed pumps—a somewhat similar service—and eight 1000-gal. 300-lb. pressure five-stage pumps were purchased. There was no attempt to obtain a high guarantee for efficiency, but the builders did state that under the above conditions an efficiency of 65 to 68 per cent would be obtained. These pumps were of the Jager type and under test showed an efficiency of about 68 per cent.

2 Fig. 5 shows that the high-pressure fire-service pumps are of the Kugel-Gelpke type and should be a trifle more efficient because of smaller friction and leakage. Seventy-one per cent seemed a very high efficiency and many doubts were expressed regarding the fulfillment of the guarantees. The extreme figure of 79 per cent obtained is probably the result of careful design and extra good shop

work and I believe has not been excelled. That this figure came as a surprise may be explained by the fact that most centrifugal pumps are stock pumps and not specially designed for the work they have to do. Pump manufacturers have been more concerned in getting a line of patterns that will suit standard conditions than in developing a line of pumps and system of patterns capable of doing the best work.

3 As a centrifugal pump is a mixed-flow or Francis reaction turbine reversed, similar care in design and construction would probably give efficiencies similar to those of the best makes of reaction turbines, which approximate 90 per cent.

Frederick Ray. The difference in efficiency of the units operated individually from that obtained when several were operated in parallel might be due to the different rates of flow through the venturi meters under the two conditions. With one pump operating, this flow would be low and the mercury column reading would be but slightly over an inch, so that with a given error of observation the percentage of error would be much greater than with two or three pumps discharging through the same meter.

2 Professor Carpenter here replying that the pipe connecting the two meters was open all the time, Mr. Ray continued:

3 This would equalize the flow in the meters, so that the mercury column reading when the whole station was running would be about 6¼ times the reading with one pump. It has not been my experience that parallel operation of a number of pumps has any tendency to decrease or otherwise change the efficiency obtained when operated individually. The efficiency should be the same, and in this case, as the pressures were taken at each pump, any losses in the piping system due to parallel operation would be external to the gages and would not show in the calculations. If the pressure had been taken at the discharge of the whole system, losses in the piping would affect the results.

4 Many pumps are running under similar conditions, at the efficiencies given. I have myself obtained efficiencies of 80 per cent and higher, but I do not rely as much on them as on some a little lower. I am now testing a 6-in., 2-stage underwriter pump, having a normal capacity of 500 gal. per min. against 100 lb. pressure, which has developed a maximum efficiency of 73 per cent.

5 I think the centrifugal pump is the ideal one for fire service, not only on account of its simplicity and reliability, but also on account of its characteristic increase in capacity as the pressure is

reduced. Thus, the 500-gal. underwriter pump referred to will discharge 870 gal. per min. at 60 lb., or enough for four streams at this pressure. It will give three streams at 90 lb., two streams at 110 lb. and one at 117 lb.—all at constant speed without any regulation whatever.

6 The City of Toronto has recently issued specifications for centrifugal pumps for their general municipal water supply, among which are several fire pumps capable of discharging against 300 lb. pressure. These pumps, however, are to be equipped with variable-speed induction motors, the pressure regulation being obtained by speed variation. This is superior to throttling regulation from the standpoint of current economy and in the case of the New York installation a considerable saving could be made by this means, as most of the fires can be handled with 200 lb. pressure or less.

H. Y. Haden. A somewhat unusual result obtained from this type of pump is that as the total head continues to increase beyond a certain point, the capacity falls off, with the result that the capacity curve, as given in Fig. 8, shows a backward tendency. It will be interesting to get the explanation of this.

2 There is unquestionably a large field in fire protection for steam-turbine-driven centrifugal pumps, and it is to be hoped that the Fire Underwriters will officially accept this type of fire protection unit. I believe that a properly designed centrifugal pump, for high speeds and of few stages, can be used to great advantage when direct-connected to high-speed turbines.

Thomas J. Gannon.[1] It was decided to use electricity as power for the pumping stations, because the first cost of installation, yearly cost of operation and maintenance and fixed charges were estimated to be lower, taking into account the intermittent service. The construction and operation of a steam plant were entirely out of consideration and the choice lay between gas-engine-driven and electric-driven pumps receiving power from outside sources.

2 It was estimated that gas operation of plants equal in capacity to the present electrically driven plants, would involve a fixed charge of $50,000 a year, in addition to the cost of the gas actually consumed. The question as to who should build and maintain

[1] Engineer, Dept. Water Supply, Electricity and Gas, Manhattan Borough New York.

the necessary large gas mains, the cost of which would approximate a million dollars, was not definitely settled. That the cost of a gas-engine-driven pumping plant would have been approximately double, both for machinery, building and area of land to be purchased, is borne out by the actual costs of the installations in Manhattan and at Coney Island.

3 The capacity of the gas-operated Coney Island plant is 4500 gal. of water per min. against a head of 150 lb. per sq. in. The combined capacity of the two pumping plants in the Borough of Manhattan, as originally laid out, was 30,000 gal. per min. against a head of 300 lb., with provision in each station for three additional pumping units of a capacity of 3000 gal. each, making a total combined capacity of 48,000 gal. per min. against 300 lb. pressure. On actual test, however, the capacity of the pumps was approximately 20 per cent greater than the designed capacity.

4 Furthermore, the flexibility of this type of pump permits of an increased discharge at lower pressures, which gives a capacity of approximately 5500 to 5600 gal. per min. for pressures between 150 and 200 lb., or a combined total capacity of 55,000 gal. per min. against 200 lb. pressure. This corresponds to the pressure at which the station is operated for most fires. In other words, the water horsepower of the electric-driven as compared with the gas-engine-driven plant is approximately in the ratio of 20 to 1.

5 The cost of the machinery in the Coney Island plant was approximately $37,000, and the cost of the building approximately $10,000. The cost of each of the two Manhattan pumping stations complete, exclusive of land, was practically $240,000. The first cost of installation of the gas-engine-driven plant is therefore more than double the first cost of installation of an equivalent electrically-driven plant, in the city of New York.

6 The high-pressure fire-service pumping stations went into official operation on July 6, 1908. It was at first decided to put the stations in service only when called on by the fire department, and up to and including November 20, 1908, the pumping stations were called upon to go into actual service for but 17 fires. On that date, the method of operation was amended so that the pumping stations are put in service in response to every alarm in the high-pressure district, and continue in operation awaiting instructions from the fire department. Under this system, from November 20 to December 31, 1908, the pumps responded to 116 first alarms. From the best available information, water was used in 55 instances, making a

total of 72 fires for which the high-pressure service had been used up to that date.

7 To insure readiness for service at all times, daily tests are made, of at least half an hour's duration, unless the station has been in actual operation during the preceding 24 hours.

8 During the first quarter of 1909 the number of alarms received was 239, and water was taken from the station for 125 actual fires. The total amount of water pumped was 17,840,000 gal., and 145,900 kw-hr. was consumed. It was on January 7, 8 and 9 of this quarter that the three large simultaneous fires mentioned in Par. 75, occurred, for which over 14,000,000 gal. of water was pumped, leaving about 3,800,000 gal. for the balance of actual fires occurring during the quarter. For these three simultaneous fires more than 81,000 kw-hr. was consumed while the total consumption of power for the quarter for all fires and testing purposes was but 145,900 kw-hr.

9 As to why a pump running singly develops a higher efficiency than when running in conjunction with several others, it is observed that pumps of the same type do not necessarily develop their best efficiency at the same speed and pressure. The pump running singly will naturally develop a pressure which corresponds to its own design, but when working in multiple, it will have to adjust itself to the common pressure.

10 As to reliability I have neither seen nor heard of any time when any one of the ten pumps installed in the Borough of Manhattan has failed to respond instantly when called on for service and to develop the full pressure on the system within one minute's time. At no time in service have the pumps shut down of their own accord.

Henry B. Machen.[1] Among the many difficulties encountered during the construction of the distribution system, perhaps the greatest was that due to the congested sub-surface of the street, which was a source of continual extra expense to the contractor, and of worry to the man in charge of selecting the location for the excavation of the trench.

2 The intersection of Sixth Avenue and Fourteenth Street may be cited as an example, since complete notes are available, due to the station excavation for the Hudson Tunnels. Here there were nine gas mains east and west, and nine north and south, belonging to

[1] Engineer, Dept. Water Supply, Electricity and Gas, Manhattan Borough, New York.

four different companies; two water mains in each direction; sewers and their connections on each side of the street; five Edison duct lines, and five duct lines with large manholes belonging to the Consolidated Telegraph and Electric Subway Company or the Empire City Subway Company; the conduits and banks of ducts of the Fourteenth Street and the Sixth Avenue trolleys; and lastly, the columns of the elevated railroad with their deep foundations.

3 Through this network the high-pressure main had to be so laid that the construction of the Sixth Avenue tunnel would not require it to be relaid. The excavation was carried on by tunneling, with here and there an opening through which the earth could be hoisted, using a pail let down by a rope. The pipe was lowered into the trench some distance up the street and pulled through, piece by piece, inspection of the running of the joint and caulking being almost impossible, since the space admitted but one man at a time after the pipe had been hauled in.

4 This condition existed at nearly all intersections of the main thoroughfares, such as Broadway, Sixth Avenue, Fifth Avenue, the Bowery, etc., and accounts for the high cost of laying the mains, averaging about $11 per ft. complete.

5 The second great difficulty encountered was in obtaining the prescribed test, which called for 450 lb. pressure per sq. in. to be held for 10 min., during which time the leakage was measured.

6 The system contained about 40,000 castings, 30,000 being straight pipe, tested at the foundry to 650 lb. The specials were not tested. All these castings, as already stated, were tested in the ground to 450 lb., the mains being under pressure in sections about one block long, between gates.

7 During the eighteen months the system has been in service, there have been but three breaks in the mains, all three in castings which had been subjected to the foundry test of 650 lb., two breaking at 150 lb. and the third at 300 lb. pressure.

8 To overcome the danger should a break occur during a fire, the proposed extensions to the distribution system now under contract, amounting to about $1,500,000, are laid out on what the department calls the duplex system. This method of overcoming the difficulty was first suggested by Mr. Blatt, assistant engineer of the high-pressure bureau. It consists of laying two entirely independent systems of mains and hydrants in alternate streets, the hydrants of one system being painted red and the other green. The mains are so laid out that at nearly all intersections of streets hydrants of both colors are available.

9 Should a break occur in either system, the operator at the pumping station would at once know in which system the trouble was located by looking at the venturi meters, and by throwing a switch he would start the closing of two electrically driven valves, separating one system from the other. Hydrants would then be available and in service pending the location and isolation of the damaged section.

10 The section now in operation was designed to give 20,000 gal. per min. on any one block with a loss due to friction from pumps to hydrant not to exceed 40 lb. The duplex extension will give the same results, and should either half be out of service by an accident, there will still be available at the same location 10,000 gal. per min., with a loss from the pumps to the hydrant in the most unfavorable location not exceeding 50 lb.

Richard H. Rice. This paper shows that the installation described was made after the most careful study and a very intelligent choice of the types of apparatus to be used. The choice of the centrifugal pump for the work described is thoroughly justified by its simplicity and by the efficiencies obtained. The centrifugal pump is today the popular means of producing pressure for emergency fire purposes, as in the fire boats of New York, Chicago, Duluth and San Francisco, and the new high-pressure service of San Francisco. In San Francisco twelve of these pumps are now being installed, four on fire boats and eight for an auxiliary fire installation. On the fire boats centrifugal pumps are particularly adaptable as they can be run in series or in parallel. In parallel they give 150 lb. pressure, and in series the pressure is doubled. This pressure is particularly valuable where walls have to be battered down, or streams thrown long distances.

2 The choice of alternating current as the source of power, in view of the unlimited supply of current existing and the duplicate means of conducting it into the station, is also justified. In cases where electricity is not so available as it is in New York, steam turbines are being installed, and they offer advantages over the gas engine, where maximum reliability is considered.

3 As an emergency installation pure and simple, I think the installation mentioned in the paper can be still further simplified. I believe the speeds chosen for operating the pumps are too low and that the pumps contain too many stages. I have had occasion to make extensive researches in centrifugal pump design with special

reference to operation at steam-turbine speeds, and have found that they can be operated at high speeds with a smaller number of stages, giving efficiencies comparable with those obtained here, although the question of efficiency is subsidiary to reliability for this service. Pumps for this service should be designed with two or three stages at the most, and with considerably higher speed.

4 Pumps can also be designed without balancing pistons, which are undesirable from the viewpoint of possible interruption of service. An inspection of Fig. 5, illustrating the construction of the pumps, will show that the balancing pistons used are quite liable to damage if water containing sand or other impurities is used, and this damage would very probably result in stoppage of the pump when it is badly needed. The use of balancing pistons is unnecessary in such emergency apparatus and should be avoided.

C. A. Hague. A question has been asked several times with reference to the results of tests of efficiency on centrifugal pumps operating singly and in multiple or group. Professor Carpenter has given the very plausible explanation that the difference in efficiency in favor of the pumps running singly is probably due to the presence of eddies and disturbances in the pipes when the pumps are operating together and the absence of such eddies and disturbances when only one pump is at work. In my experience in installing pumps and condensers singly and in groups I have found them extremely sensitive to each other in operation, both in taking in and discharging the water, when more than one pump is working on a line.

2 In the Manhattan stations, it seems to me that the suction or inlet pipes and the discharge pipes are coupled too closely for best efficiency; and also that the inlet pipe close to the pumps is not large enough for operation in multiple, although perhaps ample for a single pump when the water is undisturbed by the draft and discharge of several pumps. I have experimented considerably in that line, and have found that a comparatively large body of water next to the pumps on the suction side will materially ease the machines in their performance. The idea is to come up to the building with a normal supply pipe, and then enlarge it very considerably just where it enters the building, providing the inlet pipe with a good-sized air chamber wherever possible. I have tried this several times with excellent results.

3 Mr. Brown mentioned the difficulty of cutting in with a second

pump where the first pump was already running, a difficulty which I think is also due to too close connections along the inlet and outlet lines and a cramped condition generally. Of course, a disturbance in the water column and in the hydraulic horsepower would unbalance the electric power to a certain extent, perhaps not much, but the total disturbance may very easily result in the loss of several points in the efficiency.

4 Considering the fact that the city pays by the kilowatt-hour for its electric current as per switchboard reading, it would be no more than proper to state the efficiency of the machine as a whole, and not exclusively upon the basis of motor efficiency obtained in the shop of the makers a thousand miles or so away. In this case when 100 h.p. in current is supplied to the switchboard, the motor has shown an output by a competent test of 93.2 h.p. (Par. 37), the balance of 6.8 h.p., charged against the city in the power bills, being lost in heat and friction. Then, all that is charged against the pump is 93.2 h.p. The 67.57 h.p. shown by the pump for each 100 h.p. at the switchboard indicates only 67.57 per cent total efficiency, although the 67.57 h.p. indicates 72.5 per cent efficiency of the power delivered by the motor. I have tested several centrifugal pumping plants of various sizes and powers, and the total efficiency generally shows from 64.5 per cent to about 68 per cent and very seldom above the latter figure.

5 Mr. Bibbins touched upon the possibilities of utilizing the centrifugal pump for waterworks service, but upon investigation he would find a vast difference between emergency service, where operating economy counts for little in the face of great danger from fire, and the steady and necessarily economical service required for the continual pumping in waterworks stations. To show how deceptive a portion of the truth may be, a case is cited where a pumpage of a capacity of 10,000,000 gal. per day against 110 lb. load could easily be accomplished with displacement steam machinery by an expenditure of $10,000 per annum for coal. But an attempt to drive centrifugal pumps by electricity resulted in a cost for electrical power, at $6.50 per 1,000,000 gal., of $23,725 per annum; showing a difference in favor of displacement steam machinery equal to 5 per cent per annum on $275,940. There is no conceivable difference in cost of machinery, buildings, maintenance, attendance, or anything else, that would justify such a preference for electricity and centrifugal pumps over steam and displacement pumps. Note the following figures:

10,000,000 gal. daily, against 110 lb....................	440 pump-h.p.
120,000,000 steam duty with 8 lb. evaporation in the boilers, coal at $2.50 per net ton delivered.............	$9928 per annum
Electric power at $6.50 per 1,000,000 gal. against 110 lb. means 3,650,000,000 gal. per annum at $6.50..........	$23,725 per annum
The difference in cost for the element of power is $13,797 per annum, which at 5 per cent would capitalize at.....	$275,940

6 The steam-driven, reciprocating, displacement pumping engine can show a mechanical efficiency from the power put in through the throttle to the water-horsepower of the pumps, as high as 96 per cent, never as low as 90 per cent, under the above conditions. The centrifugal pump when steam-driven has a corresponding efficiency of about 65 per cent, and when electrically driven of about 67 per cent. A comparison of tests is given in Tables 1 and 2 in which it will be seen that the steam plant saves enough to pay 8.6 per cent on its entire cost.

TABLE 1 COST OF OWNING AND PUMPING WITH HIGHEST TYPE AND CLASS OF STEAM PUMPING MACHINERY

ONE UNIT, STEAM-DRIVEN, RECIPROCATING, DISPLACEMENT MACHINERY, CAPACITY OF 25,000,000 GAL. AGAINST 87 LB.

Pump horsepower..	870
Boiler horsepower for triple-expansion vertical pumping engine	450
Engine house and foundations and engine foundations...............	$150,000
Boiler house and foundation, boiler foundations, chimney, etc	
Vertical triple-expansion pumping engine..........................	
450 h.p. of boilers..	
Building for coal supply..	

CHARGES AGAINST PLANT—PUMPING ENGINE

Interest..	4 per cent
Sinking fund..	5 per cent
Depreciation..	2 per cent
Oil waste, etc...	1 per cent
Total...	12 per cent

CHARGES AGAINST PLANT—BOILERS

Interest..	4 per cent
Sinking fund..	5 per cent
Depreciation..	5 per cent
Total...	14 per cent

3 engineers. 6 firemen. 3 oilers.
Coal at $2.10 per net ton

Summary for Steam Reciprocating Machinery

Coal per annum	$11,957.40
Wages per annum	9,900.00
Capital charges on engine	13,920.00
Capital charges on boilers	1,260.00
Capital charges on buildings	1,548.00
Total charges per annum	$38,585.40
Cost per 1,000,000 gal	$4.11
Cost per horsepower	43.16

TABLE 2 COST OF OWNING AND PUMPING WITH HIGHEST TYPE ELECTRO-TURBINE PUMPING MACHINERY

One Unit, Electric-Driven, Centrifugal Machinery, Capacity 25,000,000 Gal. against 87 Lb.

Pump horsepower	870
Two-stage, electric-driven centrifugal pump	$43,750
Engine house and foundations and pump foundations	
Transformer house and foundations	
Transformers, lightning arresters, conductors, controllers and auxiliaries	

CHARGES AGAINST PLANT—PUMPING MACHINERY, ETC.

Interest	4 per cent
Sinking fund	5 per cent
Oil, waste, etc	1 per cent
Depreciation	2 per cent
Total	12 per cent

3 Engineers. 3 Extra men
Electric current, $4.50 per 1,000,000 gal.

Summary for Electro-Turbine Machinery

Electric current per annum	$41,062.50
Wages per annum	5,700.00
Capital charges on machinery	4,314.00
Capital charges on buildings	468.00
Total charges per annum	$51,544.50
Cost per 1,000,000 gal	$5.6[illegible]
Cost per horse power	59.2[illegible]

THOS. J. GANNON. In reply to Mr. Hague I will read the conditions which occurred on the evening of January 7, when both pumping stations were put to a crucial test:

7.22	First alarm, Hudson and Franklin Sts.
7.28	Second alarm, Hudson and Franklin Sts.
7.29	Third alarm, Hudson and Franklin Sts.
7.46	Fourth alarm, Hudson and Franklin Sts.
7.54	First alarm, Bowery and Hester Sts.
8.17	Automatic, Mercer and Houston Sts.
8.19	Second alarm, Bowery and Hester Sts.
8.29	Second alarm, Mercer and Houston Sts.
8.32	Third alarm, Bowery and Hester Sts.
8.40	Third alarm, Mercer and Houston Sts.
8.43	Fourth alarm, Mercer and Houston Sts.
8.45	Fifth alarm, Mercer and Houston Sts.

2 In due time seven pumps were put into operation, with a discharge which reached at times over 35,000 gal. per min., and it was estimated that over 52 fire streams were in service at the same time. Each pump responded instantly and remained in service until ordered shut down. The pressure was ordered gradually increased from 125 lb. to 230 lb., where it was maintained throughout the greater part of the time that the fires raged. The operating force at each pumping station consisted of but one engineman, one oiler, one telephone operator and one laborer.

PROF. GEORGE F. SEVER. A question was asked as to the feasibility of using the storage battery capacity to invert the rotaries and provide alternating current, to be spread through the alternating-current system to the sub-stations, and from those to provide alternating current to the pumping stations. In our preliminary investigation, if I recall the facts correctly, we were assured that this could be done; giving us another feature of reliability in the operation of the system. If the Waterside station should go out of business, we could still get current from the sub-station.

A. C. PAULSMEIER.[1] While the reasons given in the paper for the selection of electric-driven turbine pumps do not coincide with the conclusions as to reliability that have been reached in the West, there can be no question about the careful study given by the engineers who planned the high-pressure fire system described.

[1] Chief Engineer, Byron Jackson Iron Works, San Franciso, Cal.

2 The pumps show a remarkable efficiency, and one of the principal points that should commend them to those interested is their great flexibility as to capacity, a characteristic that every fire pump should possess.

3 The eight fire pumps now being built for the City of San Francisco are of a combined capacity of 216,000 gal. per min., under a working pressure of 300 lb. Each of these pumps is driven by a 750-h.p. Curtis steam turbine, operating at a normal speed of 1800 r.p.m.

4 In addition there are now being completed four fire pumps for the boats Dennis Sullivan and David Scannel, of an aggregate capacity of 9000 gal. per min. under 300 lb. working pressure, or 18,000 gal. per min. under 150 lb. working pressure, the pumps being so arranged that they work either in series or in parallel. The pumps have all been subjected to 24-hr. tests, and while the data on these tests are not sufficiently complete for publication, they show that the pumps are not as flexible as to capacity, or are not as capable of pumping an excess quantity of water, as are the Manhattan pumps. The reason for this is that the impellers in the San Francisco pumps are only 13$\frac{5}{8}$ in. in diameter, while the inlet to the impellers is less than 10 in. in diameter, this opening being further restricted by the pump shaft, so that it is impossible to obtain much excess water, no matter how much below the normal the discharge pressure is carried.

5 In the station pumps now being built the velocities at the entrance to the impellers have been somewhat decreased, although it is impossible to make anything like the excess capacity shown by the Manhattan pumps, which have impellers of such a size that the inlets may be made anything consistent with good practice.

Prof. W. B. Gregory. It is gratifying to know that efficiencies ranging from 70 to 80 per cent may be obtained with well-designed five-stage turbine pumps. The high-pressure fire-service pumps in New York represent one extreme of conditions, while at the other extreme is the centrifugal pump used in the rice irrigation territory of Louisiana and Texas for raising large quantities of water through comparatively small lifts.

2 The improvement in design of pumps of the latter class in the last ten years, and especially in the last five years, has made it possible to specify an efficiency of 75 per cent, even with heads as low as 10 ft. Purchasers of pumping plants in this section are no

longer satisfied with pumping outfits having efficiencies ranging from 50 to 60 per cent.

3 As examples of the results obtained with pumps of the class that deals with large volumes of water, the tables are quoted from recent acceptance tests conducted by the writer, of pumping plants used for rice irrigation.

TABLE 1 ACCEPTANCE TESTS

TANDEM-COMPOUND CONDENSING ENGINES, DIRECT-CONNECTED

Cane and Rice Belt Irrigating Company, Fulshear, Texas, August 12 and 14, 1908

WORTHINGTON PUMPS	FIRST LIFT	SECOND LIFT
Size of pump (diameter discharge pipe), in	45	45
Water pumped, gal. per min	47,620	46,430
Head on pump, ft	33.90	13.95
Efficiency of engine and pump, %	69.5	73.6
Efficiency of pump (engine 93 %)	74.7	79.2

CROSS-COMPOUND CONDENSING CORLISS ENGINE, DIRECT-CONNECTED

Sabine Canal Company, Vinton, La., May 22, 1909

WORTHINGTON PUMP	
Size of pump (diameter discharge pipe), in	45
Water pumped, gal. per min	44.010
Head on pump, ft	23.26
Efficiency of engine and pump, %	69.5
Efficiency of pump (engine 90 %)	77.3

TANDEM-COMPOUND CONDENSING CORLISS ENGINE, DIRECT-CONNECTED

Second Lift, Neches Canal, July 16, 1909

MORRIS MACHINE WORKS PUMP	
Size of pump (diameter of discharge pipe), in	48
Water pumped, gal. per min	60,300
Head on pump, ft	10.12
Efficiency of engine and pump (maximum), %	69.9
Efficiency of pump (engine efficiency 93.2 % max.)	75

CHARLES B. REARICK. Electrically driven fire pumping-stations for large cities are dependent upon current from an outside source, usually a large central power plant. It would seem quite practicable in many cases to locate new fire pumping stations adjacent to some large power plant having considerable boiler capacity. In such cases it would be possible to drive the centrifugal or turbine pumps with steam turbines, and thus eliminate the necessity of large overload capacity in electric generating units for the central station, and also the liability of derangement of the lines between the power stations and the pumping stations. The charge for standby service per annum should be less than for similar electric service.

2 The steam turbines have the advantage of being operative at any speed, and in this manner will maintain in the discharge mains any pressure desired. Furthermore, automatic regulating valves can be used in connection with the turbine to maintain constant pressure irrespective of demand or flow.

3 It is probable that the cost of installation would be less than for electric-driven units. The turbine could run non-condensing, as the question of steam consumption is of small moment for fire service.

Henry E. Longwell. The last paragraph of the paper furnishes a striking illustration of how purely academic is the ordinary official efficiency test, and of how little value as a basis on which to predicate the results that may be expected when the plant is operated under service conditions.

2 This paragraph gives general figures on the performance of the pumps during the fire run. There were 14,095,000 gal. pumped, with a current consumption of 81,450 kw-hr. The average net pressure against which the pumps operated is not stated, but assuming it was 300 lb. per sq. in., the pump efficiency, after allowing for the losses in the motor, would be only 40 per cent. However, we know that for part of the time the pressure did not exceed 225 lb., or, considering the pressure in the suction mains, about 200 lb. net. If the entire quantity of water had been pumped against this lower pressure, the efficiency would be well under 30 per cent. It is therefore perhaps fair to assume that the actual average efficiency was not far from 35 or 36 per cent, or say, in round numbers, only one-half that shown on the official test, when the load and other conditions of operation were more favorable.

W. M. Fleming. With the rapidly increasing size and height of office buildings, the annual fire loss in the business districts of the cities of the United States is increasing to an alarming extent. The installation of these tremendously effective fire-fighting systems has already proved of definite value in the reduction of city fire losses, and consequently of insurance costs.

2 What was probably the pioneer large and independent so-called high-pressure fire system in this country was installed at

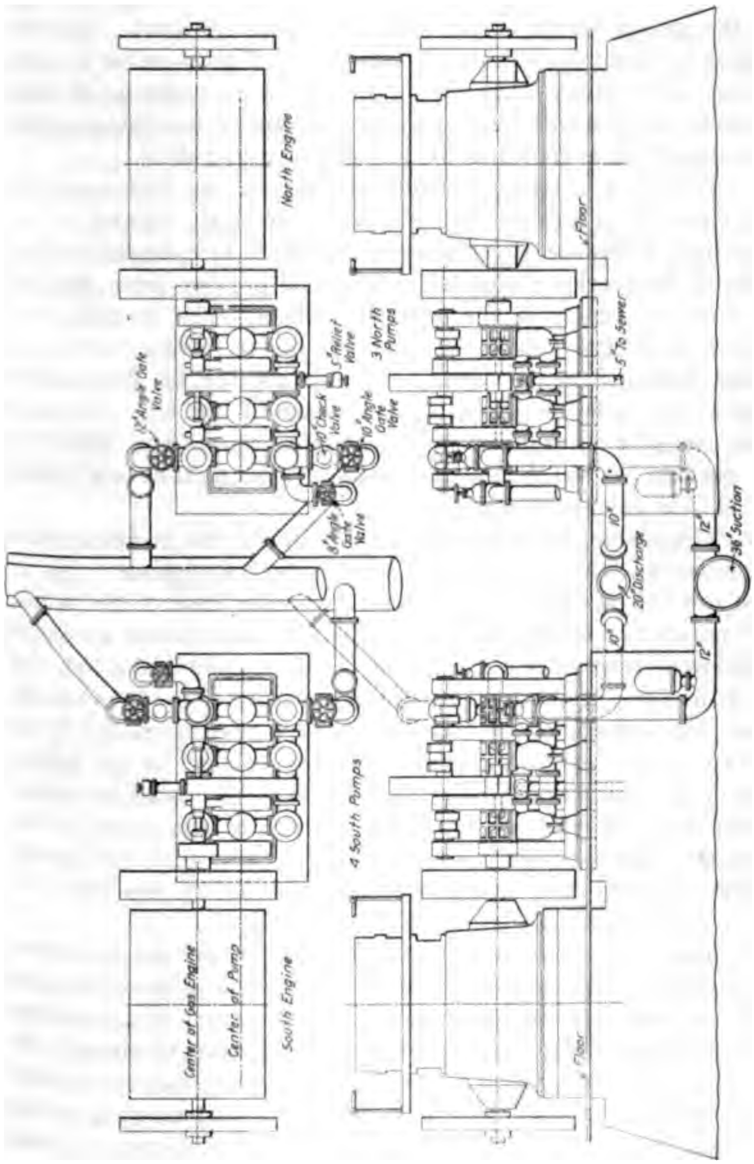

FIG. 1 GENERAL ARRANGEMENT OF THE PHILADELPHIA HIGH-PRESSURE FIRE-PUMPING STATION

Philadelphia in 1903-1904. This plant differs in almost every important detail from the New York system more recently installed; yet the general results in each case have been excellent. In Philadelphia the plant has so many times proved of great value in actual service that a much larger fire-fighting system, consisting of pumping units identical with those originally selected, is now being installed to protect what is known as the Kensington mill district.

3 From the original Philadelphia station at Delaware Ave. and Race St., a location unlikely to be seriously injured by conflagration, Delaware River water is supplied to independent high-pressure fire-service mains which effectually cover more than 425 acres at the center of the business district. The pumping units consist of vertical double-acting triplex power pumps built by the Deane Steam Pump Company, direct-connected to Westinghouse vertical 3-cylinder 4-cycle gas engines each of 280 h.p. The seven large pumping units have each a nominal capacity of 1200 U. S. gal. per min., at 300-lb. pressure, and two small units have a capacity of 350 U. S. gal. at the same pressure.

4 The general arrangement of the Philadelphia pumping station is similar to that of the large New York installations (Fig. 1). Two rows of pumping units occupy the main floor of the station. The pumps are nearest the center, and the gas engines are located in the same relative positions thereto as the motors in the New York pump houses. A platform extending along the sides of the building, about ten feet above the floor, serves as a working gallery for the operation of the engine throttles. Space is provided for the installation of three additional pumping units, and all mains are proportioned with the ultimate probable capacity of the plant in view. Suitable connections are provided to the mains so that the capacity of the pumping station may be supplemented by the use of the city's powerful fire boats, should occasion require.

5 The internal-combustion engines are of the well-known standard Westinghouse type and require little explanation. Speed regulation with varying loads is accomplished by the action of a centrifugal governor controlling the quantity of combustible admitted to the cylinders. Ignition is by a very neat type of make-and-break mechanism contained in a cylindrical plug. Two independent igniters are provided in each cylinder, and three independent sources of ignition current are available at all times. The engines are started by the use of compressed air, which is admitted to one of the cylinders at the proper time to secure rotation in the direction required until the

regular cycle of operation is established. The pumps are started under no-load.

6 The pumps are of the vertical, double-acting piston, triplex power type, requiring comparatively small floor space and giving a rate of discharge so smooth and uniform as to make imperceptible at the hose nozzles any pulsation in pressure.

7 In Fig. 2 is a sectional view of one of the pumps, indicating quite clearly the extreme simplicity and accessibility of the machine, and its general construction. All valves are of the poppet type, readily accessible through handhole openings. Valve areas and waterways naturally are comparatively large, so that friction losses

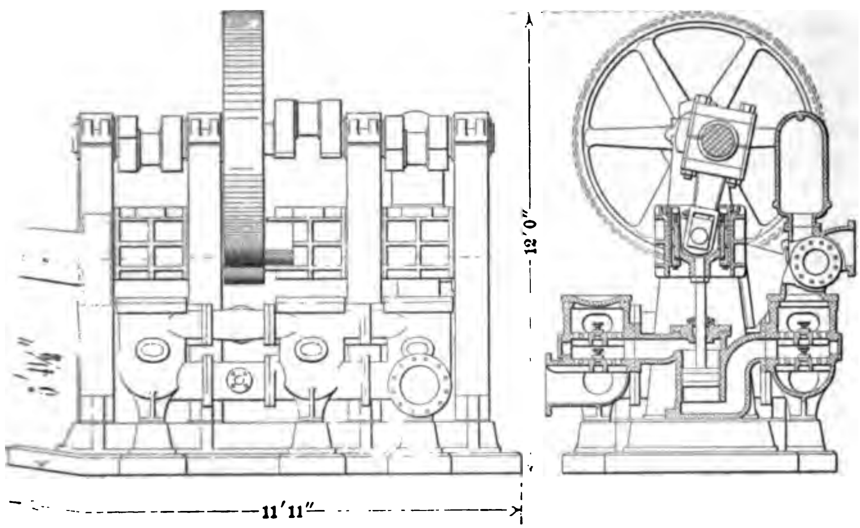

Fig. 2 Side and Sectional End Elevation of Triplex Pumps for the Philadelphia High-Pressure Fire-Pumping Station

are reduced to a minimum. The water ends are thoroughly brass-fitted in order that the pumps may be readily started after a long period of disuse.

8 There is a connection through a 12-in. check valve, from the city mains to the high-pressure system, so that the mains and pumps are constantly primed with a pressure of 60 lb. and are ready for service at all times. A complete system of fire-alarm boxes and telephones, with underground wires, permits direct communication between the vicinity of any fire and the pumping station. On the sounding of the alarm, the station force, consisting of an engineer and his assistant, can bring the total plant of seven large units

into service in seven minutes, and have repeatedly done so. Working pressure is invariably available at the hydrants one minute from the time of the alarm. Such a result would be impossible with ordinary movable apparatus.

9 The pumping units are started up under no-load, by the use of a motor-driven by-pass valve, through which the pump discharges into an overflow, until the normal cycle of operations has been set up in the gas engine, when the switch is closed, causing the by-pass valve to close and the discharge to be directed into the fire mains.

10 Experience has indicated that the maximum pressure of 300 lb. is required only for the most extensive fires, and for fires in the higher parts of tall buildings. The pressure records show that probably 75 per cent of the water pumped is required at not more than 150 lb. to 175 lb. pressure. The pressure desired in each case, is dictated over the telephone by the fire chief, the required pressure regulation being obtained by proportioning the number of units in operation to the requirements.

11 The practical results of the use of the Philadelphia fire system have been: material reduction in fire losses in the protected district, large decrease in fire insurance rates, and a greater willingness on the part of property owners in the protected section to erect pretentious office buildings.

12 Though the writer is unable to present a statement as to the annual saving to property owners by the installation, yet in view of the low cost of operation of the plant, there can be no question but that it presents a considerable yearly saving to the city. During the year 1907, which is perhaps typical, water was delivered to 16 fires, the longest one lasting 44 hr. The plant responded to 116 alarms at which no service was required. The operating expenses for the year were as follows:

Gas, 839,488 cu. ft. at $1.00	$839.49
Electric lighting	343.99
Electric power	7.98
65 tons pea coal at $3.50	227.50
Supplies furnished the pumping station for the entire year 1907	1,500.00
Total fixed charges for 1907	$2,918.96

SUMMARY

Salaries (Total for entire staff)	$8,389.72
Total cost materials	2,918.96
Total operating expenses	$11,308.68
Total daily maintenance charge, salaries and operation	$31.12

13 No mechanical defects have yet developed in either engines or pumps, and practically the only replacements have been a few rubber valves for the pumps and ignition details for the engines.

14 While no definite comparison can be made between the small plant in Philadelphia and the comparatively large plants in New York, which have not yet been in operation for an appreciable length of time, the operating expenses of the Philadelphia plant seem likely to prove much less for a given quantity of service. This is largely due to the so-called "readiness-to-serve" charge made by the company furnishing power to the New York plants. To this charge must, of course, be added the cost of the current actually consumed.

15 Unfortunately no mechanical efficiency test has ever been made on any of the Philadelphia pumping units. Judging from tests of similar machinery, an efficiency of 80 to 85 per cent is to be expected from pumps of this character operating against 150 to 200 lb. pressure. If this is the case, knowing that 75 to 80 per cent of the water to be used will be required at pressures not to exceed 175 lb., it would seem that the plant efficiency in Philadelphia would prove greater than in New York, where we understand that the water must be delivered through reducing valves from 300 lb. to any lower pressure required.

DISCUSSION AT ST. LOUIS

HORACE S. BAKER[1] presented some very complete notes on the proposed high-pressure system for Chicago, an abstract of which is given herewith. After telling of that city's need of a high-pressure system, Mr. Baker illustrated the effect of such an installation on insurance rates by citing the reductions brought about in other cities, details of which are given in Table 1, herewith.

2 The costs of maintaining and operating the proposed system for Chicago should not be more than the following figures, and probably much less:

Operating costs of three pumping stations, including interest and depreciation	$180,000
Interest on cost of distribution system, 4 per cent of $3,000,000	120,000
Depreciation of distribution system, 2 per cent of $3,000,000	60,000
Maintenance of distribution system	50,000
	$410,000

[1] Engineer, Department of Public Works, Chicago.

TABLE 1 HIGH-PRESSURE FIRE SERVICE IN THE UNITED STATES

System	Estimated Population	Date of Installation	Source of Pressure	Gal. per Min.	Max. Pressure, Lbs.	Lineal Ft. of Mains	Sizes of Mains, Ins.	No. of Hydrants	Total Cost of System	No. of Acres	Cost per Acre	Connection with Building	Effect on Insurance Rates
Atlantic City	40,000	Proposed	1 Station Elec. Turb. Pumps	7,000	225	38,590	8–14	82	$187,272	306	$612	....	....
Baltimore	575,000	Proposed	Pump. Sta.	......	......	75,900	10–20	.....	*397,999	360	.....	Standpipes on Buildings	....
Boston	620,000	1898	Fire Boat	6,000	200	4,700	12	14	30,080	65	463	....	....
Brooklyn	1,400,000	1906	2 Pump. Sta. Elec. Turb. Pumps	32,000	300	....	8–20	....	1,384,500	1420	975	....	....
Buffalo	420,000	1897	3 Fire Boats	......	300	12,756	12	.....	....	.....	.....	....	Reduction of $0.30 per $1000
Chicago	2,229,000	....	....	......	......	....	......	.....	....	.....	.....	....	....
Cleveland	480,000	Constructing	2 Fire Boats to Slave Pump. Sta.	10,000	300	32,524	8–20	96	*170,000	338	.....	May have Connection with Auto. Sprinklers.	Reduction of $0.80 per $1000 Prop.
Coney Island	....	1905–6	1 Station Gas Triplex Pumps	3,600	150	....	8–16	.....	90,000	147	612	....	Reduction of 25 %
Detroit	380,000	1893	2 Fire Boats	10,000	210	25,831	8–10	95	....	356	135	....	Probably has Prevented Increase
Fitchburg	33,000	....	Gravity†	......	180	28,250	8–16	.....	50,000	346	144	Boiler Feed, Elevators and Sprinklers	Prevented Increase
Hartford	98,000	Proposed	1 Station	10,000	300	53,430	8–24	198	796,277	731.3	1080	No open Connection	....
Lawrence	75,000	1906	Gravity †	......	134	10,200	10–12	39	....	120	.....	....	No change
Milwaukee	340,000	1880	3 Fire Boats	15,000	250	45,717	6–12	183	....	630	.....	....	10% Reduction

Newark...........	290,000	1905	Gravity	3,500	165	15,000	20–30	52	135,000	303	446	Some Connection	10% Reduction
New York.......... (Manhattan)	2,100,000	1908	2 Stations Elec. Turb. Pumps	30,000	300	....	12–24	1200	3,950,400	1430	2763	Water Curtains Provided for	No change
Philadelphia........	1,500,000	1903	1 Station Gas Triplex Pumps	9,100	300	35,300	8–16	166	700,000	512	1367	None on or in Buildings	Penalty of 25% Removed‡
Providence.........	200,000	1897	Gravity †	3,472	116	29,000	12–24	89	143,136	358	400	5 Automatic Sprinklers	No change
Rochester..........	185,000	1874	2 Stations Elec. Turb. Pump, Steam Turb. Turb. Pump	9,000	140	102,960	4–20	....	....	....	....	Some Connection	Graded Reduction
Toronto...........	215,000	Constructing	1 Station Elec. Turb. Pump	14,000	300	40,000	8–20	.....	500,000	287	1742	Considering Connection	Uncertain
Winnipeg..........	110,000	1908	1 Station Gas Producer Gas Eng., Tripl. Pumps	10,800	300	15,840	8–20	.....	650,000	275	2364	Connection with Auto. Sprinkler	Uncertain
Worcester..........	138,000	....	Gravity†	.....	165	100,320	8–30	.....	....	1380	.....	Elevators	No change

* Exclusive of pumping station and equipment.

† System consists of extension of pipes from high service into district covered by low service.

‡ Board of Fire Underwriters have voted to reduce rates to the amount of 10 cents per $1000 = a total of $40,000, if extensions costing $150,000 are made to the system.

NOTE:—This table is taken from the report on the proposed high-pressure fire system for Hartford, Conn.

TABLE 1 HIGH-PRESSURE FIRE SERVICE IN THE UNITED STATES

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Baltimore	575,000	Proposed	Pump. Sta.			75,900	10–20		*397,999	360		Standpipes on Buildings	
Boston	620,000	1898	Fire Boat	6,000	200	4,700	12	14	30,080	65	463		
Brooklyn	1,400,000	1906	2 Pump. Sta. Elec. Turb. Pumps	32,000	300		8–20		1,384,500	1420	975		
Buffalo	420,000	1897	3 Fire Boats		300	12,756	12						Reduction of $0.30 per $1000
Chicago	2,229,000												
Cleveland	480,000	Constructing	2 Fire Boats to Slave Pump. Sta.	10,000	300	32,524	8–20	96	*170,000	338		May have Connection with Auto. Sprinklers.	Reduction of $0.80 per $1000 Prop.
Coney Island		1905–6	1 Station Gas Triplex Pumps	3,600	150		8–16		90,000	147	612		Reduction of 25 %
Detroit	380,000	1893	2 Fire Boats	10,000	210	25,831	8–10	95		356	135		Probably has Prevented Increase
Fitchburg	33,000		Gravity†		180	28,250	8–16		50,000	346	144	Boiler Feed, Elevators and Sprinklers	Prevented Increase
Hartford	98,000	Proposed	1 Station	10,000	306	53,430	8–24	198	796,277	731.3	1089	No open Connection	
Lawrence	75,000	1906	Gravity †		134	10,200	10–12	39		120			No change
Milwaukee	340,000	1889	3 Fire Boats	15,000	250	45,717	6–12	183		630			10% Reduction

TABLE 3 STEAM TURBINE PUMPING STATION

Approximate Estimate of Cost

Capacity 10,000 gal. per min.; pressure 250 lb. per sq. in.

1	Excavation:		
	Pump pit	2300 cu. yd.	
	Boiler room	3865 " "	
	Stack	565 " "	
	Conveyor tunnel	70 " "	
		6800 cu. yd. at $1	$6,800
2	Concrete:		
	Retaining walls for pump pit	616 cu. yd.	
	Boiler room foundations	453 " "	
	Stack foundations	430 " "	
	Pump house foundation	101 " "	
		1600 cu. yd. at $7	11,200
3	Building:		
	Pump room, 60 ft. by 54 ft. =	3240 sq. ft.	
	Boiler room, 78 ft. by 84 ft. =	6552 " "	
		9792 sq. ft.	
	Assume 10,000 sq. ft. by 30 ft. = 300,000 cu. ft. at 15 cents		45,000
4	Foundations for pumps and turbines, 150 cu. yd. at $10		1,500
5	Four 2500-gal. centrifugal pumps at $5,000		20,000
6	Four 600-h.p. steam turbines at $12,000		48,000
7	Boilers, 2400 h.p. at $15		36,000
8	Chain grates, hoppers, conveyors, etc		15,000
9	Stack		8,000
10	Suction piping from city main and tunnel		6,500
11	Discharge piping		5,000
12	Steam piping		7,500
13	Condenser		6,200
14	Boiler auxiliaries, heater, purifier, pumps, etc		9,000
15	Two 20-in. venturi meters and recorders		3,000
			$228,700
	Add 15 per cent		34,305
			$263,005

Approximate Estimate of Operating Expense

1	Interest, 4 per cent of $263,005		$10,520
2	Depreciation, 4 per cent of $263,005		10,520
3	Coal:		
	200 hr., 5 tons at $2.50 } 8560 hr., ½ ton at 2.50 }		13,200
4	Oil, waste and supplies		1,500
5	Repairs		2,500
6	Labor:		
	Men, cost per annum three 8-hr. shifts:		
	1 engineer	$6600	
	1 oiler	4500	
	1 fireman	3000	
	2 coal passers	5100	
	1 janitor	700	
			20,200
	Total		$58,440

TABLE 2 COST DATA FOR VARIOUS TYPES OF APPARATUS

NAME OF SYSTEM	TYPE	PRESSURE AND CAPACITY PER MIN.	TOTAL COST EXCEPT DISTRIBUTION SYSTEM	ANNUAL OPERATING EXPENSE OF PUMPING STATIONS[1]	COST PER 1000 GAL. PER MIN. CAPACITY[1]	ANNUAL OPERATING COST PER 1000 GAL. PER MIN. CAPACITY[1]
Manhattan	Electric Centrifugal Pumps	300 lb. 30,000 gal.	$670,000	$139,250	$22,333	$4,642
Coney Island	Gas Engines Triplex Pumps	150 lb. 4,500 gal.	47,000	14,186	10,444	3,152
Philadelphia	Gas Engines Triplex Pumps	300 lb. 9,100 gal.	260,000	11,978	28,571	1,316
San Francisco Estimate 1	Steam Turbines Centrifugal Pumps and boiler Plant Oil Fuel	300 lb. 20,000 gal.	622,228	34,630	31,111	1,732
San Francisco Estimate 2	Gasolene Engines Turbine Pumps Rope Drive	300 lb. 20,000 gal.	737,848	30,595	36,892	1,529
Hartford Estimate 1	Steam Turbines Centrifugal Pumps Coal Fuel	300 lb. 12,600 gal.	257,620	45,320	20,466	3,597
Hartford Estimate 2	Gas Engines Triplex Pumps	300 lb. 12,600 gal.	377,905	8,648	29,992	686
Chicago, Estimate	Steam Turbines Centrifugal Pumps	250 lb. 10,000 gal.	263,005	37,400	26,300	3,740
Chicago, Estimate	Gas Engines Triplex Pumps	250 lb. 10,000 gal.	248,112	24,626	24,811	2,463
Chicago, Estimate	Electric Motors Centrifugal Pumps	250 lb. 10,000 gal.	122,882	57,700	12,288	5,770

[1] Exclusive of Interest and Depreciation.

3 In the light of current practice as shown in Table 2, it seems advisable to consider and estimate on the following types of pumping stations:

a Steam turbines and centrifugal pumps.

b Electric motors and centrifugal pumps.

c Gas engines and triplex pumps.

TABLE 3 STEAM TURBINE PUMPING STATION

Approximate Estimate of Cost

Capacity 10,000 gal. per min.; pressure 250 lb. per sq. in.

1	**Excavation:**	
	Pump pit .... 2300 cu. yd.	
	Boiler room .... 3865 " "	
	Stack .... 565 " "	
	Conveyor tunnel .... 70 " "	
	6800 cu. yd. at $1	$6,800
2	**Concrete:**	
	Retaining walls for pump pit .... 616 cu. yd.	
	Boiler room foundations .... 453 " "	
	Stack foundations .... 430 " "	
	Pump house foundation .... 101 " "	
	1600 cu. yd. at $7	11,200
3	**Building:**	
	Pump room, 60 ft. by 54 ft. = 3240 sq. ft.	
	Boiler room, 78 ft. by 84 ft. = 6552 " "	
	9792 sq. ft.	
	Assume 10,000 sq. ft. by 30 ft. = 300,000 cu. ft. at 15 cents	45,000
4	Foundations for pumps and turbines, 150 cu. yd. at $10	1,500
5	Four 2500-gal. centrifugal pumps at $5,000	20,000
6	Four 600-h.p. steam turbines at $12,000	48,000
7	Boilers, 2400 h.p. at $15	36,000
8	Chain grates, hoppers, conveyors, etc	15,000
9	Stack	8,000
10	Suction piping from city main and tunnel	6,500
11	Discharge piping	5,000
12	Steam piping	7,500
13	Condenser	6,200
14	Boiler auxiliaries, heater, purifier, pumps, etc	9,000
15	Two 20-in. venturi meters and recorders	3,000
		$228,700
	Add 15 per cent	34,305
		$263,005

Approximate Estimate of Operating Expense

1	Interest, 4 per cent of $263,005	$10,520
2	Depreciation, 4 per cent of $263,005	10,520
3	**Coal:**	
	200 hr., 5 tons at $2.50; 8560 hr., ½ ton at 2.50	13,200
4	Oil, waste and supplies	1,500
5	Repairs	2,500
6	**Labor:**	
	Men, cost per annum three 8-hr. shifts:	
	1 engineer $6600	
	1 oiler 4500	
	1 fireman 3000	
	2 coal passers 5400	
	1 janitor 700	
		20,200
	Total	$58,440

TABLE 6 ESTIMATED COST OF PROPOSED CHICAGO SYSTEM

MAINS, VALVES AND HYDRANTS

District No.	Cost
1	$477,508
2	329,321
3	152,018
4	128,457
5	109,178
6	314,569
7	82,791
8	178,420
9	146,432
10	118,916
11	113,268
12	85,852
13	75,918
14	175,811
Total	$2,488,459
Engineering and contingencies	373,269
	$2,861,728
4 stations at $250,000 =	1,000,000
	$3,861,728

No allowance made for land.

River crossings are assumed to be made as follows: (a) North branch in present Grand Ave. water pipe tunnel; (b) Main River in proposed LaSalle St. water pipe tunnel, to be built by Chicago Railways Company; (c) South branch in present Harrison St. water pipe tunnel.

4 For the purpose of estimate it seems proper to assume a station of a capacity of 10,000 gal. per min. against 250-lb. pressure, the working pressure to be probably 150 to 200 lb. To avoid the crippling of a station by the shutdown of any unit it seems advisable to consider units of 2500 gal.

5 In discussing the various types of installations proposed, Mr Baker cited the advantages of each type. The direct-acting duplex pumps are rugged and ready for immediate service, but their steam consumption is large. The independent boiler plant necessary, moreover, would be costly to build and to operate.

6 The gas-engine station has the advantage of lower first cost, and no cost for power when not in operation. Though failure of the [illegible] is unlikely, gasolene could be used with a change of

TABLE 5 ELECTRIC PUMPING STATION

Approximate Estimate of Cost

Capacity 10,000 gal. per min.; pressure 250 lb. per sq. in.

1	**Excavation:**		
	Pump pit	63,936 cu. ft.	
	Retaining wall footings	8,640 " "	
	Pump foundations	2,048 " "	
	Building wall	1,692 " "	
		76,316 " " = 2,826 cu. yd. at $1	$2,826
2	Concrete:		
	Wall of pump pit	15,264 cu. ft.	
	Footings	7,892 " "	
	Bldg. foundation wall	920 " "	
	Bldg. foundation footings	329 " "	
		24,405 cu. ft. = 904 cu. yd. at $7	6,328
3	Building:		
	Pump room, 36 ft. by 56 ft. =	2016 sq. ft.	
	Switch room, 16 ft. by 56 ft. =	896 sq. ft.	
		2912 or say 3000 sq. ft.	
	3000 sq. ft. by 30 ft. = 90,000 cu. ft. at 15 cents		13,500
4	Foundations for pumps and motors, 150 cu. yd. at $10		1,500
5	Four 2500-gal. centrifugal pumps at $5000		20,000
6	Four 600-h.p. 3-phase induction motors at $10,800		43,200
7	Suction piping from city main and tunnel		6,500
8	Discharge piping and valves in station		5,000
9	Switchboard and wiring in station		5,000
10	Two 20-in. Venturi meters and recorders		3,000
			$106,854
	Add 15 per cent		16,028
	Total		$122,882

Approximate Estimate of Operating Expense

1	Interest, 4 per cent of $122,882		$4,915
2	Depreciation, 4.3 per cent of $122,882		5,284
3	Power bill:		
	Ready-to-serve charge, $25 per kw. = $37,500		
	$0.005 per kw. per hr., 200 hr. of full load $1,500		39,000
4	Labor, 3 shifts:		
	3 engineers	$6600	
	6 asst. edgineers	9000	
	1 janitor	600	16,200
5	Miscellaneous: oil, supplies, etc		1,500
6	Repairs		1,000
			$67,899

3 The pumping capacity of the station at Bissell's Point will be over 100,000,000 gal. of water every twenty-four hours, which is more than twice the amount ordinarily consumed; the excess being sufficient to supply more than 30 fire-streams through 3-in. hose continuously, assuming 300-lb. pressure at the fire pumps.

4 The 5-stage centrifugal pumps proposed for the Chestnut St. station will have a capacity of 150,000 gal. per hr. each, against a pressure of 300 lb. per sq. in. It is proposed to connect the station with the power plants of the Union Electric Light and Power Company and the United Railways, so that two sources for power will be available.

5 The three discharge mains from these pumps will be 24 in. in diameter, the district supplied by them to be gridironed by a system of 12-in. mains laid on the enclosed streets and occasionally connected, at crossings only, by by-passes, that the breakdown of one main may not necessitate the cutting out of any other line. The pipe used will be cast iron, extra heavy, with bell and spigot joints, double-grooved. All fire-hydrant leads will be 8 in. in diameter.

6 The system will be under the ordinary distribution pressure when the fire pumps are not in use, so that for small fires the hydrants will be available for use; when the fire pressure is put on the system, the check valves on the by-passes will prevent additional pressure from coming on the distribution system.

7 While the arrangement of machinery for the pumping station, and the details of operation, have not been definitely decided upon, it is possible that gas engines may be used instead of electric motors. The questions of automatically starting and stopping the pumps, maintaining the pressure during a fire, and the general details of operation of the station, as well as the minor points of weight of pipe, design of hydrants, etc., have all to be worked out. It is estimated that the cost of this system will approximate $3,000,000.

H. C. Henley[1], speaking on the advantages of high-pressure fire systems, said that they were chiefly valuable for the numerous powerful streams which can be quickly brought into service and concentrated to advantage. For the prevention of conflagrations and for keeping serious fires from spreading, more powerful streams are needed than can be supplied by portable fire engines without considerable delay. To obtain such streams from fire engines, it is necessary to "siamese" two or more lines into one nozzle, requiring considerable time; and

[1] Chief Inspector, St. Louis Fire Prevention Bureau.

if a change in the location of engines becomes necessary, considerable time is again lost in re-assembling the hose lines.

2 The high-pressure system permits the use of hose of large diameter—3 in. and 3½ in.—and direct connection to hydrants furnishes a supply to nozzles of large area, without the necessity of siamesing two or more hose lines. The 2-in. nozzle is best adapted for use with high-pressure systems, this nozzle, under 75 lb. nozzle pressure, discharging approximately 1000 gal. per min. A nozzle of this area provides very effective service, as the loss of pressure, due to friction in fire hose, decreases as the area of the hose is increased. The data given in the table are derived from experiments by John R. Freeman, and show the pressure required at the hydrant in

PRESSURE REQUIRED AT HYDRANT TO OVERCOME FRICTION LOSS

Hose Diameter		2½ in.		3 in.		3½ in.	
Hose lines		Single	Siamesed	Single	Siamesed	Single	Siamesed
Smooth bore nozzle		1⅛ in.	2 in.	1⅛ in.	2 in.	1⅛ in.	2 in.
Length of hose line, ft.....	100	121	139	92	101	84.5	88
	150	139	170	99.5	113	87.5	93.5
	200	158	201	107	125	91	99
	250	176.5	232	114.5	137	94.5	104.5
	300	195	263	122	149	98	110
	400	232	325	137	173	105	121

For the 2-in. nozzle it is assumed that two hose lines of the length given are siamesed together.

hose streams of various lengths, to overcome friction loss and maintain 75-lb. nozzle pressure, the nozzle being at the same level as the hydrant.

3 High-pressure systems should be considered as auxiliary protection and there should be no attempt at abandonment of engines or other apparatus.

4 Direct connection from a high-pressure system to interior standpipes, sprinkler equipments and open sprinkler systems, should be made through siamese connections and not through direct pipe connection.

5 The inability of portable steam fire engines to furnish a stream efficient to cope with serious fires is made apparent by tests made by the engineers of the National Board of Fire Underwriters. The steam fire engines for test were picked at random from the equipment of many of the best city fire departments in the country.

Number of engines tested	102
Nominal capacity, gal	69,800
Actual capacity, gal	55,900
Percentage of efficiency	80

In many cases the efficiency of individual "steamers" is less than 50 per cent.

EDWARD FLAD. It appears to me that a cast-iron pipe is rather dangerous for high pressure. A cast-iron pipe tested under 300-lb. pressure will often break at 75 lb. A wrought steel pipe is much more reliable, and if properly coated, should last 25 or 30 years under ordinary conditions. If steel pipe is absolutely reliable we can afford to relay it at the end of 25 years rather than to use cast-iron pipe, which is liable to break.

2 In answer to a question by Mr. Flad as to the flexibility of the joint used in Baltimore, Professor Carpenter replied that it is flexible in the sense that it can be laid at an angle; it is not flexible so far as change of form is concerned.

H. S. BAKER asked what kind of steel pipe would be used in Baltimore, Professor Carpenter answering that it is extra heavy steel welded pipe, $\frac{7}{16}$-in. thick, the ends being expanded into semi-spheres an 8-in. or 12-in. pipe being expanded just enough to get a ring in it, and the whole bolted on the outside by external bolts, very like a steam pipe.

PROF. H. WADE HIBBARD. It is a fact that a cast-iron water main has been in satisfactory use in city service for twenty years and then a piece has blown out. It seems to me that the use of cast-iron pipe should be prohibited for this special emergency purpose of fire protection on account of its unreliability. In fact, in one of the high-pressure systems using cast-iron mains, leaks have been known to take place and the pumps to run for a considerable interval, some hours, I will say, and the pressure could not be maintained under test, until it was finally discovered that the water had been pouring out into a very large excavation and flooding it, unknown to those operating the pumping station. Steel will show approaching deterioration as cast iron will not.

2 Steel pipe ought to be good for thirty years of service. That period of service should be sufficient, and cities having such pipe

hould then be willing to replace it, having had more reliable protection during that period of years, than cast-iron pipe could possibly give.

H. C. HENLEY asked if there had been any attempt made to prevent the pipes from deteriorating through electrolysis, Professor Carpenter answering that the Baltimore system is a continuous metallic structure, from one end to the other, and he believed would be thoroughly protected from electrolysis; or at least, better than by any other system.

E. E. WALL. It is a fact that actually and not figuratively, steel pipe must be handled with gloves when it is laid, because the coating has to be very carefully preserved and can hardly be repaired if it is broken in handling before the pipe is laid. This is a very serious objection to the laying of steel pipe on account of exposure to corrosion after it is laid.

W. H. REEVES. Owing to the magnitude and prominence of these plants, the pump performances should be of interest to those desiring information on centrifugal and turbine pumping machinery. The foremost consideration in the art of building machinery of this class is accuracy in design. Without accuracy in design it is not possible to secure the maximum efficiencies within reach. A closely designed pump should deliver exactly its contract number of gallons against the contract pumping head, and the capacity should not run over nor under. From a pump builder's point of view the misfortune of falling short of the contract capacity needs no discussion here, but the other misfortune of running over on capacity may not be so clearly understood. One effect of running over is an overload on the motor, engine or steam turbine driving the pump, and another result is that the average efficiency of the equipment in daily operationuis below what it should be, for if it runs over in capacity its maximum efficiency does not occur at its contract capacity.

2 It will be noted that each of these pumps had a contract capacity of 3000 gal. per min., against a total head of 308.66 lb. per sq. in. Table 2 of the paper shows the performances of the five pumps at the South Street pumping station. This table does not show the averages, but it will be found that each pump averaged approximately 3761 gal. per min. against a mean total head of about 313.1 lb. per sq. in. Although the head was about 5 lb. above the contract condition, the pumps exceeded the contract capacity by about 25 per cent. This, no

doubt, caused the motor overload mentioned in Par. 64. The contract conditions implied 540 h.p. actually delivered, and at the guaranteed pump efficiency 770 b.h.p. would be needed. The delivered work under test was 686 h.p., and according to the test efficiency of 72½ per cent, 946 b.h.p. was used, that is, approximately 23 per cent excess motor load.

3 There appear to be no data on tests made on contract conditions. As the pumps were tested at a great excess in capacity it is quite probable that the efficiency would have been lowered several points if the pumps had been throttled to the agreed capacity and head. The tests as per Table 2 show about 686 h. p. delivered and 946 b.h.p., or a pump loss of 260 h.p. For a considerable range it is probably safe to assume this 260 h.p. loss to be fairly constant. Assuming this to be correct and adding this loss to the 540 h.p. delivered represented by the agreed contract conditions, would give 800 b.h.p., thus showing a pump efficiency of but 67½ per cent. If these pumps had been accurately designed, undoubtedly they would have shown as high efficiency under the contract conditions as was obtained with excess capacity condition.

PROF. E. L. OHLE. There seems to be quite a difference in opinion among engineers as to the reasons for the variation in efficiency of the pumps when working singly and in multiple. It seems to me that the reason is the one suggested by Professor Carpenter. It is practically impossible that all should work at the same speed, as they are independently driven. If then the pressure in the main should exceed the pressure which any pump was capable of delivering, the runner of that pump would simply revolve without delivering any water. This seems to be borne out by the experience of one pump builder, as stated by J. J. Brown.

THE AUTHOR. The discussion of the paper has been so voluminous that there is really but little needed from the author. In most of the discussion additional information of value has been contributed which I am sure will be appreciated by members of the Society.

2. The difficulties in connection with an installation of the kind described in the paper, involving a complete system of piping and hydrants capable of withstanding high pressures, as well as the necessary pumping machinery, are well brought out. I think the general conclusion will be that the piping difficulties to be overcome, especially when cast iron is employed, are very serious and require

special skill and the best of material. Attention has also been called to the fact that the city of Baltimore has adopted a system in which steel pipe is employed in order to overcome the difficulties due to the breakage of cast-iron pipe.

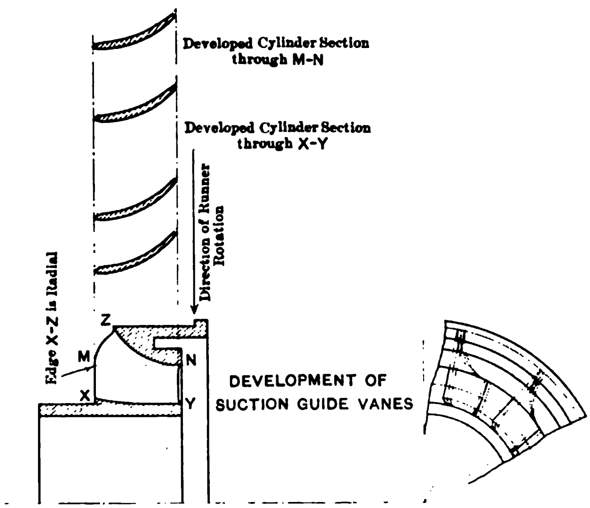
3 The discussion has disclosed the construction of several stations in which the motive power has been obtained from gas engines, and the advantages, disadvantages and expense of such installation.

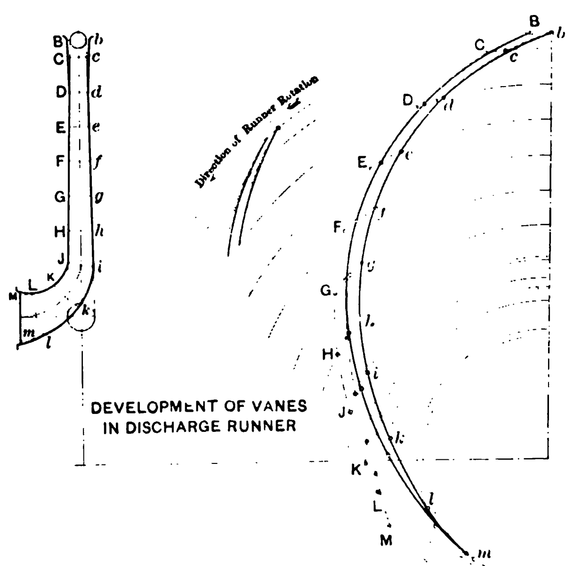
4 It is pointed out that although the centrifugal pumps are capable of operation at the high efficiencies shown by the paper, yet at the lower heads at which they are frequently operated the efficiency would be less. I do not believe there is any serious commercial disadvantage because of that fact, since it is true that the cost of operation of a fire station is principally due to other items than the cost of power. A fire station is required to be, above all things, reliable, and it is of very little importance whether or not the pumping be done under the most economical conditions for the reason that the total cost of pumping is only a small portion of the operating expense.

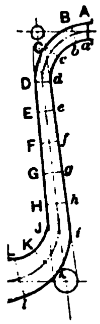
5 It is claimed by one of the discussors that the test should have been made by the city at the exact capacity called for and the efficiency should have been based on the result of such a test. This doubtless would have produced a lower efficiency than that obtained. In the light of the information now at hand, there would have been no injustice in such a requirement, but at the date of making the contract matters were different and such a requirement would have imposed a penalty on the builders, which would have been of no advantage to the city. The reason for that opinion is, that at the time of making the contract the information regarding multi-stage pumps operating at high heads was quite meagre. Mr. Sando, the designer of the pumps, secured all the data he could both in this country and in Europe. The result of his investigation led him to believe that it was to the advantage of the city and of the builders to put in a pump of such capacity that it would surely meet the requirements in that respect. It was believed that this would result in a considerable increased capacity over contract requirements. The motors were designed with an equally liberal capacity so that the machine was intended, even in the beginning, to be capable of a continuous large overload. The statement that the motors showed any evidence of being overloaded is in error, possibly because a certain remark which I made was misunderstood. It strikes me that the city is the principal gainer by such a system of design and that as a consequence it owns considerable more pumping capacity than was called for in the specifications, and so far as I know, without extra cost.

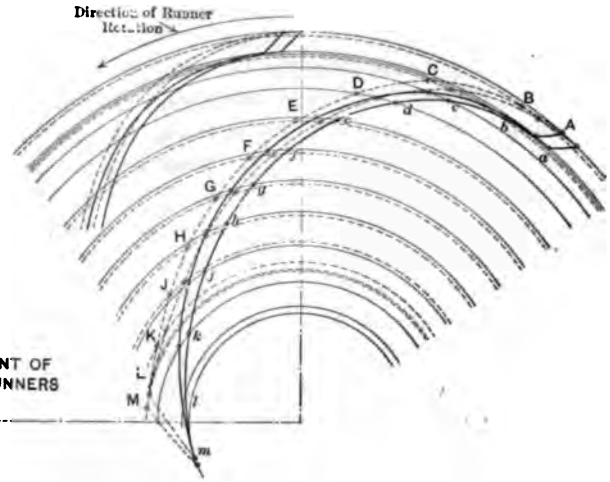
6 I believe that with the present data it would have been pos to design both pumps and motor to carry 25 per cent less load the same efficiency as was obtained by the larger pumps and mo In that case, a test at the specified capacity would have been a one.

[The following curves show the development of the runners, guide w and guide vanes of the pumps installed in the New York high pre pumping stations.—EDITOR.]

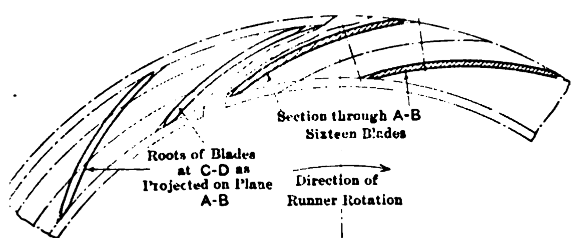


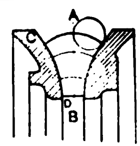




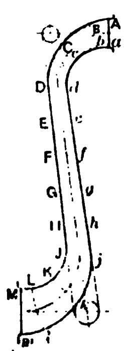


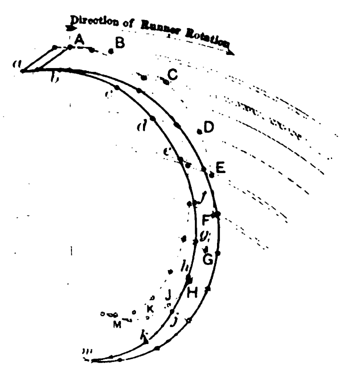
DEVELOPMENT OF VANES IN RUNNERS





DEVELOPMENT OF DISCHARGE GUIDE VANE





DEVELOPMENT OF VANES IN STATIONARY GUIDE WHEELS

7 The interesting question brought out by these tests regarding the higher efficiency obtained with a single pump as compared with all the pumps discharging into the main, has not been satisfactorily answered. Such results, however, seem to have been noted by every engineer who has made similar tests.

8 In Par. 65 of the paper I made one suggestion concerning this point. I have since thought that the variation in construction or in detailed shape of the discharge volume might possibly account for some of these differences. It is hardly possible that all the pumps can be made exactly alike and small inherent differences, which would be obliterated in the operation of all the pumps together, might account for the higher efficiency of the pumps operating singly. As suggested by Mr. White, the measurements were of a character which did not consider the pipe resistances, and the figures given apply to the delivery from the pump before the water was subjected to pipe resistances in any case.

No. 1250

STRESSES IN REINFORCED-CONCRETE BEAMS

COMPARISON OF EXPERIMENTAL RESULTS WITH RESULTS OBTAINED FROM THE USE OF THREE THEORIES OF DISTRIBUTION OF STRESSES

By Prof. Gaetano Lanza, Boston, Mass.
Member of the Society

Lawrence S. Smith,[1] Boston, Mass.
Non-Member

Many experiments have been performed on the breaking strength of reinforced-concrete beams, and in the course of them many observations have been made to determine quantitatively some of the phenomena attendant upon the application of the breaking load, and also upon that of smaller loads. Nevertheless it is well known that the observations made thus far are not sufficient to furnish the means for determining the actual distribution of the stresses, and hence for the deduction of reliable formulae for the computation of the direct stresses, shearing stresses, diagonal stresses, deflections, position of the neutral axis, etc., under a given load.

2 The test of the validity of such formulae should be their agreement with the results of experiments when the loads employed are about one-fourth or one-third the ultimate loads, because, when the loads are greater, the ratio of stress to strain varies very considerably for the different fibres, while for loads smaller than one-fourth of the ultimate, unknown initial stresses are liable to exert so great an influence as to interfere with the deductions.

3 The object of this paper is to make a comparison of (*a*) the position of the neutral axis, (*b*) the stress in the steel, (*c*) the stresses in the concrete, and (*d*) the deflection, as determined by experiment, with the same quantities as computed by three well-known theories of the

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distribution of the stresses. The comparison was made in the cases of eleven beams, in the testing of which the necessary observations were taken. Of the eleven, five were tested in the laboratory of Applied Mechanics of the Massachusetts Institute of Technology, and six in the laboratory of the University of Illinois.

4 The reinforcement consisted in each case of one or more longitudinal bars placed near the bottom of the beam, and equal loads were applied at the two points which divided the span into thirds.

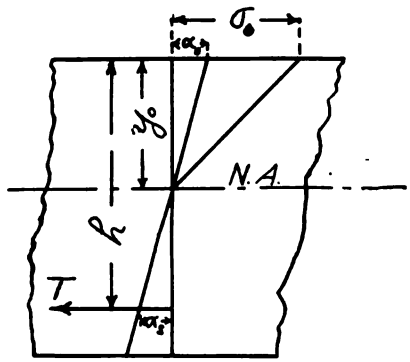

FIG. 1 THEORY A: DISTRIBUTION OF STRAINS AND STRESSES AT A CROSS-SECTION

5 The three theories employed in making the calculations, all of which assume that at any given section the strain in any fibre is proportional to the distance of the fibre from the neutral axis, will be denoted by A, B and C respectively, and may be described as follows, the notation used in lettering the figures being explained subsequently:

A This theory, which is very extensively employed, makes the assumption that at any given section none of the concrete below the neutral axis can be relied upon to resist tension; and further that the stress is proportional to the strain not only in the steel but also in the concrete.

6 This method is used by those who employ it to determine (*a*) the position of the neutral axis, (*b*) the stress in the steel, (*c*) the stress in the concrete, and sometimes the shearing stress at the neutral axis; but practically no attempt is made to compute the deflections by it. Nevertheless for the purpose of comparison, deflection formulae deduced on this basis will be given. The distribution of the strains and stresses at a cross section is shown in Fig. 1.

B This theory, which was proposed by Prof. A. N. Talbot, also makes the assumption that at any given section none of the concrete below the neutral axis can be relied upon to resist tension; but instead of assuming the proportionality of stress to strain in the concrete, the assumption is made that the stress at any fibre can be represented graphically by the corresponding abscissa of a parabola drawn through the neutral axis; the axis of the parabola being at right angles to the section, and its vertex at the end of the abscissa which would represent the crushing strength per square inch of the concrete, were the plot continued to such a height as to correspond to this crushing strength.

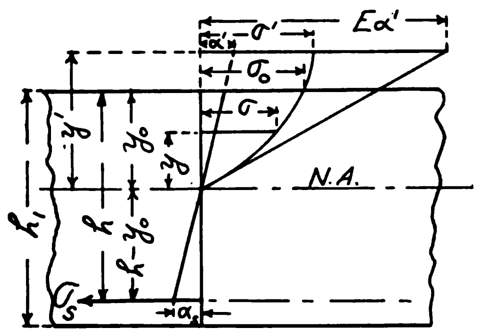

FIG. 2 THEORY B: DISTRIBUTION OF STRAINS AND STRESSES AT A CROSS-SECTION

7 The quantities calculated by this theory are the same as in case *A*, and again deflection formulae will be deduced on this basis for the same reason as there stated. The distribution of the strains and stresses at a cross section is shown in Fig. 2.

C The third theory is that proposed by M. Considère. He claims that whereas in a plain concrete beam, the concrete on the tension side cracks when the extension has reached 0.01 to 0.02 per cent, in a reinforced-concrete beam the concrete on the tension side can undergo many times this extension without cracking.

8 Among the tests which he cites in confirmation of this view is the following: He says that he subjected one reinforced-concrete beam to a load that produced in the lower fibre of the concrete an

elongation of 0.063 per cent as determined by measurement, and another such beam to a load that produced in the lower fibre of the concrete an elongation of 0.13 per cent, that he then removed the loads, chipped off the concrete below the reinforcement, and removed the reinforcing bars, after which he smoothed off the lower surface of the remaining portion of the beam and sawed out a concrete plank from the lower side. He says that not only did this plank not fall to pieces, but that on loading it transversely it bore as much as would be expected from a plain concrete plank of the same dimensions.

9 In view of the above, M. Considère suggests that the distribution of the stress at a section is as shown in Fig. 3, the compressive strength being represented by the triangle $OABO$, and the tensile strength in the concrete by the trapezoid $OCEFO$, the value of CD

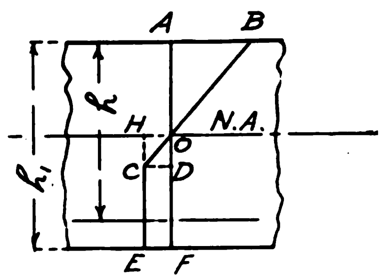

FIG. 3 THEORY C: DISTRIBUTION OF STRAINS AND STRESSES ACCORDING TO CONSIDÈRE

being equal to the yield point of the concrete in tension; and that for greater elongations the tensile stress does not increase.

10 However, inasmuch as the assumption of this distribution would lead to great complexity in the calculations, he proposes as a sufficiently close approximation that for the trapezoid $OCEFO$ in Fig. 3 we substitute the rectangle $OHEFO$. In this paper this approximation will be made in obtaining the formulae on the basis of Theory C.

11 Before obtaining the formulae needed for making the calculations, the notation used throughout will be explained.

Let α_0 = strain in concrete at upper fibre of beam.

α_s = strain in steel reinforcement.

E = ratio of stress to strain in concrete. In B this will denote the initial ratio of stress to strain.

E_s = ratio of stress to strain in steel.

$r = \frac{E_s}{E}$

σ_0 = compressive outside fibre stress per square inch in concrete.

σ_s = stress per square inch in steel.

a_s = area of section of steel reinforcement in square inches.

$T = \sigma_s a_s$.

b = breadth of beam.

$n = \frac{ra_s}{b}$

h = distance from top of beam to centre of reinforcement, inches.

h_1 = total depth of beam, inches.

y_0 = distance from top of beam to neutral axis.

ρ = radius of curvature of vertical longitudinal section of neutral layer.

W = total load applied.

M = bending moment at section at the center of the beam.

l = span, inches.

v = deflection at distance x from left-hand support.

v_0 = greatest deflection, i. e., deflection at middle.

The above is the notation needed for A.

12 In B the same notation is used with the following in addition:

Let y = distance of any fibre above the neutral axis.

y^1 = distance above neutral axis at which the fibre would be subjected to the crushing strength.

α = strain of fibre at distance y above neutral axis.

α^1 = ultimate compressive strain of concrete.

σ = stress in fibre at distance y above neutral axis.

σ^1 = ultimate compressive strength of concrete.

$q = \frac{\alpha_o}{\alpha^1}$

$n_1 = \frac{3\,r\,a_s}{b\,(3 - q)}$

d_1 = distance above neutral axis to point of application of resultant of compression.

d_2 = distance below top of beam to point of application of resultant of compression.

13 In C the same notation is used with the following in addition:

Let t = yield point of concrete in tension.

$d = h_1 - h$.

FORMULAE

14 Taking up the three theories successively, the formulae needed to make the computation of the values of y_0, σ_s, σ_0 and v_0 will now be given, the deduction being left to the reader.

Method A:

$$y_o = \sqrt{n^2 + 2\,n\,h} - n \quad \ldots\ldots (1)$$

$$\sigma_s = \frac{3\,M}{(3\,h - y_o)\,a_s} \quad \ldots\ldots (2)$$

$$\sigma_o = \frac{6\,M}{(3\,h - y_o)\,by_o} \quad \ldots\ldots (3)$$

$$v_o = -\frac{23\,Wl^3}{1296\,A} \quad \ldots\ldots (4)$$

where

$$A = E\left\{ra_s\,(h - y_o)^2 + \frac{b\,y_o^3}{3}\right\}$$

15 In order to find y_0, σ_s, and σ_0 by Method B, we have the equations below all of which include q.

Method B:

$$y_o = \sqrt{n_1^2 + 2\,n_1\,h} - n_1 \quad \ldots\ldots (5)$$

$$\sigma_s = \frac{M}{a_s\left\{h - y_o\,\dfrac{4-q}{4\,(3-q)}\right\}} \quad \ldots\ldots (6)$$

$$\sigma_o = \frac{M}{b\,y_o\left\{\dfrac{3-q}{3\,(2-q)}\,h - \dfrac{4-q}{12\,(2-q)}\,y_o\right\}} \quad (7)$$

Hence before we can find the values of y_0, σ_s and σ_0 we need to determine q, and this will have to be done approximately. For this purpose we can use the equation:

$$q^2 - 2q = -\frac{\sigma_o}{\sigma^1} \quad \ldots\ldots (8)$$

16 Plot a curve having $\dfrac{\sigma_0}{\sigma^1}$'s for abscissae, and q's for ordinates; then using for σ_0 a first approximation to its value, determine a first

approximation for q. Then determine a second approximation for σ_o and from it a second approximation for q, etc.

17 In the calculations made here, with the load approximately one-third of the breaking load, the value that has been employed is $q = 0.2$.

18 For the deflection we have

$$v_o = -\frac{23\,Wl^3}{1296\,A} \quad \ldots\ldots (9)$$

where

$$A = E\left\{ r\,a_s\,(h - y_o)^2 + b\,y_o^3\,\frac{8 - 3q}{24} \right\}$$

19 In order to obtain y_o by Method C we need to solve the equation of the fourth degree in y_o.

Method C:

$$b\,y_o^4 + 2\,\{ b\,(h + d) + 3\,r\,a_s \}\,y_o^3 - 3\,\{ b\,(h + d)^2$$

$$+ 6\,r\,a_s\,h\left\{ - 2\frac{M}{t} \right\} y_o^2 + 6\,r\,a_s\left\{ \frac{2\,M}{t\,b} + 3\,h^2 - d^2 \right\} y_o$$

$$- 6\,r\,a_s\,h\left\{ \frac{2\,M}{t\,b} + h^2 - d^2 \right\} = 0 \quad \ldots\ldots (10)$$

20 The solution can be readily effected graphically for any numerical case by writing u equal to the entire left-hand side of the equation, and plotting the resulting curve with y_o's as abscissae and u's as ordinates; then the value of y_o where this curve crosses the axis of abscissae will be the value of y_o desired. Of course the equation has four roots, but the one required can be easily identified as it must give a neutral axis that lies within the section.

21 In solving this equation, some value of t, the tensile yield point of the concrete, must be used. Considère suggests 170 lb. per sq. in. for the concrete used by him, which was about six months old and of a composition of nearly 1 – 2.5 – 2.5.

22 In the calculations made in this paper, $t = 100$ lb. per sq. in. has been used, as the concrete was from 30 to 60 days old and its composition was 1 – 3 – 6. After y_o has been found we can find σ_s and σ_o from the following equations respectively:

$$\sigma_s\,a_s\,\frac{3\,h - y_o}{3} = M - t\,b\,(h_1 - y)\,\frac{3\,h_1 + y_o}{6} \quad \ldots\ldots (11)$$

$$\sigma_o\,\frac{b\,y_o}{2}\,\frac{3\,h - y_o}{3} = M + t\,b\,(h_1 - y_o)\,\frac{2\,h - h_1 - y_o}{2} \quad \ldots\ldots (12)$$

or more easily from the formulae

$$\sigma_o = \frac{t\,b\,(h_1 - y_o)}{\frac{b\,y_o}{2} - r\,a_s\,\frac{h - y_o}{y_o}} \qquad (13)$$

$$\sigma_s = r\,\sigma_o\,\frac{h - y_o}{y_o} \qquad (14)$$

For the deflection we have

$$v_o = -\frac{23\,W\,l^3}{1296\,A} \qquad (15)$$

where

$$A = E\left\{\frac{r\,a_s\,(h - y_o)\,(2\,h - h_1 - y_o)}{2} + \frac{b\,y_o^2\,(3\,h_1 + y_o)}{12}\right\}$$

COMPARISON OF THE VALUES OF y_0, σ_s σ_0 AND v_0 AS COMPUTED BY THE VARIOUS THEORIES WITH THOSE DETERMINED BY EXPERIMENT

23 This comparison is exhibited in the tables. The first five beams were tested in the laboratory of applied mechanics of the Massachusetts Institute of Technology, and for these we have used $E_s = 28,000,000$ and $E = 2,335,000$, and hence $r = 12$. The last six beams were tested in the laboratory of the University of Illinois, and for these we have used $E_s = 30,000,000$, and $E = 2,000,000$, and hence $r = 15$. All eleven were made of $1 - 3 - 6$ concrete, the ages being given in the tables. All were loaded with two equal loads applied at points dividing the span into thirds.

TABLE 1 DETAILS OF REINFORCED-CONCRETE BEAMS

ALL BEAMS LOADED AT THIRD POINTS

DESIGNATION OF BEAM	AGE DAYS	b INCHES	h INCHES	h_1 INCHES	SPAN FEET	RODS Number	RODS Size Inches Round	STEEL a_s AREA IN SQUARE INCHES	p*
Massachusetts Institute of Technology									
A—1	53	8	10	12	11	1	1†	1.00	1.25
A—2	49	8	10	12	11	1	1‡	1.00	1.25
B—3	43	8	10	12	11	2	3/4†	1.125	1.41
C—5	35	8	10	12	11	4	1/2†	1.00	1.25
E—9	54	8	10	12	11	2	7/8‡	1.53	1.91
University of Illinois									
11	65	8	10	11	12	4	1/2	0.785	0.99
27–'04	63	12	12	13½	14	4	3/4†	2.25	1.56
28	60	8	10	11	12	4	3/4	1.77	2.22
33	60	8	10	11	12	3	3/4	1.325	1.66
35	60	8	10	11	12	{3, 2}	{1/2, 3/4}	1.473	1.84
45	61	8	10	11	12	{3, 2}	{1/2, 3/4}	1.473	1.84

* Reinforcement of area above center line of steel, per cent.

† Square.

‡ Twisted.

TABLE 2 DATA FROM TESTS ON REINFORCED-CONCRETE BEAMS

ALL BEAMS LOADED AT THIRD POINTS

DESIGNATION OF BEAM	BREAKING LOAD*	NEAREST ⅓ LOAD†	ACTUAL OR (PLOT) y_0 Inches	ACTUAL OR (PLOT) σ_0	ACTUAL OR (PLOT) σ_s	DEFLECTION Load Considered W	DEFLECTION v_0 (Plot)
Massachusetts Institute of Technology							
A—1	15000	5250	5.4	726	7941	4000	0.0731
A—2	16500	5250	5.3	650	7644	4000	0.0749
B—3	15950	5250	5.5	565	6615	4000	0.0660
C—5	16240	4600	4.6	781	8246	4000	0.1015
E—9	21000	6250	5.1	776	7563	5000	0.0769
University of Illinois							
11	11000	4000	4.8	740	11700	4000	0.175
27–'04	26900	9000	6.8	680	8250	9000	0.162
28	14300	5000	5.8	760	7800	5000	0.141
33	14400	5000	4.9	580	9000	5000	0.137
35	15000	5000	6.0	660	6750	5000	0.100
45	12400	4000	6.0	660	6750	4000	0.150

* Exclusive of weight of beam.

† Used in plots and in calculation for y_0, σ_s, σ_0

TABLE 3 RESULTS OBTAINED BY EXPERIMENT AND BY COMPUTATION

Massachusetts Institute of Technology Beams: $r = 12$, $E = 2{,}335{,}000$.
University of Illinois Beams: $r = 15$, $E = 2{,}000{,}000$.

Designation of Beam	y_0 Actual or Plot	y_0 A	y_0 B $q = 0.20$	y_0 C $t = 100$	σ_0 Actual or Plot	σ_0 A	σ_0 B $q = 0.20$	σ_0 C $t = 100$
				Massachusetts Institute of Technology				
A—1	5.4	4.18	4.29	4.94	726	801	760	759
A—2	5.3	4.18	4.29	4.94	650	801	760	759
B—3	5.5	4.36	4.47	5.10	565	774	734	640
C—5	4.6	4.18	4.29	5.06	781	703	666	653
E—9	5.1	4.86	4.97	5.45	776	844	802	810
				University of Illinois				
11	4.8	4.15	4.27	4.93	740	670	634	645
27-'04	6.8	5.88	6.01	6.60	680	711	678	690
28	5.8	5.49	5.59	5.98	760	669	638	662
33	4.9	5.00	5.10	5.54	580	722	686	717
35	6.0	5.17	5.28	5.71	660	701	667	675
45	6.0	5.17	5.28	5.83	660	561	534	548

TABLE 4 RESULTS OBTAINED BY EXPERIMENT AND BY COMPUTATION

Massachusetts Institute of Technology Beams: $r = 12$, $E = 2{,}335{,}000$.
University of Illinois Beams: $r = 15$, $E = 2{,}000{,}000$.

Designation of Beam	σ_s Actual or Plot	σ_s A	σ_s B $q = 0.20$	σ_s C $t = 100$	v_0 Actual or Plot	v_0 A	v_0 B $q = 0.20$	v_0 C $t = 100$
				Massachusetts Institute of Technology				
A—1	7941	13420	13510	9333	0.0731	0.1163	0.1193	0.0921
A—2	7644	13420	13510	9333	0.0749	0.1163	0.1193	0.0921
B—3	6615	12000	12100	7374	0.0660	0.1076	0.1105	0.0865
C—5	8246	11755	11840	7650	0.1015	0.1163	0.1193	0.0887
E—9	7563	10720	10810	8110	0.0769	0.1105	0.1139	0.0942
				University of Illinois				
11	11700	14190	14210	9950	0.175	0.1778	0.1824	0.1500
27–04	8250	11160	11240	8468	0.162	0.1822	0.1878	0.1573
28	7800	8296	8386	6672	0.141	0.1346	0.1400	0.1216
33	9000	10880	10960	8662	0.137	0.1592	0.1647	0.1406
35	6750	9842	9926	7609	0.100	0.1501	0.1549	0.1329
45	6750	7890	7941	5878	0.150	0.1201	0.1240	0.1033

TABLE 5 COMPARISON OF RESULTS

PER CENT OF VARIATION OF RESULTS ON TABLES 3 AND 4 FROM THE ACTUAL VALUES DETERMINED BY EXPERIMENT

Designation of Beam	y_0 A	y_0 B	y_0 C $t = 100$	σ_s A	σ_s B	σ_s C $t = 100$
			Massachusetts Institute of Technology			
A—1	−22.59	−20.56	−8.52	69.00	70.13	17.54
A—2	−21.12	−19.05	−6.67	75.60	76.74	22.09
B—3	−20.73	−18.73	−7.27	81.41	82.92	11.47
C−5	− 9.13	− 6.74	10.00	42.55	43.53	−7.23
E−9	− 4.71	− 2.55	6.86	41.74	42.93	7.24
			University of Illinois			
11	−13.54	−11.04	2.71	21.28	21.45	−14.96
27	−13.53	−11.61	−2.94	35.27	36.24	2.64
28	− 5.35	− 3.62	3.11	6.36	7.51	−14.45
33	2.04	4.09	13.06	20.83	21.78	− 3.76
35	−13.89	−12.00	−4.84	45.81	47.05	12.72
45	−13.84	−12.00	−2.84	16.89	17.65	−12.91
Average...	−12.40	−10.32	0.24	41.52	42.55	1.85

Values less than the actual are called negative.

TABLE 6 COMPARISON OF RESULTS

PER CENT OF VARIATION OF RESULTS ON TABLES 3 AND 4, FROM THE ACTUAL VALUES DETERMINED BY EXPERIMENT

Designation of Beam	σ_0 A	σ_0 B	σ_0 C $t = 100$	v_0 A	v_0 B	v_0 C $t - 100$
			Massachusetts Institute of Technology			
A—1	10.33	4.68	4.55	59.09	63.20	25.99
A—2	23.23	16.92	16.77	55.27	59.23	22.96
B—3	36.99	29.91	13.27	63.03	67.44	31.07
C—5	− 9.98	−14.72	−16.39	14.58	17.54	−12.61
E—9	8.76	3.35	4.38	43.70	48.11	22.50
			University of Illinois			
11	− 9.46	−14.34	−12.83	1.60	4.23	−14.29
27	4.56	− 0.29	1.47	12.47	15.92	− 2.90
28	−11.98	−16.05	−12.89	− 4.54	− 0.71	−13.76
33	18.28	18.28	23.62	16.20	20.22	2.63
35	6.21	1.06	2.27	50.01	54.90	32.90
45	−15.00	−19.10	−16.97	−19.93	−17.34	−31.15
Average...	6.19	0.88	0.66	26.50	30.25	5.76

Values less than the actual are called negative.

REMARKS AND CONCLUSIONS

24 The results seem to warrant the statement in Par. 1 that "the observations made thus far are not sufficient to furnish the means for determining the actual distribution of the stresses, and hence for the deduction of reliable formulae for the computation of the direct stresses, shearing stresses, diagonal stresses, deflections, position of the neutral axis, etc., under a given load." It follows therefore that whichever of the theories is adopted for practical use, it can be regarded only as a sort of working hypothesis.

25 It seemed therefore desirable to compare the results of these three well-known theories with those obtained by experiment. This comparison can best be made by a detailed study of the tables, especially Table 5 and Table 6.

26 However, it seems plain, as far as the evidence of these eleven tests goes, that in deducing the values of y_0 and σ_s theory *C* gives results much nearer those determined by experiment than either *A* or *B*, and the same is true to a lesser degree in the case of v_0, whereas the differences are not so marked in the case of σ_0.

27 It also seems hopeless to obtain a reliable deflection formula without taking into account the tension in the concrete.

28 Of course the computations are more easily made when *A* is used rather than *B* or *C*, but in the cases of *B* and *C* the complexity is not so great when designing a beam as when determining the stresses in a given beam under a given load.

DISCUSSION AT BOSTON

Chas. T. Main. All engineers, civil, mechanical or any other, want to know the most accurate way of figuring the stresses in reinforced concrete. What I am more anxious to know is that the proper ingredients are used, with proper mixing and good workmanship, so that we may be reasonably sure of a factor of safety in the finished work somewhere near what was intended. I have done no work of this sort without constant supervision, and am obliged to say that I have done no work that has been a source of pleasure to me. All of the building materials in common use are, I think, more certain in results than reinforced concrete. It is quite necessary to improve in the use of this material and in workmanship, in order to produce work which will inspire confidence.

Sanford E. Thompson. Professor Lanza's paper is of much value as a means of comparing the various formulae used in designing reinforced-concrete beams, with the behavior of test beams under load. Of the three theories the straight-line theory A is the simplest, and to the writer this still seems the best from a practical standpoint.

2 The formula derived by this theory as now used for determining the depth of a reinforced-concrete rectangular beam (using the notation adopted by the Joint Committee on Concrete and Reinforced Concrete[1]) may be expressed simply as

$$d = C\sqrt{\frac{M}{b}}$$

and the ratio of steel required is $A_s = pbd$

where d = depth of beam from compressed surface to center of steel, in inches.

C = a constant for a given steel and a given concrete.

M = moment of resistance or bending moment in general, in inch pounds.

b = breadth of beam, in inches.

A_s = area of cross-section of steel, in square inches.

p = ratio of cross-section of steel to cross-section of beam above the center of gravity of the steel.

3 Theory B, where the stress is taken as varying according to a parabola, is perhaps more exact than theory A, but at the same time more complicated and difficult in practical application. Theory C agrees more closely in the earlier stages of loading with the tests, although tests made both in the United States and in Europe indicate that Considère was not entirely correct in his assumption that steel when combined with concrete permits the concrete to stretch to a greater degree than when not reinforced. However, at earlier stages of loading the cracks in the concrete do not extend up to the neutral axis, so that more or less of the concrete is resisting tension and assists the steel in taking the stress. For this reason a method taking into account the tensile value of concrete gives results closer to the tests at early periods of loading than either formula A or B. There are, however, quite important reasons, as will be shown in succeeding paragraphs, why theory A is preferable.

[1]The Joint Committee is composed of representatives from the American Society of Civil Engineers, the American Society for Testing Materials, the American Railway and Maintenance-of-Way Association, and the Association of American Portland Cement Manufacturers.

4 Reinforced concrete is a complex material, which if properly used gives very safe and satisfactory structures. It is not, however, of a kind to which hair-splitting accuracy may be applied. In selecting a formula to use, the aim should be to choose one which will give results always on the safe side and at the same time not very wide of the mark. Referring to the paper, formula A gives results on the safe side, while C errs nearly as often on one side as on the other.

5 The behavior of a reinforced-concrete beam under load may be divided into two stages, the earlier stage where the concrete under the neutral axis bears tension, which gradually merges into the later stage, when the tensile strength of concrete is overcome and all the tensile stress is taken up by the steel. In the earlier stage the stress in steel increases proportionally to the moment, while in the later stage the increase in stress in steel is composed not only of the increase proportional to the moment, but also of the stress which in the previous stage was carried by the concrete and after its cracking transferred to the steel. Thus, for example, if a certain load W stresses the steel up to, say 16,000 lb. per sq. in., an addition to the load of *less* than W will double the stress. Therefore, a beam designed for a load which would produce an actual stress in steel of 16,000 lb. per sq. in. would have a factor of safety smaller than the ratio of that stress to the elastic limit of the steel. It is safer, then, to base the design on the results at the breaking load rather than on the results at earlier stages of loading, and to use theory A, which at the breaking load corresponds closely to the tests, and so be sure of the required factor of safety. In designing, working stresses and working moments should be used in the formulae.

6 The strongest argument against computing the concrete to bear tension, in practical design, is the fact that reinforced-concrete floors and other structures usually have to be built with joints between two days' work. The bond of the concrete on the joints is imperfect, and consequently the tensile strength of concrete at that point is small and cannot safely be counted upon in design.

7 Theory A is very simple and clear. It has been adopted quite generally in Germany and England, and I believe also in France, although that is the home of Considère, while the Joint Committee in this country has recently adopted it.

8 Theory A when used in figuring deflection does not give very satisfactory results, but this is not an important factor in reinforced-concrete design. When necessary to compute deflection, a more complicated formula may be used which considers the tensile strength of

concrete. The best of such formulae known to the writer are those derived by Professor Thullie of Austria, which are based on more logical assumptions than are the formulae of Considère.

9 It must not be forgotten that the computation of the stress in the middle of a supported beam is only one part of the theory of reinforced-concrete design. Just as important as the design of the beam in the center, since reinforced concrete is usually built continuous over several supports, is the design of the ends of the beam, and of no less importance is the part designed to resist the tendency of the diagonal tension to produce diagonal cracks.

10 It may be said then in conclusion, that although not corresponding strictly with tests, the ordinary straight-line theory is the one which will probably be used for some time to come because of its simplicity, and because reinforced-concrete beams, designed according to this theory, with due regard to other details, will produce, with good workmanship, structures which are unquestionably safe and conservative.

11 Except for a few isolated examples, it is less than ten years since reinforced-concrete buildings began to be erected; the 16-story Ingalls building in Cincinnati was built in 1903, and still stands as the most notable example of a concrete office building. And yet, as has been stated by Professor Burr, we already know more about concrete columns than about steel columns; the tests have been more exact, and more nearly conform to practical conditions. The beam theory is still in the stage of development, and tests and mathematical demonstration which tend toward more economical and rational detailing are welcome. Nevertheless, we may say with surety that buildings all over the country which are being designed by the common formulae with conservative stresses, and erected with proper care, are safe and conservative.

F. S. Hinds said that he had had a very profitable experience in the last two or three years in the construction of a large office building built entirely of reinforced concrete, erected for the Phelps Publishing Company at Springfield, Mass. The building covers an area of 30,000 sq. ft. and is eight stories above the sidewalk. In the construction of the building it was demonstrated that good work can be done with reinforced-concrete, and that there was no mistake in selecting concrete for both the interior and the exterior of the building.

2 His observations had led him to believe that this con-

struction in buildings even higher than eight stories will yet be seen. In fact, there is such an office building in Cincinnati, 16 stories above the sidewalk, showing that reinforced-concrete can be used in competition with the steel frame.

3 Answering a number of questions by Desmond FitzGerald, Mr. Hinds said that the concrete for the building was mixed by machine, crushed stone of "pea" size being used. The proportions of the mixture were 1-2-4, just enough water being added to make the mixture solid and yet make it flow easily. The ramming of columns was not done in the usual way, but the concrete was settled by means of four or five poles. Both round and twisted rods were used, held in place by small wood blocks which were withdrawn as the mixture was poured into the form.

4 Continuing, Mr. Hinds said that the great secret in concrete work is in getting the rods in the proper places. Supervision and careful preparation of the mixture and handling of materials will bring the best results. An oil paint and cold water paint without plastering have been used on the inside of the building, showing how smoothly the surface was finished.

5 In answer to a question Mr. Hinds said that moisture was prevented from going through the walls by their thickness—none being less than 8-in. thick—and by the density of the concrete. He had seen no cracks whatever in the reinforced-concrete proper, the only crack in the building being one near the top of the elevator-well partition, caused by expansion and contraction. Here and there a small crack appeared in the granolithic floor.

Prof. C. M. Spofford.[1] I presume we all agree with the previous speakers that concrete should be handled carefully, as it is subject to great variations. I feel, however, that merely to be careful is not enough; we should determine the theories as correctly as possible, and use them to eliminate so far as possible such uncertainties as now exist.

2 I am surprised that the C formula, as Professor Lanza has called it, gives results closer to those of actual experiment than the other formulae, and hope that the present data may be extended by further tests and computations. As far as actual use in design is concerned, any one of these theories may be safely used, provided a liberal factor of safety is employed, but further study and in-

[1] Massachusetts Institute of Technology.

vestigation along the lines indicated may enable us to determine more precisely what the factor of safety should be.

J. R. WORCESTER.[1] The careful study which the authors have devoted to these eleven beams is of great value, and their deductions show how much can be learned from a few experiments made with care and recorded with scientific accuracy.

2 It seems to the writer, however, that a few other points of interest in the tables are worthy of comment; as, for instance, the fact that in two of the beams, A-1 and A-2, alike so far as dimensions and amount of reinforcement are concerned, there appears to be a variation of 0.1 in. (1.9 per cent) in the actual location of the neutral axis; of 76 lb. per sq. in. (12 per cent) in the stress in the concrete; of 297 lb. per sq. in. (3.9 per cent) in the stress in the steel, and of 0.007 in. (10 per cent) in the deflection.

3 Another remarkable variation in the behavior of beams apparently alike is that of No. 35 and No. 45, where the latter with 80 per cent of the load of the former had the same actual deformations in steel and concrete, indicating the same location of neutral axis, and at the same time 50 per cent greater deflection. These great differences may perhaps be due to the fact that No. 45 was cracked before the test began, and therefore possibly should be excluded from such a comparison as this, though the cracking did not prevent the beam from developing fairly satisfactory strength. These striking instances of variation in observed results, where every precaution was taken to make the conditions identical, render it important to select theories of computation safe for the worst results found experimentally.

4 Speaking from a practical standpoint, several of the elements compared are not of vital importance. The location of the neutral axis is used only as an intermediate step in the process of calculation, and, if fairly correct results can still be obtained, error in this part of the calculation is not serious.

5 Then, again, the deflection is rarely of great importance. It is comforting to know that beams do not deflect as much as if the concrete had no tensile strength, but practically this is as far as we are usually concerned.

6 The actual compressive stress in the concrete may also be eliminated from consideration in actual construction, if only we can limit the area of steel to such a percentage that we are sure failure

[1] J. R. Worcester, 79 Milk St., Boston, Mass.

from the compression of the concrete will not occur until the steel has been stretched beyond the elastic limit. In this connection it is worthy of note that the beams quoted were with one exception more heavily reinforced than is usual at the present time. With 0.8 per cent of steel, or even with 1 per cent, it is safe to base our calculations for moment upon the stress in the steel only.

7 The element then about which the most interest centers is the stress in the steel, and it is important that we should adopt a method of computation which gives this with the least error practicable, and with that on the safe side.

8 Looking at Table 5 with these considerations in mind, we find little difference between methods *A* and *B*, both giving results well on the safe side. Method *C*, while averaging very closely to actual results, gives errors on the wrong side in five out of the eleven cases cited; in one case, and that the one most resembling usual practice, having an error of nearly 15 per cent on the unsafe side.

9 It is noticeable also that the loads assumed are considerably less than what would usually be considered working loads for the beams in question. Following almost universal practice at the present time, the stress in the steel as computed would be allowed to go to 16,000 lb. per sq. in. This would permit loads on the University of Illinois beams as follows:

No. 11, 5,000 lb in place of 4000 lb.
No. 27, 12,000 lb. in place of 9000 lb.
No. 28, 10,000 lb. in place of 5000 lb.
No. 33, 7,000 lb. in place of 5000 lb.
No. 35, 8,000 lb. in place of 5000 lb.
No. 45, 8,000 lb. in place of 4000 lb.

Only these six are quoted because the essential facts regarding them are given in the bulletins of the University of Illinois, while we have not at hand the details of the tests at the Massachusetts Institute of Technology.

10 The diagrams of these beams indicate under the above loads the stresses in the steel indicated herewith in Table 1, using the authors' modulus of elasticity, 30,000,000 lb. In the same table are given the stresses in the steel as calculated by methods *A* and *C*, and the percentage of error by each method.

11 Comparing these results with those obtained by the authors as shown in Table 5, we find that the common method of computation, *A*, gives considerably closer results to those observed than under the lower loading. The error ranges from 5 to 20.6 per cent, with an

TABLE 1—STEEL STRESS UNDER HEAVIER LOADING

Beam No.	Load Used	Stress in Steel, Lb. per Sq. In.			Error of Calculation Per Cent	
		Actual	By A	By C	By A	By C
11	5,000	15,600	17,700	13,600	+13.5	−12.9
27	12,000	13,500	14,900	12,600	+10.4	− 6.7
28	10,000	14,700	16,600	15,000	+12.9	+ 2.0
33	7,000	12,600	15,200	12,900	+20.6	+ 2.4
35	8,000	13,800	15,750	13,900	+14.1	+ 0.7
45	8,000	15,000	15,750	13,900	+ 5.0	− 7.3
				Average error	+12.75	− 3.6

average of 12¾ per cent, always on the safe side. On the other hand, by the Considère method, C varies from $+ 2.4$ per cent to $- 12.9$ per cent, with an average of 3.6 per cent on the unsafe side. This would indicate that there is no advantage in adopting the more laborious method, involving the solution of an equation of the fourth degree, at least so far as proportioning the steel is concerned.

12 The chief difference between the two methods, as explained in the paper, is in the assumption in the Considère method of a certain value for tension in the concrete below the neutral axis, and the disregard of this in method A. There is no question that under ordinary conditions the concrete has a small amount of tensile strength while the loads are small, but there is grave doubt as to the safety of relying upon a crystalline material under such conditions. Many conditions in actual construction may tend to destroy the tensile strength. There may be set-joints near the center of the beam; there may be voids near the bottom where the mortar has leaked out; there may be incipient invisible cracks extending to an unknown distance. It is a fortunate circumstance that ease of calculation is on the side of the safer method, for this is a powerful incentive to its adoption.

13 The statement at the opening and close of the paper that "the observations made thus far are not sufficient to furnish means for determining the actual distribution of the stresses," etc., is undoubtedly true, speaking literally and with scientific accuracy. At the same time it should be borne in mind that we are dealing with a crude product which cannot in practice be made with scientific accuracy. It is doubtful whether absolute knowledge of the laws of distribution

of stress in a theoretically perfect material would be of any great advantage in designing structures of every-day material. The important question is whether we know enough to design our beams with entire safety and reasonable economy. To this query the writer would unhesitatingly give an affirmative answer. The investigation of these beams tends to confirm this opinion, which is also supported by the constantly accumulating experience with actual construction. We would therefore venture to add two other conclusions to those advanced by the author, namely:

a Experiments indicate that, though precise determination of the laws of stress distribution may be impossible in the present state of our knowledge, sufficiently close approximations may be made for all practical purposes.

b The simple method of calculation, by neglecting tension in the concrete and assuming a straight-line distribution of the compressive stress, is the easiest to apply and gives satisfactory results for the determination of the stress in the steel.

PROF. GEO. F. SWAIN. I notice that Professor Lanza has used a value of $E = 2{,}335{,}000$ for the beams tested at the Massachusetts Institute of Technology, while for the beams tested at the University of Illinois he has used a value for E of 2,000,000. The beams tested at the Massachusetts Institute of Technology were from 35 to 54 days old, while the beams tested at the University of Illinois were from 60 to 65 days old. The modulus of elasticity ought to increase with age, other things being equal, yet in these tables the reverse is assumed. This fact might account for some of the peculiarities and the results. Professor Lanza does not state whether he measured the modulus of elasticity.

2 In Table 2, I think the heading of the column "Nearest one-third Load," is a little confusing. Those figures are not very close to one-third the load, and beam *C*-5, which has a larger load than the first three beams, has a smaller value in the third column. I suppose the third column simply means the loads for which computations were made, and that the loads were applied in such increments that the figures given represent the nearest third of the load for which computations were made. Yet it seems rather confusing that for a load of 16,240 lb., the nearest one-third should be given as 4600 lb.

3 With reference to the three theories, I have never believed in Considère's main contention, namely, that by reinforcing concrete such great strains could be produced without fracture; though his explanation is in a certain degree plausible. If a body is stretched so that the molecules are a certain distance apart, nothing can prevent fracture. Ductile material like steel draws down at the point of fracture and is stretched much more there than on the average through the length of the piece. If concrete were a ductile material, its adhesion to the steel bars might prevent any such phenomena as drawing down and thus distribute the strain; but concrete is not a ductile material, and there seems to my mind to be no possibility of the great stretch without fracture which Considère claims.

4 As to the results obtained by the three formulae, I think those given in the tables were precisely what might be expected, because these loads were only large enough to be called working loads; that is, they were nothing like the ultimate load. As a matter of fact there was tension in the concrete, under which condition the steel would be relieved; we would therefore expect that in case C the stress in the steel would be very much less than in the other two cases. In practice, also, there is undoubtedly tension in the concrete unless cracks occur. The results of tests made by the Boston Transit Commission show large tensile stresses in concrete beams without reinforcement.

5 However, the question is what to do in designing. In practice there may be cracks in the concrete, not due to stress, but to the moving of blocks on which the rods are set, making the cement run out, or due to shrinkage or joints or other causes; for which reason it seems to me that in practical designing, engineers are not justified in assuming any tension in the concrete.

Henry F. Bryant.[1] Mr. Worcester stated (Par. 9) that on applying his reasoning to the University of Illinois experiments, the nearest one-third load for 16,000 lb. of stress on the steel would be found to be nearly double that given in the paper as approximately one-third the breaking load. This emphasizes the question of the yield point. The rather common practice, as Mr. Worcester states, is to take from 12,000 lb. to 16,000 lb. on mild steel and with this to use about 500 lb. as the concrete compressive strength, which, with concrete of 2000-lb. compressive strength, gives a factor of safety of four or possibly five. If the yield point is the critical point in the steel, we are using a factor of safety of only between two and three in the steel. Mr.

[1]Engineer, 334 Washington St., Boston, and Brookline, Mass.

Worcester's analysis of the Illinois experiments would indicate that instead of breaking at three times what would be considered a safe working load, the beam would break at not over twice the load. I think that using mild steel and a factor of at least four, and figuring that the yield point is the critical point of the steel, we should apply to the steel something like 7500 lb. or 8000 lb., with 500-lb. compression on the concrete. That means a little larger percentage of steel than is common practice, though it is not unusual to adopt this reasoning with high-carbon steel. I am very glad to see that these experiments point that way.

H. E. SAWTELL.[1] Considère's theory of stress distribution agrees very well with the actual tests at about working loads on the eleven beams mentioned in the paper. We know, however, that his theory will not agree with breaking-load results as well as either the straight-line or the parabolic theory, each of which considers that concrete takes no tension stress. We should adopt a theory which will agree quite closely with tests at breaking loads, but which will always be on the safe side for intermediate loads. We can then get a real factor of safety.

2 Referring to Par. 24, it seems likely that when applied to floor beams, a formula will remain only a sort of working hypothesis if our theories are to be based upon test beams which are not more like the beams used in actual practice, and if our compressive value for concrete is based upon plain concrete. The present uncertainty may appear to favor the side of safety, but on the other hand, when too many assumptions have to be made, there is little real satisfaction in working with the material.

3 Tests on rectangular beams are necessary for determining as nearly as possible the stresses and deflections in slabs and separately molded beams, but do not seem to solve the problems of beams and girders as used in actual construction. Let us first note some of the stresses as they exist in a beam in actual construction, assuming Fig. 1 to be the cross section of a beam at its place of maximum flexural stress. The slab steel is placed at the beam, as a great many designers consider necessary, in order to resist fully and reliably the negative slab stress, etc., at the beam. These slab rods always are only a few inches apart, and pass through the top of the beam concrete at right angles to the compressive stress of the beam.

[1] Structural engineer, with Chas. T. Main, Boston, Mass.

4 Assuming that the concrete in both beam and slab is poured at the same time, we know of course that for some distance each way from the beam the slab will work with the beam in resisting compressive stresses. Assumptions are made as to what part of the slabs will work safely with the beam, and then the beam is calculated for and designed as a T-beam. In doing this the full working stress for concrete in compression is used. The concrete at *G*, *E* and *F* has a large share of the compression to take care of. Also, as a result of placing the slab steel at the top, as it passes over the beam, the concrete at *G*, *E* and *F* is again put in compression, this time at its full working value, but at right angles to the compressive stress in the beam.

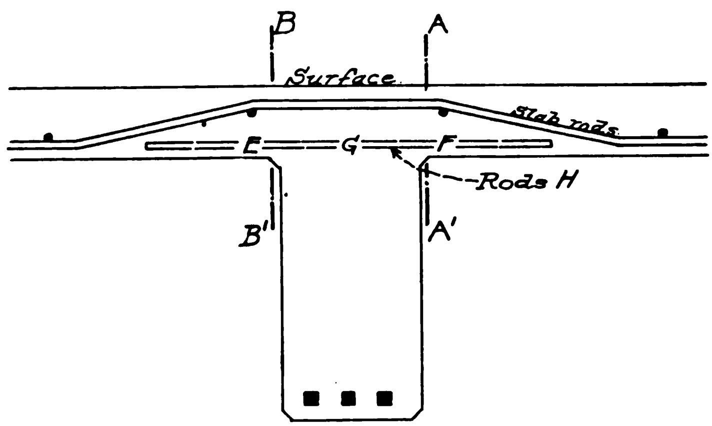

FIG. 1 CROSS-SECTION OF BEAM AT MAXIMUM FLEXURAL STRESS

5 Again, the maximum vertical shear in the slabs is along the lines *BB′* and *AA′*, this shear, it will be noted, being through concrete already doing double duty in compression. The concrete at the surface is at the place of maximum compressive stress of the beam and it also has a maximum tensile stress due to the negative slab moment.

6 The total compression at *G*, *E* or *F* is very much higher than we would willingly put upon plain concrete as a working stress, while the concrete at points *E* or *F* is in a worse condition. At the surface the material is nearly cracking from a tensile stress, even under working loads, and it cannot be of much service in compression where it is most needed by the beam.

7 If these conditions are correctly noted, and if the actual stresses

are to be kept down to the unit working stress of *plain* concrete, then it will be necessary either to assume a much lower unit stress for concrete when designing T-beams, or to design a rectangular beam whose effective top surface does not extend above the slab rods shown in Fig. 1. But is it necessary to use the value of plain concrete when designing T-beams? Are we not justified in saying that concrete at *G* is confined, and being reinforced, has a much higher ultimate strength than plain concrete?

8 The compressive strength of concrete in beams is increased in two ways (*a*) by lateral restraint, brought about by the surrounding compressive forces; (*b*) by reinforcing its shearing resistance, which may be greatly assisted by placing the rods *H* at the points shown in Fig. 1. These *H* rods are to be used only at and near the place of maximum moment in the beam and should be quite close together.

9 But how much does this increase the strength? As bearing upon the subject, an extreme case may be cited from a paper by Leon S. Moisseiff read before the American Society for Testing Materials. The compressive strength of cubes of concrete, reinforced in every direction by a large percentage of metal in the form of nails, was increased to two to three times the strength of plain concrete. Some designers have already noticed an increase of strength under similar conditions and are taking advantage of it, but are making assumptions regarding its amount for different percentages of reinforcement.

10 So far as the writer knows, no T-beams have been tested with their flanges reinforced and loaded in such a way as to carry their loads to the beam and thus to strain the beam in the same manner as in actual practice. It seems that future tests should be along such a line, if greater economy is to be reached in design and our knowledge is to become more exact with fewer assumptions made.

11 In conclusion, it would seem as though the slab concrete were overstrained at *E* and *F*, where it is used for T-flanges, for negative slab compression and for vertical shear from slab loads. Unless it can be ascertained whether lateral restraint, and the use of the rods as shown, will increase the strength necessary to resist this strain safely, it would be better not to calculate for T-beams, but to make the rectangular section sufficient to meet the stress. Even this rectangular section should be designed with a conservative concrete compressive stress, because its top surface is generally considered at about the point where the slab rods pass over it, this including the concrete at *G*.

12 Fig. 2 shows the cross section at the center of a T-beam, and a method of loading which seems to give promise of results which will come nearer to showing how beams in actual construction are stressed than rectangular beams whose compressive side is wholly plain concrete. The load over the stem should be less than the flange loads; and should agree with actual floor loading where the slabs carry most of the loads to the beam and produce tension in the rods and concrete at the surface over the stem, compression at the under side of the slab at the stem and shear near the stem. As tie rods are always

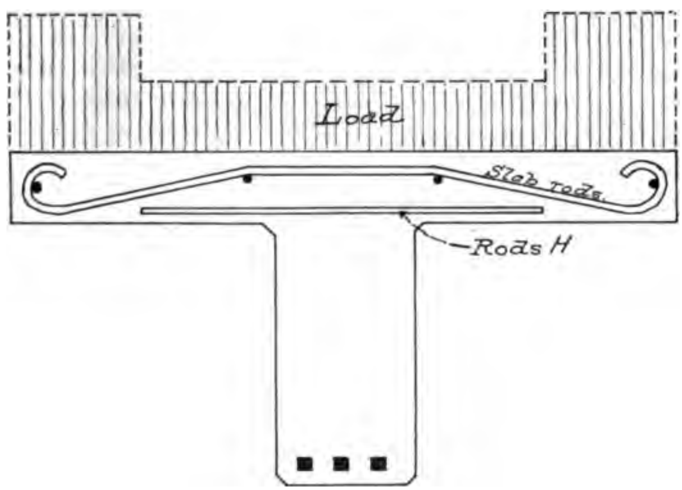

FIG. 2. CROSS-SECTION AT CENTER OF T-BEAM, SHOWING METHOD OF LOADING.

used in practice it would be well to use them here. They are shown by dots in the diagram. The slab rods in this case are bent to act as anchors, and the tie rod at the edge is wired to them on the inside.

13 It is acknowledged that the loads on the flanges do not stress them quite as they would be stressed in a floor system; but if the compression, tensile and shear stresses are not more than those that would be produced, were the slabs continuous, it is thought that as their stress is at right angles to the beam this difference will make no practical difference with the results on the beam.

DISCUSSION AT NEW YORK

E. P. Goodrich.[1] The several theories which were the basis of the formulae used by Professor Lanza are approximations to actual conditions, and are made the basis for calculating special points in construction work. The first method is used primarily because of its ease of application to ordinary conditions, and the factors now introduced into the formulae are based almost exclusively on the results of actual tests. For instance, in the particular series of tests made at the Massachusetts Institute of Technology the ratio of the modulus of elasticity as found by experiment to the computed value is only eight and a fraction. On the other hand, diagrams of Professor

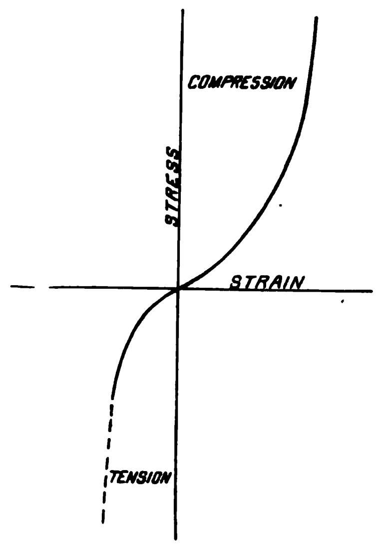

Fig. 2 Stress-Strain Diagram for Tension and Compression

Talbot's beam tests, in which the position of the neutral axis is shown, give a ratio of more nearly eighteen, showing that the factor introduced has no real relation to actual conditions. It is the adaptation of a formula to tests, rather than the use of a formula to check various kinds of investigations. Occasionally the straight-line formula has been used to compute deflections and stiffness, as was reported not long ago in an article published in Engineering News; but as to the accuracy of this use there has been some adverse criticism.

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2 As has been said, Considère's theory was based on certain experiments, the accuracy of which has also been questioned. Professor Mörsch of Zürich argues both for and against them in his book entitled Eisenbetonbau, describing certain experiments with concrete beams, in which he determined the stress-strain diagram for both tension and compression, finding some such conditions as that shown in Fig. 1. If in any beam section, the neutral axis be established, and the actual stresses laid down graphically above and below this neutral axis at any point, and if the centroids in each section are determined, and the distance between them measured, the moment which must theoretically be sustained by the beam can be computed. Mörsch tested some specimens both in compression and tension, and also in bending, and computed the theoretical bending moment and ultimate strength by methods similar to Considère's, using a practically constant stress in the concrete below the neutral axis. He found that the theoretical bending stress in kilograms per square centimeter was 20.7, while that found as an average of three actual experiments was 21.4, showing a very close agreement in this particular instance.

3 In the case of three other beams in which the percentage of steel varied from one-half of one per cent to very nearly two per cent, Mörsch made a similar computation based entirely on a stress relation similar to that of Fig. 1. He found the resultant of the two tensile stresses, in the concrete and the steel, then measured the distance on his diagrams between the centroids of compression and tension, and computed the moment which was found to correspond cosely with the test conditions.

4 Another series of tests of considerable interest is that made by Dr. Müller for his doctor's thesis for the Hanover Technical High School. He treated concrete beams in a manner similar to that of Professor Lanza, except that he used thirteen points in the depth of the beam, and measured by three methods the actual strain relation which existed at different times. In all his work he used simply a safe working stress, to the limit allowed by the German government regulations. He found that in a solid beam the stress varied to a certain extent, was very nearly of the straight line type when measured at all his thirteen points; while with a beam in which he built in fourteen artificial cracks by putting sheets of metal close together in the beam, he found that the stress relation more nearly corresponded with Considère's theory These artificial cracks produced a variable stress between the sections, so that the stress in the steel was actually less between the cracks, some of the stress being thrown into the con-

crete, as illustrated graphically in Fig. 2, in which the ordinates above the base measure the tensile stress.

5 The question of shear has been mentioned, but its effect upon deflections has not been discussed. The writer believes this is very important, because of two series of tests which he made some years ago on beams, one series of which was reinforced only by horizontal rods, and the other by vertical stirrups also. The deflection was three or four times as much in the case of the beams without the vertical steel—shear reinforcement—as in the case of beams with considerable vertical reinforcement. Each series had exactly the same amount of steel in tension. Of course theoretically the vertical stirrups could not affect the tensile stresses in the bottom of the beam. The ordinary theory by which deflection is computed does not include a factor for shear, which actually does have some effect on the deflection, both theoretically and, as shown by these tests, practically. It must be taken into account, as well as the tension in the concrete, if the actual conditions in the beam, especially with regard to stiffness and deflection, are to be considered.

Fig. 2. Variable Stress Produced by Artificial Cracks

6 It seems necessary that some relation between deflection and stress should be definitely determined, because deflections can be more easily measured in any beam test than any other phenomena Almost every novice determines the deflection, although he does not know the relation between it and the stresses involved. It is only through discussions such as this that some true basis can be reached for the computation of the stresses involved in continuous members.

7 There is another point concerning which the writer has made some experiments. By means of plaster of Paris, ordinary sharp carpet tacks were applied to the sides of a beam, with the points sticking outward. The beam was loaded centrally, and the actual deflection curve was simply picked through a piece of paper from time to time as the load was increased. The curves were then enlarged and used as a basis for comparison with the theoretical elastic curve of a beam loaded centrally. There was a very large discrepancy,

which was more nearly coördinated when it was assumed that the load was distributed over a length something like one and one-half or two times the height of the beam. It is to be hoped that experiments will be made in regard to the deflection of beams and the distribution of stresses, so that some true relation can be determined, between this element, which is easily measured, and the other elements which are usually unknown: that is, in regard to the relation between deflections and the actual stresses of compression and tension.

PROF. WALTER RAUTENSTRAUCH. I regret that more observations are not recorded and plotted in the paper and that the methods of making the computations and obtaining the data are not given. It would be interesting to plot the variation of deflection with load as observed, and as computed by the three formulae selected for comparison.

2 I would ask Professor Lanza how he made his observations for the strain in both concrete and steel and also how he determined from these the neutral axis of the section. If these data were submitted it would be possible to make a comparison with results obtained by assuming other possible values of E, for example, and thus to ascertain to what extent the differences reported might be due to assumed and possible actual values.

3 As concrete construction is for the most part monolithic, and very few beams of the particular kind tested are used, I believe it is of much broader interest to investigate methods of measuring strain and computing stress than formulae for simple beams. It is a fact, I believe, that all the data reported in this paper as actual stresses in concrete—actual stresses in steel—were obtained, not actually, from direct observations, but rather from relations between stress and strain assumed to exist in the concrete or steel. The same I believe is true in regard to the determination of the neutral axis. If Professor Lanza will tell us what assumptions he made in determining these values we will be in a better position to judge their worth.

4 I need hardly call attention to the fact that the modulus of elasticity for concrete in tension and compression is quite variable. It seems to depend upon the age of the concrete and the intensity of the stress. I believe it would have been of some value to take a slice from the end of these beams and obtain a stress-strain diagram, in order to compute the several values of E and the limits of stress for which each value of E is constant. Otherwise the actual values of the stress are not much more reliable than the values as computed by the formulae, since both are computed from assumed relations.

5 It is interesting to note that Formula B is based on a rational assumption concerning the variations in compressive stresses above the neutral axis. The fact has been well established that the stress varies as the ordinates of a parabola, and not as the ordinates of a straight line. On the other hand, I am inclined to doubt the statement of Considère that the concrete on the tension side can undergo an extension much greater than 0.02 per cent without cracking, when the beam is reinforced, whereas when not reinforced the concrete cracks when the extension is from 0.01 to 0.02 per cent. The mere fact that a reinforcing rod is present does not seem sufficient to change the physical properties of the concrete.

6 I believe Professor Turneaure has shown Considère to have been wrong in this assumption. It is not at all unlikely that Considère removed a piece of concrete in which no cracks had developed. Furthermore, if cracks are allowed to develop on the tension side—and this has frequently been observed in beams under working load—might not this crack gradually extend under repeated loading and seriously impair the safety of the structure?

B. H. Davis.[1] Certain practical considerations may be cited to illustrate the difficulties confronting the experimenter seeking a rational solution of the deflection problem. Shrinkage is the worst, or perhaps the most indeterminate factor to be eliminated, since it spoils so many carefully performed experiments, being a large cause of the lack of uniformity so generally noted in experimental data.

2 The shrinkage of a concrete block 8-in. square by 2-ft. long has been shown to shorten appreciably a bar ½-in. square embedded in it and accurately measured before and after the setting of the concrete around it. This produces an initial tension in the concrete and an initial compression in the steel. In the case of a beam reinforced in only one plane, as perhaps some of the beams tested may have been, these initial strains may largely account for the lack of uniformity in the results obtained.

3 The shrinkage of concrete in setting, nearly always a variable factor, has almost completely upset the theory of arch-ring deflections when the arch centering is struck. Some settle very considerably upon striking the centering, especially when the arch ring is a monolith from skewback to skewback, while others settle hardly at all

[1] Assistant Engineer, Lackawanna R. R., Hoboken, N. J.

when alternate voussiors are made and allowed to set and shrink before the ring is keyed. Shrinkage, it has been proved, almost entirely causes this lack of agreement between the theoretical and the actual deflections when arch centers are struck. It would therefore seem logical to assume that the same cause figures prominently in the deflection phenomena of beams.

4 The shrinkage of a beam of large cross section, acting in opposition to that of a smaller beam, has been known to crack the weaker member from top to bottom, breaking up any dependence that might otherwise have been placed upon the concrete in tension, before the beam had been called upon even to support its own dead load.

5 In designing for a given load by the commonly accepted straight-line formulae for obtaining stresses in steel and concrete, and using the prescribed unit stresses of the building code, a certain factor of safety results. In other words, an overload of two or three times the load assumed in the design, may be applied, and when removed, the structure should be just as capable of supporting the working load for which it was designed as before the overload was applied.

6 Now, granting the conclusion of the author, in Par. 27, that tension in the concrete materially affects the deflection and strength of beams (between certain limits of load), would it not still seem unwise to take advantage of this tension factor in any design where the assumed load limits might be overstepped at some time, leaving the beam to serve the remainder of its period of usefulness without the tension factor counted upon in its design?

7 Almost every design is over-stressed sooner or later, occasionally by test load, but more often, perhaps, because of the enthusiasm of some shop foreman in showing what his building will stand in the way of abuse. For example, loaded cars of gravel and broken stone, and later a 600-class standard-gage locomotive, were run across a machine and erecting shop floor that was designed for a uniformly distributed load of considerably less than one-half the concentrated moving loads applied. this without any apparent damage to the floor.

8 Settlement, which very often upsets carefully made calculations, causes even more indeterminate stresses in reinforced-concrete than in other types of construction, this being due to the continuity and the monolithic character of the material. This fact further emphasizes the necessity for conservatism in working formulae.

9 Construction joints, put in as they usually are, at points of maximum moment, make any reliance upon the concrete in tension

entirely out of the question where such joints occur. It is not generally conceded that construction joints so located do materially weaken a beam except in shear.

10 A beam, accidentally cracked entirely through near its middle while being placed in a testing machine, tested higher than the average of several other beams of the same size and reinforcement, showing that a plane of fracture approximately normal to the center line of a beam had not, in this particular case, unfavorably affected the ultimate strength of a beam equally loaded at its third points.

11 Until more is definitely known concerning the shrinkage of concrete and the many other stresses in reinforced-concrete beams at present indeterminate, as a matter of conservatism it would seem better to disregard tension in concrete as a moment-resisting factor.

Chas. B. Grady.[1] Professor Lanza and Mr. Smith have clearly brought out the fact that three of the formulae used for the design of reinforced-concrete beams are approximate with a load of one-third the breaking load. The writer will say a few words in reference to the use of these formulae in the design of beams.

2 Formulae *A* and *B*, which are used by a large number of engineers, do not allow anything for the tension in the concrete and therefore must give for rectangular beams results which are mere approximations up to a point at which the concrete fails to act in tension, but the writer believes that if a comparison had been made at say double the load used, Formulae *A* and *B* would have given better results, and possibly nearer those found by actual test, than Formula *C*, especially for the value of σ_s (stress in steel per square inch).

3 In tests of similar beams made at Cornell University by Messrs. Paulus, Tripp and Davis, the average variation in the values of σ_s (stress in steel per square inch) deduced by formula *A* from those found by experiment was 34 per cent with a load of 4000 lb., and less than one per cent with a load of 8000 lb. The above figures are for five beams having an average breaking strength of 13,200 lb.

4 The errors in values deduced by Formulae *A* and *B* are more liable to be on the side of safety than the errors in values deduced by Formula *C*, and while there is no doubt that Formula *C* will give more accurate results when the stress in the steel is comparatively small, it is the opinion of the speaker that Formula *C*, and

[1] Asst. Mechanical Engineer, New York Edison Co.

other formulae making allowance for the tension in concrete, should be used with caution.

5 It is the practice of many engineers to design reinforced-concrete beams in accordance with certain working stresses and to endeavor so to proportion the beam that it will fail by tension, that is, by either breaking or stressing the steel to a point considerably past its elastic limit, thus making the factor of safety a function of the stress in the steel. In such cases, no matter how much the concrete has helped out the steel under working conditions, when the beam is overloaded the steel must take care of practically the entire tension; and therefore the writer believes that it is wiser not to introduce a value for the tension in the concrete into the formulae used in the design of reinforced-concrete beams.

6 The speaker believes that the formulae for deflection deduced by Professor Lanza and Mr. Smith will be of great value to engineers, and that any one of the three formulae will give results accurate enough for practical purposes in figuring the deflections of T-beams, more of which are used in buildings than rectangular beams.

FRANK B. GILBRETH. The most important subject related to reinforced-concrete, from the standpoint of the mechanical engineer, is the design of forms, for it is the forms that afford the greatest opportunity for the saving of money, and the consequent reduction of price per cubic foot of new buildings.

2 Beams have been designed and built of rectangular section and over 64 ft. 0 in. long, and have been perfectly satisfactory. The most successful building of today as well as of the future must be designed with regard to the economical design and use of forms, and not to the greatest saving in the quantity of steel and concrete used. The forms are the most expensive single item of reinforced-concrete work.

3 It is by no means rare to see designs for saving concrete where the value of the concrete saved amounts to much less than the cost of the special or odd-sized forms required.

PROF. WM. H. BURR. Much has been said about the disagreement of theoretical results with the results of experiments. That is an observation which may be made, I believe, in the case of every material which has ever been used by the engineer; scarcely more so of concrete, either plain or reinforced, than of other material. When a comparison of this kind is made, I think we should bear in mind, first, what theory is used.

2 The so-called common theory of flexure probably is not strictly applicable to any reinforced-concrete beam which has been broken. It is a theory which applies to a beam of very small depth, compared with the length of span. This is not the kind of beam usually found either in plain or reinforced-concrete, and usually not even in steel. It is not a matter of surprise, therefore, that such a theory does not give the results found by experiment.

3 It seems to me we shall have to proceed with reinforced-concrete beams precisely as with beams of other material, viz., use a simple working hypothesis for the purpose of securing a formula in which empirical quantities may be used. That is the case with wrought-iron and steel beams, with timber beams, and with all other beams, and it is markedly so, even to a greater extent, with columns.

4 The three theories, A, B and C, may be considered in view of the varying conditions at different stages of stress. It would be difficult to show from any results of tests of concrete, that the law of distribution of stress in theory B is justified. There are some tests which show a graphic relation between the intensities of stress and strain, which approximates a parabolic curve, but probably no nearer than a circular curve or some other. The majority of tests show that line much more nearly straight than parabolic within the limits of stress found in ordinary concrete beams.

5 It is true that concrete has considerable tensile resistance, when it possesses any, but I think there are few engineers who have used much plain or reinforced-concrete, who would be willing to trust the tensile part of the beam to carry load, and to be so recognized in the working formula.

6 The result of the slight contraction of concrete, possibly not within the first two months, perhaps not within the first year of its life, is to create fine hair cracks. We do not know how far these enter the mass; they may be only skin-deep, but in some cases they are much deeper. Hence if the beam should show a continuous concrete structure on the tension side for the first two or three months, it does not follow that it is going to remain so. If we are to recognize such a possibility, and it seems to me we would not be justified in neglecting it, the only safe procedure is that usually followed, of neglecting tension in concrete. That does not mean that concrete may not sometimes have considerable tensile resistance. It simply means that such resistance cannot safely be recognized in ordinary concrete work.

7 These cracks may be very much reduced by continual wetting of concrete after it has been put in place. That is one direction in which the concrete work may be improved. We do not wet the concrete nearly enough after the forms are taken away. If it were feasible, concrete should be kept thoroughly wet from three to six months after being put in place. This is not practicable; but after the forms are taken away, the concrete should be kept soaked with water just as long as possible. The contraction will be less and there will be fewer hair cracks, but it will be impossible to eliminate them entirely.

8 We should be sensible, as engineers, in connection with reinforced-concrete work, precisely as we are or ought to be in everything else, and use the simplest possible formula, i. e., the straight-line formula, and not strain after some ultra-refinement which, when we come to examine it, has little or no solid basis. We should resort to proper theories and select a simple working hypothesis, and then use the test beams to determine such theoretical coefficients or quantities as will make the resulting formulae represent actual results as nearly as possible.

PROF. J. C. OSTRUP.[1] Within a short time, from fifteen to twenty years, at most, reinforced-concrete has gained an enviable position in the construction world, and unquestionably, in spite of many inherent shortcomings, will better its reputation in the future among both engineers and laymen. It is, therefore, to be regretted that the trend of the authors' paper is toward a negative rather than a positive support.

2 It is a well-known fact that the greater number of the deductions and working formulae obtained from the science of applied mechanics are based upon certain assumptions which to a greater or usually less extent circumscribe the use of such formulae. The errors resulting from these fundamental assumptions vary considerably with the different engineering materials with which we deal; they often vary considerably even with the same material, changing somewhat with the extreme fibre stress, the manner of application of the load, etc. The assumptions made in regard to the behavior of structural steel are probably nearer the absolute truth than for any other engineering materials, so near, in fact, that many engineers have come to regard the theory of steel design as following an unassailable mechanical law. Nevertheless this is not so.

[1] Professor Structural Engineering, Stevens Institute of Technology.

3 On the other hand, the theory of reinforced concrete is based upon many assumptions, some of which can be better defended than others, and some of which have undergone, and will continue to undergo, modifications from time to time. It is also based upon many widely varying experiments which the experimenters themselves have been struggling to reconcile. Some of the most important of these assumptions, together with a brief account of their probable effect, are:

a That the applied forces in bending are perpendicular to the neutral axis.

4 This is incorrect, of course, inasmuch as the neutral axis under deflection follows a curve resembling a parabola. The resulting error is, however, extremely small.

b That a sectional plane, true before bending, also remains true after.

c That each fibre acts independently of adjacent fibres.

5 The last of these assumptions is particularly faulty, inasmuch as the ordinary reinforced beam usually has its reinforcement varying in amount, both horizontally and vertically, throughout its length. In other words, unlike a rolled-steel beam whose moment of resistance is uniform from end to end, the reinforced beam is not uniform in strength, the stronger parts tending to assist or restrain the weaker. The error from this assumption cannot be evaluated.

d That the concrete and the reinforcement will stretch or compress together without breaking the contact bond between them.

6 This condition, when complied with, as it infallibly must be in all cases, unquestionably sets up secondary local stresses, the magnitude of which cannot be even guessed.

e That there are no initial stresses.

f That the stress-strain curve for compression is a parabola.

7 The fulfillment, or the non-fulfillment, of the last two assumptions, is probably what causes the greatest divergence between theory and tests. A concrete beam is a casting, in a sense. If the mixture were perfectly uniform throughout, there would most probably not be any initial stresses due to the chemical action of setting. This is evidently not possible; hence throughout the beam there undoubtedly exist initial stresses of uncertain magnitude. This fact, in itself,

would surely affect the stress-strain curve, but in addition we must consider the variable modulus of elasticity for the concrete. This varies not only in the same beam, according to unit stress in the extreme fibres, but also in beams of the same identical composition according to its depth, i. e., to the relation between the extreme fibre stress and the average fibre stress.

8 In addition to these mechanical considerations, we have many physical considerations governing the strength of concrete and reinforced-concrete beams. Such physical conditions must largely depend upon the personal equation of the engineer in charge; they may be guarded against, and their effect minimized but not wholly eradicated. When present, their influence can only be surmised.

9 To make this a little clearer, let us assume a case where a number of beams were to be prepared for a testing machine and where great uniformity naturally would be sought; to insure which, only one grade of cement, one of sand and one of broken stone, would be employed. Next let us look into some of the more important points affecting the strength of concrete, as follows:

a Condition of the cement; whether all the bags in a cargo are of the same age, or manufacturing batch; quantity of carbonic acid contained; degree of moisture (since the outside bags in a stack, and even the outside layer in the same bag, often absorb considerably more moisture than the inside).

b Uniformity of quality of the sand; whether or not it contains in spots, loam, clay or other impurities, etc.

c Uniformity of the broken stone; whether or not the stones are alike in strength and texture; whether or not they are broken to a uniform size, etc.

d Quality or purity of the water; method of mixing the concrete, or difference in methods of mixing from batch to batch, even by the same gang.

e Tamping and placing of the concrete, including the often unavoidable variations in the degree of flexibility of the support between the ends and the center of the beam while the concrete is being tamped.

f Workmanship. A man is not a machine, consequently the materials mixed and the beams made, even by the same gang, will often vary considerably in spite of precautions. May not beams vary much more when made by different sets of workmen?

10 Besides the foregoing points affecting the mechanical laws governing the strength of the concrete, there are others; but enough have been indicated here to show that, when tested, a variation in their strength must exist.

11 Since each experimenter must base his deductions upon the results of his own observations, a divergence in theres ulting formulae is the natural result, and furthermore, were he to repeat the same tests under similar circumstances, his second results, in view of the foregoing, would vary from his first. With all this in mind, is it any wonder that closer agreement between the various working formulae most generally in use, has not so far been reached? To an unbiased mind the wonder is that the divergences are not even greater.

12 Returning to the conclusions of the author, he states in Par. 24 " . . . . the observations made thus far are not sufficient to furnish the means for determining the actual distribution of the stresses, and hence for the deduction of reliable formulae . . etc." This may be strictly true in theory, but will hardly be generally accepted as a matter of fact. On the contrary, it is quite within good reason and good practice to deduce reliable formulae, even where the action of some of the minor points involved is in doubt, so long as the effective range of such points is known. In this connection it may be recalled that concrete and masonry structures, centuries old, are still standing and doing effective service, though they were designed from formulae and data far less reliable than those now at our disposal.

13 The author further says: "It follows therefore that whichever of the theories is adopted for practical use, it can be regarded only as a sort of working hypothesis." This, of course, is a sweeping condemnatory statement which, if it can be applied to the theory of reinforced-concrete construction, can, it is believed, be equally well applied to the theories underlying any form of construction; for no amount of theory, unaccompanied by practical experience and sound judgment, will prevail, either in the mechanical or in any other engineering field. This fact cannot be too strongly emphasized.

14 In Par. 26 the author states that theory *C* gives results in closer agreement with experiments than does either *A* or *B*. This is undoubtedly true, but so far as the evidence in Tables 5 and 6 is concerned, any one of the three theories is based upon "reliable formulae" or, what is more to the point, the designs resulting from their use would be wholly reliable. As a matter of opinion, the preference should be for *A* or *B*, since they are nearly correct in regard to the

unit stresses in the concrete,—the weaker material,—whereas they give somewhat smaller stresses for the steel than those expected.

15 It is equally true that no reliable deflection formulae can be deduced without taking into consideration the tension in the concrete. We can, however, go a step further, and state that such formulae, to be correct, must also include a provision for a deflection increment due to shear.

16 In concluding these remarks, the writer would suggest a caution to such alarmists as are prone to appear from time to time against a useful and excellent building material. No public good can result from arousing the apprehension of either engineer or layman with respect to reinforced concrete, and those of us who have had the opportunity of using it for a number of years cannot help but be impressed with its increasing serviceability and scope.

E. Lee Heidenreich.[1] The tests at the Massachusetts Institute of Technology, as well as those at the University of Illinois, were based upon a concrete mixture of 1 : 3 : 6, while those of Considère are based upon a mixture of 1 : 2½ : 2½. I have repeatedly at meetings of the "Joint Committee" urged the desirability of employing stronger mixtures, and mixtures of a "maximum density" rather than a certain proportion; and I believe that with such stronger mixtures Formula *C* will come still nearer to a correct interpretation of stresses and strains. If so, is it not natural to hope that in our reinforced-concrete building constructions, lesser dimensions of beams and girders, thinner floor slabs, and consequently a reduced item of *dead load* will result, also materially reducing the present disadvantages of heavy columns and foundations?

2 The most wonderful constructions of tanks, reservoirs and bridges in Europe have resulted from mixtures of 1 : 3 or 1 : 5, properly graded. Why should not our beam tests be based upon such mixtures, notwithstanding the fact that at first glance they may not appear commercially advantageous for building constructions? I wish to place myself again on record as advocating a larger percentage of cement and a mixture representing a maximum density of the ingredients.

Prof. C. E. Houghton. The paper adds to our knowledge of the probable magnitude and sign of the errors due to the use of formulae

[1] Special Engineer, N. Y. C. & H. R. R. R., New York City.

deduced from a simpler theory. When the size of a structural member has been calculated by the use of a formula known to give a greater value to the unit stress than actually exists, the designer need not worry about the safety of that member. If in addition the probable magnitude of the error is known, corrections may easily be made where it is considered necessary to reduce the cost or weight of the member.

2 The neglect of tensile resistance in calculations of the strength of reinforced-concrete beams finds a parallel in the common practice for the calculation of the strength of riveted joints. The friction between the plates unquestionably adds to the strength of the joint, yet as far as the writer knows, no theory has been accepted in American practice that considers this friction as acting. This friction, like the tensile resistance of concrete, may vary from zero to a maximum value, and therefore should be neglected, as neither can be depended on for additional strength.

3 All formulae for the strength of reinforced-concrete beams contain a factor whose value is the ratio of the modulus of elasticity of steel to that of concrete, and any error made in the assumption of that value affects the result in the same proportion. The modulus of elasticity of steel is practically a constant term, but that for concrete varies through a wide range of values depending to a certain extent on the proportions of cement, sand and broken stone used in the concrete.

4 With the large possible variation of this ratio in mind, it would seem reasonable to suppose that the probable error, either in assuming a straight-line law for the variation of the compressive stress in the concrete, or in the neglect of its tensile resistance, will be less than that due to the choice of the value of this ratio. What is needed is a value for this ratio, determined by applying the formula derived from the straight-line no-tension theory, to the results of a great many tests on specially prepared beams.

5 The number of variable conditions that would affect the results in any such investigation is so great that unless one of our national engineering societies will undertake it there seems to be but little prospect of obtaining anything more than an approximate value based on the results of compressive tests on concrete.

Wm. Wallace Christie. The writer is particularly interested in the applications of reinforced-concrete in engineering work, and has had to do with the designing of a great many floors, foundations and other work. He agrees with Professor Burr, and others not prepared to accept or consider a theory of design of concrete-steel beams allowing tension in the concrete, or an increase by reinforcement of the ability of the concrete to resist tension.

2 After concrete work has been erected for a time, hair-cracks, and others more decided, often develop in the beams. An example of this has already been cited: a 70-ft. or longer concrete girder, with its center, at least, resting on hard pine timbers.

3 With the large factor of safety necessary in the design of concrete-steel beams, one cannot go very far wrong in using any of the three methods mentioned, but the writer prefers a straight-line formula.

4 The paper deals in particular with beams, which in practice are seldom used, except as lintels, or over openings in building walls. The experiments conducted with these beams will not give the results obtainable by the use of T-beams, and the writer doubts whether the test of a single T-beam, made in the test room, will develop the same strength or other features, as a test made on a similar T-beam which is part of a floor system. The beam tested in the laboratory is not joined tightly with the rest of the floor, while in actual construction the iron would necessarily be secured to the other parts of the floor system.

THE AUTHORS. The data and the results of observation for the first five beams, which have been asked for, are contained in a paper by Gaetano Lanza, published in the proceedings of the American Society for Testing Materials for 1906.

2 The modulus of elasticity of the concrete was obtained from tests made upon seven 8 in. by 8 in. by 60 in. plain compression pieces of the same age, materials and mixture as the beams. The values of E are as follows:

	2,479,000
	2,223,000
	2,367,000
	2,264,000
	2,670,000
	2,623,000
	2,341,000
Average.................	2,424,000

In our paper we have used 2,335,000 in order to permit of the use of $r = 12$.

3 It may be added that the neutral axis was determined for each load from the strain diagrams (which are shown graphically in the paper referred to) at the intersection of the plotted line with the vertical datum line. Numerical details of the strains will be given in Appendix No. 1, as they seem to be desired.

4 As reference has been made to evidence tending to discredit Considère's theory regarding the ability of concrete to stretch when reinforced, it may be well to say that it is neither the object of the paper to discuss this question, nor to take sides for or against this theory. The history of the main part of the controversy is as follows:

5 The theory was attacked by Kleinlogel in an article published in Beton u. Eisen, Hefte 2 and 4, 1904, in the light of certain tests which he had made. The two tests of Considère (Par. 8) were made as a refutation of Kleinlogel's argument. An account of them may be found in Considère's book on reinforced concrete. A subsequent reply by Kleinlogel, and a reply to this by Considère, are to be found in Beton u. Eisen, but no new matter is given.

6 In Beton u. Eisen, Hefte 11–1905, Professor Ostenfeld gives an account of the results of some computations made by him upon the beams tested by Kleinlogel, and in the light of these he says, "Thus far I regard Kleinlogel's tests as a beautiful though unwilling confirmation of Considère's theory." To this Kleinlogel replies in Beton u. Eisen, Hefte 1–1906, but this reply contains no new evidence.

7 Fear seems to be expressed by some that pointing out the very considerable discrepancies between the results of computation made by theory A, and the results obtained by experiment, is equivalent to a condemnation of all structures where theory A was used in the computations. No such condemnation, however, is intended by the authors. They believe, however, that the more we realize the facts in any case, the better prepared are we to use our judgment as engineers, in designing any construction.

8 Most of the arguments advanced in support of the entire sufficiency of theory A may be summarized as follows:

a The calculations can be more easily made.

b That the mere fact of neglecting the tension in the concrete results in safety, though practically all admit that the concrete does resist tension in the early stages.

c The use of construction joints, which often take the form of a vertical joint at the middle of the span when work on a given floor extends over a period greater than one day.

9 These matters will be considered in the same order:

a There is no doubt that the calculations are more easily made when theory *A* is used.

b Whichever of the three theories is used, it is not customary to calculate by means of it, the stresses which produce diagonal cracks, and it is a fact that in a very large percentage of the beams that have been tested, the failure has been due to these diagonal cracks. Hence it seems to us that until we have arrived at some means of making calculations to determine these stresses and strains in such a way that the calculated results shall have a fair degree of agreement with the results obtained by experiment, we can hardly claim to have an all-sufficient theory. Moreover, in the case of beam A-1, the only one for which the shear has been figured, it is greater when determined from theory *C* than when obtained from theory *A*, the difference being in one case 57 per cent.

c When a construction joint is introduced, the beam is necessarily weak, and until tests of such beams are made, we cannot claim to know what theory will apply to them.

10 Other considerations which it would seem worth while to discuss are the following:

a The presence of initial stresses due to shrinkage.

b The variation in the value of the compressive modulus of elasticity of concrete.

c The recommendation made by some that the formulae to be used be based upon loads larger than one-third the breaking load, and by some upon the breaking load.

d The question of so proportioning the reinforcement that the breaking shall be due to the tension in the steel exceeding the elastic limit.

11 Discussing these in order we have:

a The presence of initial stress is of course a great source of uncertainty in reinforced-concrete, as well as in cast iron, and hence we should expect irregularities due to this cause, the amounts of which are very difficult to estimate. Whether their influence is still large or not at one-third the breaking load, is a debatable question, though it must be comparatively less at one-third than at smaller loads. On the other, hand with loads greater than one-third the ultimate, the ratio of stress to strain becomes quite variable, and any rational formula becomes inaccurate.

b In the light of the experiments made by different men and in different places, it would seem to the authors that the variations of the modulus of elasticity for compressive stresses in the concrete, not more than one-third the ultimate, would not be very excessive.

c In the case of steel or other beams it is well known that the ordinary formulae do not apply when the stresses in any of the fibres have passed the elastic limit; hence the difference between modulus of rupture and outside fibre stress at breaking.

d Regarding the question whether theory *A* will agree better with experiment when the percentage of reinforcement is kept so low that the elastic limit in the steel will be exceeded before any fibre of the concrete has to bear a stress equal to its crushing strength, the only evidence in the paper is the following: in one case the percentage of reinforcement was as low as 0.99 per cent, and in three others, 1.25 per cent, and in these three cases the discrepancies of theory *A* are large.

12 In general, it seems to us that thus far not enough systematic work has been done by way of experimenting and calculating in order that we may have more accurate knowledge about a number of matters, among which may be mentioned:

a The actual distribution of stresses not merely in the case of longitudinal reinforcement, but also with diagonal and other reinforcements, and also in T-beams.

b A study of the diagonal tension, not only at the neutral axis, but elsewhere.

c A study of the conditions necessary that the breakage may always be due to the reinforcement exceeding the elastic limit, and whether diagonal cracks occur in those cases.

d A study of the effect of construction joints.

13 There only remain for discussion a few additional matters raised by different gentlemen. While it appears from the last

column of Mr. Worcester's table that method C gives average results on the negative side, it must be remembered that they depend upon the value taken for t (the tensile strength of the mixture). This table, as well as Table 5, clearly shows that if a slightly lower value of t had been used for these six beams, their average error would have been a positive one, and also smaller than that by using A.

APPENDIX NO. 1

STRAINS FOR THE MASSACHUSETTS INSTITUTE OF TECHNOLOGY BEAMS

The strains were measured at four points in the depth of the beam on each side as described in the paper before the American Society for Testing Materials, already referred to. Columns 1, 3, 4, 2 in the following tables give the strains for these points. Points 3 and 1 were one and five inches, respectively, *above* the center of the beam, while points 4 and 2 were one and five inches, respectively, *below* the center.

BEAM A-1

One 1-in. Plain Rod — Age 53 Days
Initial Load 1250 Lb. — Breaking Load 15000 Lb.

Loads Lb.	Strains 1	3	4	2
2250	0.000023	−0.000008	0.000033	0.000060
3250	0.000064	−0.000009	0.000048	0.000108
4250	0.000107	0.000008	0.000057	0.000195
5250	0.000186	0.000022	0.000071	0.000262
6250	0.000228	0.000009	0.000124	0.000352
8250	0.000345	0.000005	0.000186	0.000569
10250	0.000448	−0.000022	0.000274	0.000795
12250	0.000543	−0.000026	0.000337	0.001017
14250	0.000672	−0.000088	0.000466	0.001279

BEAM A-2

One 1-in. Twisted Rod — Age 49 Days
Initial Load 1250 Lb. — Breaking Load 16500 Lb.

Loads	Strains. 1st application			
Lb.	1	3	4	2
2250	0.000044	0.000012	0.000003	0.000033
3250	0.000082	0.000012	0.000027	0.000093
4250	0.000138	−0.000013	0.000077	0.000174
5250	0.000172	0.000016	0.000073	0.000251
6250	0.000216	0.000018	0.000108	0.000358
8250	0.000317	−0.000004	0.000202	0.000595
10250	0.000405	−0.000009	0.000271	0.000835
12250	0.000505	−0.000063	0.000391	0.001039

BEAM B-3

Two ¾-in. Plain Rods — Age 43 Days
Initial Load 1250 Lb. — Breaking Load 15950 Lb.

Loads	Strains. 1st application			
Lb.	1	3	4	2
2250	0.000073	0.000013	0.000017	0.000081
4500	0.000100	−0.000003	0.000059	0.000175
5250	0.000144	0.000015	0.000060	0.000223
6250	0.000195	0.000002	0.000096	0.000289
8250	0.000398	−0.000020	0.000182	0.000428
10250	0.000519	−0.000066	0.000301	0.000587

BEAM C-5

Four ½-in. Plain Rods — Age 35 Days
Initial Load 600 Lb. — Breaking Load 16240 Lb.

Loads	Strains			
Lb.	1	3	4	2
2600	0.000083	0.000018	0.000026	0.000087
4600	0.000219	−0.000024	0.000133	0.000296
6600	0.000337	−0.000067	0.000239	0.000532
8600	0.000444	−0.000059	0.000297	0.000751
10600	0.000542	−0.000091	0.000406	0.001023
12600	0.000631	−0.000137	0.000525	0.001272
14600	0.000765	−0.000209	0.000653	0.001525

BEAM E-9

TWO ½-IN. TWISTED RODS
INITIAL LOAD 1250 LB.

AGE 54 DAYS
BREAKING LOAD 21000 LB.

Load Lb.	Strains. 1st application 1	3	4	2
2250	0.000037	−0.000012	0.000029	0.000037
4250	0.000107	0.000003	0.000046	0.000134
5250	0.000155	0.000008	0.000060	0.000175
6250	0.000202	0.000004	0.000081	0.000256
8250	0.000275	0.000004	0.000122	0.000402
10250	0.000403	0.000010	0.000161	0.000541
12250	0.000486	0.000003	0.000212	0.000680

No. 1253

THE DESIGN OF CURVED MACHINE MEMBERS UNDER ECCENTRIC LOAD

By Prof. Walter Rautenstrauch, New York

Member of the Society

Machine members, such as frames for punches, shears and riveters, hooks and the like, when subjected to load are generally supposed to behave like beams originally straight and subjected to the same conditions. The usual analysis applied to such beams in determining the proportions required to withstand safely a given stress assumes that the maximum tensile stress at a in Fig. 1 = load considered as uniformly distributed over the section + the stress due to the eccentricity of the load. Symbolically expressed

$$f_t = \frac{W}{A} + \frac{Wle}{I}$$

where

f_t = maximum intensity of tensile stress.
W = load on beam.
A = area of section.
l = eccentricity of loading.
e = distance from gravity axis of section to point under stress f_t.
I = moment of inertia.

2 This analysis is unfortunately prevalent in textbooks on the design of machine elements and strength of materials, and has been accepted generally because of long standing. However, it does not agree with the results of experiment on members of this kind; in fact such experimental results are so different from results calculated by this formula that no confidence whatever can be placed in it and safe proportions can be obtained only by the use of a large factor of safety.

Presented at the New York monthly meeting (November 1909) of The American Society of Mechanical Engineers.

3 The writer has recently published[1] the results of a series of experiments which are remarkable in their disagreement with the results obtained by the formula. The crane hook was taken as an example of a beam of this sort and experiments were conducted on ten hooks ranging from 2 to 30 tons rated capacity. All hooks were furnished by the manufacturers. In Table 1 the results of the experiments are compared with the results by formula.

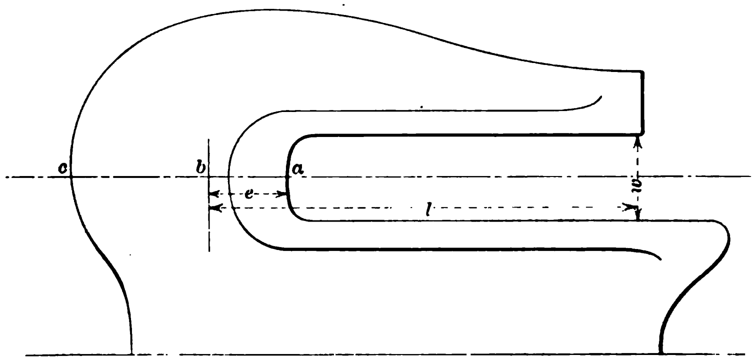

Fig. 1 Machine-Tool Frame Considered as a Loaded Bea

4 It is very evident that the assumptions on which the above formula is based are not correct, and that machine members designed on this basis have a much smaller factor of safety than is generally supposed. While this has been known in some quarters and attempts have been made to bring about an adjustment, no theory which has been developed seems to fit the case better than that evolved by

TABLE 1 COMPARISON OF RESULTS

Description of Hook	Load at Elastic Limit by Test Pounds	Load at Elastic Limit by Standard Formula Pounds
30-ton cast steel	56,000	115,000
20-ton cast steel	30,000	70,000
15-ton cast steel	48,000	145,000
15-ton wrought iron	16,000	73,000
10-ton cast steel	18,000	43,000
10-ton wrought iron	16,000	26,000
5-ton cast steel	18,000	52,300
5-ton wrought iron	14,000	20,800
3-ton cast steel	8,500	14,900
2-ton cast steel	4,700	14,900

[1] An Investigation of the Strength of Crane Hooks, American Machinist, October 7, 1909.

E. S. Andrews and Prof. Karl Pearson of London University. This analysis is deserving of much more attention than it has received, and it is surprising that even some who have had access to it have made the statement that the old theory was sufficiently accurate for the usual case of design. It will not be my purpose to give a complete derivation of the new formula, which has been published elsewhere[1] in complete detail, but rather to show how the results of the analysis may be made directly applicable to design.

5 The investigation referred to gives the following expression for the tensile stress at the most strained point in the principal section of beam:

$$f_t = \frac{W}{A}\left\{\frac{l}{\rho\gamma_2}\left(\frac{1}{\left(1-\frac{e}{\rho}\right)^{\frac{3}{2}}}-\gamma_1\right)+1\right\}$$

where

f_t = tensile stress at most strained point of section, pounds per square inch.

W = load on hook, pounds.

A = area of section, square inches.

l = distance from load line to gravity axis of section.

ρ = radius of curvature of belly of hook at gravity axis.

e = distance from gravity axis to point of maximum tensile stress.

γ_1 = a function whose value is equal to $\dfrac{\Sigma \dfrac{dA}{\left(1+\frac{y}{\rho}\right)^{\frac{3}{2}}}}{A}$ in which

dA is any differential area of section, a distance y from the gravity axis, Σ denoting the sum of all the operations indicated by the symbols.

γ_2 = a function whose value is equal to $\gamma_1 - \dfrac{\Sigma \dfrac{dA}{\left(1+\frac{y}{\rho}\right)^{\frac{3}{2}}}}{A}$

in which y is any distance from b equal to $\dfrac{dg}{\left(1+\frac{y}{\rho}\right)^{\frac{5}{4}}}$ at which an ordinate such as de is erected.

[1] Technical Series 1, The Draper Company's Research Memoirs, 1904.

6 The functions γ_1 and γ_2 are to be found for any section, as shown in Fig. 2. The half-area of the hook is *akhlc*. All possible ordinates of the nature of *de* determine the curve *aehjc*. Clearly then

$$\gamma_1 = \frac{\text{area } aehjc}{\frac{1}{2} \text{ area of section}}$$

Likewise, if at any distance y from b an ordinate is erected whose

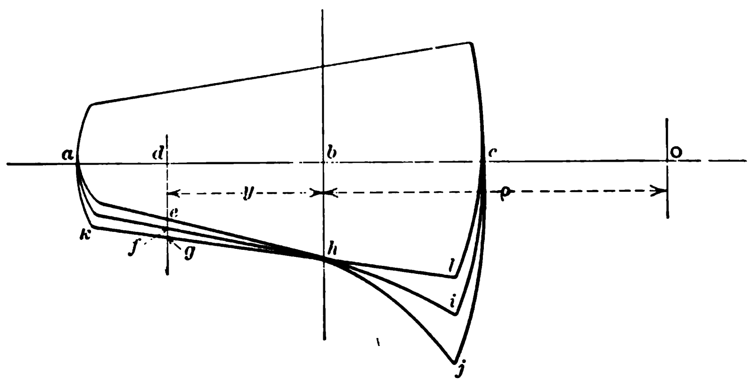

FIG. 2 CROSS-SECTION OF A HOOK

length is $\dfrac{dg}{\left(1 + \dfrac{y}{\rho}\right)^{\frac{1}{2}}} = df$, all possible ordinates of this nature form the curve *afhic*, and $\gamma_2 = \dfrac{\text{area } aehjc - \text{area } afhic}{\frac{1}{2} \text{ area of section}}$

7 The above analysis was applied to each of the hooks tested, with the results recorded in Line 3 of Table 2. An inspection of

TABLE 2 ANALYSIS OF HOOKS TESTED

LOAD AT ELASTIC LIMIT, POUNDS

RATED CAPACITY	30-TON	20-TON	15-TON		10-TON		5-TON		3-TON	2-TON
	Cast Steel	Cast Steel	Cast Steel	Wrought Iron	Cast Steel	Wrought Iron	Cast Steel	Wrought Iron	Cast Steel	Cast Steel
By test	56,000	30,000	48,000	16,000	18,000	16,000	18,000	14,000	8,500	4,700
By standard formula	115,000	70,000	145,000	73,000	43,000	26,000	52,300	20,800	14,900	14,900
By new formula	55,080	29,925	50,570	15,000	16,500	15,000	18,950	14,100	8,600	4,400

this table will show how nearly the analysis of Mr. Andrews and Professor Pearson fits the case and how far from correct are the results from the old formula. The new formula appears then to be based on a correct theory and to be perfectly safe for use in the design of all machine members of this general type.

8 In its present form it is a rather unwieldy instrument in the hands of a designer, but it may be made more applicable to design than might be thought at first. Upon examination it will be seen that the functions γ_1 and γ_2 are constants for all sections of similar form, that is, for all sections the proportions of which may be expressed as a function of some unit of dimension, for example, the radius of curva-

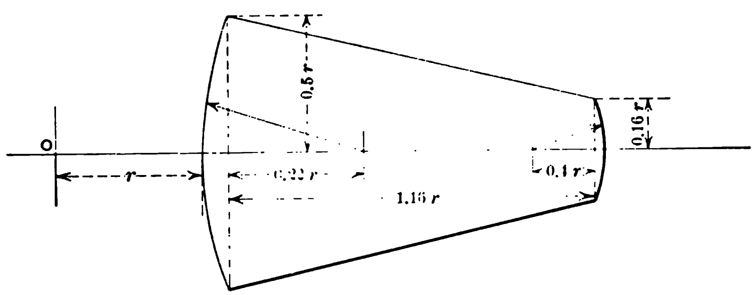

Fig. 3 Standard Hook Section

ture. Under the same circumstances the entire expression within the braces is a constant. The equation for a series of sizes and sections may therefore be written $f_t = \frac{W}{A} K$, or $A = \frac{W}{f_t} K$. The area is a function of the unit squared and therefore we may write $A = C'r^2$, or

$$r = \sqrt{\frac{K}{C'} \frac{W}{f_t}} = C\sqrt{\frac{W}{f_t}}$$

Applying this to the case of a series of hooks ranging from the minimum to the maximum to be manufactured, a standard form of section may be laid out as in Fig. 3, and the constant established. For the hooks tested by the writer the following values for the constant were found:

30-ton hook, cast steel	3.00
20-ton hook, cast steel	3.10
15-ton hook, cast steel	3.23
15-ton hook, wrought iron	4.29
10-ton hook, cast steel	3.49
10-ton hook, wrought iron	3.42
5-ton hook, cast steel	3.12
5-ton hook, wrought iron	3.12
3-ton hook, cast steel	3.78
2-ton hook, cast steel	3.74
Average	3.43

9 To make the case representative of present practice let such ratio of proportions be assigned to the section shown in Fig. 3 that $C = 3.4$. The design of a series of wrought-iron hooks to sustain loads of from 2 to 40 tons with a limiting intensity of tensile stress of 30,000 lb. per sq. in. will require the following computations:

$$\text{40-ton hook, } r = 3.4\sqrt{\frac{80000}{30000}} = 5.54$$

$$\text{30-ton hook, } r = 3.4\sqrt{\frac{60000}{30000}} = 4.7$$

$$\text{20-ton hook, } r = 3.4\sqrt{\frac{40000}{30000}} = 3.94$$

$$\text{10-ton hook, } r = 3.4\sqrt{\frac{20000}{30000}} = 2.76$$

$$\text{5-ton hook, } r = 3.4\sqrt{\frac{10000}{30000}} = 1.95$$

$$\text{2-ton hook, } r = 3.4\sqrt{\frac{4000}{30000}} = 1.23$$

10 The proportions obtained above will be for loads giving a maximum stress at the elastic limit of the material. For cast steel different values will necessarily be obtained. The establishment of such a standard would lead to a very simple process for the determination of the principal section of a hook for any capacity; the proportions of the shank and other parts of the hook may readily be established on the same basis. The bottom of the hook, being subjected to much wear, cannot of course be proportioned on the basis of the stress analysis. The above standard section selected as an average representative of present practice is not, however, the most economic form of section from the standpoint of equal maximum tensile and compressive stresses. It has been pointed out by Professor Pearson that a

section with such proportions is approximately an isosceles triangle with a radius of curvature of 1.75 of the height. The more nearly this form could be approached the less would be the weight of hook for the same capacity.

11 Professor Goodman[1] points out that for hook sections the functions γ_1 and γ_2 are expressed approximately as follows:

$$\gamma_2 = \frac{ke}{1.2\,\rho^2}$$

$$\gamma_1 = 1 + 1.1\,\gamma_2$$

where k = radius of gyration of the sections, the other symbols being as before noted.

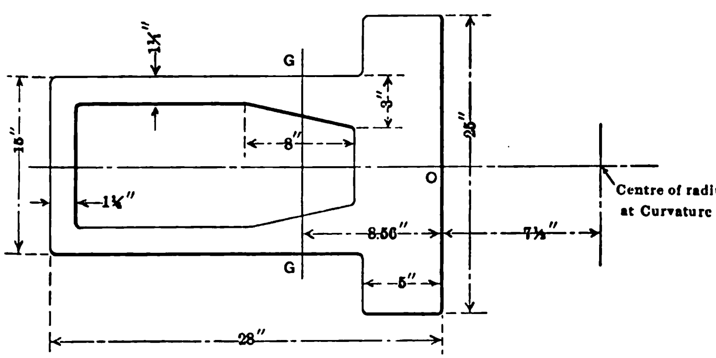

Fig. 4 Design for a Punch Frame

$A = 212 \quad I = 14{,}533 \quad k^2 = 68.56 \quad l = 38.56 \quad \rho = 16 \quad e = 8.56$

12 It will be found much more convenient to use these empirical expressions, which give quite accurate results, than to determine the value of the functions by the more tedious graphic method.

13 In applying these empirical formulae to punch and riveter frame sections the writer has found that the results are not accurate but that the values are better expressed as follows:

$$\gamma_2 = \frac{ke}{0.7\,\rho}$$

$$\gamma_1 = 1 + 1.1\,\gamma_2$$

For example, consider the design for a punch frame shown in Fig. 4.

[1] Institution of Civil Engineers, Proceedings, vol. 167.

Computing the values for the functions γ_1 and γ_2 by the graphic method, $\gamma_1 = 1.4$, $\gamma_2 = 0.405$. Whereupon the intensity of stress according to the new method of analysis for a force of 90,000 lb. at the punch will be

$$f_t = \frac{W}{A}\left\{\frac{l}{\rho\gamma_2}\left(\frac{1}{\left(1-\frac{e}{\rho}\right)^{\frac{5}{4}}}-\gamma_1\right)+1\right\} = 8500 \text{ lb. per sq. in.}$$

The values of γ_1 and γ_2 by the empirical formula are 1.44 and 0.4 respectively. Whereupon the intensity of stress becomes $f_t = 8500$ approximately. According to the old formula used almost exclusively in textbooks, the value of f_t is expressed by $\frac{W}{A}+\frac{Wle}{I}$, whence $f_t = 2450$.

14 The above empirical formulae are derived from the results of computation of two sections. I am not prepared to state that they will work out in all cases and must therefore caution anyone against using these values to check the results by the graphic method. It may be clearly seen that were the punch in question designed for a limiting intensity of stress of 2450 by the old formula, there would actually be a maximum stress of 8500 lb. per sq. in., which is hardly a safe value for cast iron and particularly for a large casting.

DISCUSSION

Prof. Gaetano Lanza. A careful perusal of the articles of Messrs. Pearson and Andrews in The Draper Company's Research Memoirs, containing the formulae referred to by the author, reveals no flaw in the deduction of the formula for the greatest tensile stress at the section of greatest bending moment, provided it is regarded as a formula which gives the relation between the load on the hook and the tensile stress mentioned, and provided the section of greatest bending moment remains plane.

2 To determine in all cases, however, the relation between the load corresponding to a greatest stress at the above stated section, equal to the tensile elastic limit, and the elastic limit as determined by the methods of measurement employed, would, in my opinion, require a set of tests upon a series of hooks varying in their proportions to a much greater extent than those mentioned by Professor Rautenstrauch, in which the formula of Professor Pearson would make the

two loads cited nearly equal. An example of such a case, in which this result does not hold, is a set of hooks tested under the direction of Prof. C. E. Fuller, which were really open links of circular form, made by bending hot and annealing square bars, the side of the square being 0.75 in., where $\rho_0 = 3$ in., and where the load at the elastic limit, as determined by a method similar to that employed by Professor Rautenstrauch, was 1100 lb.

3 For these hooks we should have $\gamma_1 = 1.0074$ and $\gamma_2 = 0.00658$.

4 The greatest tensile fibre stress at the section of greatest bending moment, if computed by the ordinary formula, would be 48,600 lb. per sq. in. and, if computed by the theory of Messrs. Andrews and Pearson, would be 59,300 lb. per sq. in., whereas the tensile elastic limit of the material was 30,000 lb. per sq. in.

5 In seeking an explanation of these apparently discordant facts the following observations should be kept in mind:

- *a* In a straight beam we would naturally expect the elastic limit as determined by measuring deflections to be greater than that corresponding to a greatest fibre stress equal to the tensile elastic limit, the excess varying with the span.
- *b* The methods used in all the experiments cited have been practically the measurement of deflections.
- *c* The deflections, whether of beam or hook, cannot be determined by computation from the stresses at the section of greatest bending moment only, but depend also upon the stresses at the other sections.
- *d* In the hooks tested by Professor Rautenstrauch the section of the hook is a varying one in which the stresses at sections other than that of greatest bending moment have not been examined.

Hence it seems to me that before we can consider that a complete solution of this problem has been attained, we need

- *a* A more extended series of tests which shall include a considerable number of hooks of each kind.
- *b* An experimental determination, both for beams and hooks, of the relations between the elastic limit as determined by deflections, and the load corresponding to greatest fibre stress equal to the tensile, or compressive, elastic limits, in the case of varying spans and other proportions.

Chas. R. Gabriel. The results of tests of crane hooks and the figures obtained by the old and new formulae, to which Professor Rautenstrauch calls attention, are very important as regards crane hooks and similar members of machines. If such members are not as strong as computed by the usual formula for combined bending and tension it is none too soon for engineers to be made acquainted with the fact. This is especially so because of the fact that metal beams of solid cross section, similar to the cross section of a crane hook when subjected to simple bending, show greater strength than that due to computation, at least when subjected to a breaking test. This excess of strength is so great, especially in beams of cast iron of certain cross sections, as to justify confidence in lesser dimensions for straight beam members than those that would be prescribed by calculation based on the tension and compression moments of beam sections. Similar excess over calculation, of ultimate breaking resistance by test, exists in shafts subjected to torsion.

2 One would naturally expect to find a similar excess of strength in crane hooks when put to test, but the results to which consideration is invited show quite the reverse to be the case, and are none the less valuable because disappointing.

3 As regards machine members, such as the overhung frames of presses, punching and shearing machines, etc., the large majority of such frames require to be rigid under their working loads, to an extent that renders them perfectly safe from failure by breaking. A great many points have to be considered with respect to dies being thrown out of line by the springing apart of the upper and lower arms of the frames. A small amount of such deflection would in some cases be sufficient to cause the shearing of expensive punches by the dies, rendering them unfit for the accurate work intended. In some few other cases, such as riveting, a comparatively large amount of deflection is permissible, and in some instances the proportions of a frame may be considered with respect to safety from rupture alone.

4 The cross sections of overhung frames must of necessity differ a great deal in different machines, also the relative amount of overhang or throat, depth of gap and general form of frame, whether curved similar to a crane hook, or extending straight up and down comparatively short or long distances. Various kinds of cross section such as solid rectangular, T, H, box or combination of box and rib, all have their appropriate uses. The successful designer has at times to depart considerably from formulae that have been in use and must combine much practical judgment and observation in his work.

Factors of safety must vary from 3 to 50 or more, and stresses accordingly.

5 It is hardly to be expected that a formula for strength of crane hooks can be immediately applicable to all the various cases of overhung machine frames, but we judge it might be applicable to small frames of solid section and short overhang. Frames having a long overhang, such as that represented by Fig. 1 in the paper, would in our opinion be a more trustworthy subject for the application of the useful bending formula than frames having relatively a much shorter overhang, such as that indicated by the dimensions in Fig. 4. This is because the greater the overhang the more significant becomes the simple bending moment and the less significant the direct tension in the back of the frame.

6 Referring to Fig. 4, it is noticeable that the metal in the back of the frame is very thin. In frames where rigidity is the prime consideration, we believe it is a common error of designers when using cast iron to place too little material in the back. This no doubt arises from the known high compressive resistance of cast iron, without regard to its elasticity under compression; frames being designed, accordingly, with regard to resistance to breaking rather than with regard to resistance to deflection. We have known of many cases where frames could be greatly stiffened merely by taking metal from the front web and putting it on the back web.

George R. Henderson. That we get a rather greater strength than would be expected from the Unwin formula, especially in the case of hooks, agrees with my practical experience. A few years ago we purchased some 60-ton cranes, and when it came to the detail of the hook to lift the 60 tons, the design submitted by the manufacturers was for a hook smaller than we thought good practice would accept. We calculated to reduce the total strain due to the vertical stress and the bending moment to about 12,000 lb., which we considered would give a factor of safety of five with the material used. It was pointed out that the hook did not conform to the specifications, and that a larger hook was desired. These larger hooks were provided and they looked gigantic.

2 A little later the question came up again, when the manufacturers stood on their dignity and claimed that the hook was stronger than my calculations showed, and to confirm their case referred to tests at the Watertown Arsenal, which we all consider pretty good authority. The hook tested was rated as a 20-ton hook, but it had

been subjected to a weight of 162,000 lb., at which it merely bent but did not break.

3 These tests were to determine the ultimate strength, whereas the paper deals with the elastic limit; but practically, I think, the ultimate strength interests us as much as the elastic limit. By the regular Unwin formula, which has been somewhat condemned, the stress per sq. in. in the hook, when weighted to 162,000 lb., at which it simply opened, would indicate 142,000 lb. per sq. in. fibre stress, which of course is absurd. So, from the actual tests, it is very evident that the hooks are considerably stronger than the Unwin formula could indicate. In discussing this matter with well-known machinery builders, we found that while the strain on the hooks might figure at 17,000 lb. per sq. in. from the formula, and show a factor of safety of only three, actually the factor of safety must have been five or six.

4 If possible, I would like to know how the author can reconcile these facts, with the practical ultimate strength tests in connection with the elastic limit.

PROF. WM. H. BURR. Professor Rautenstrauch has added a very interesting chapter to the literature of this subject, but there is perhaps a little more to the matter than has been indicated, and it bears a good deal upon what has been said by the last speaker. Doubtless the analysis based upon Professor Pearson's paper, as an analysis, is a decided improvement upon the Unwin formula, but again there comes in the same question raised in connection with reinforced-concrete beams. This analysis, whether by Professor Pearson or Professor Unwin, is based upon what is ordinarily known as the common theory of flexure, which belongs accurately only to straight beams of very small depth in comparison with the length.

2 Hooks and all such members as those shown by the author are exceedingly short as beams, and they are also curved. These conditions completely demoralize the analysis as based on the common theory of flexure, and it is not a matter of surprise that hooks should show so much greater carrying power than the computations would indicate. In fact, it is precisely in line with what we find in other short beams.

3 The pins at the panel points of pin-connected bridges are designed by the common theory of flexure. Yet if one should compute the extreme fibre stresses in those pins at some panel points as they have existed, they would be found to run up not only to

142,000 lb. per sq. in., but to 180,000 or 190,000 lb. in structural steel. A partial explanation lies in the fact that an analysis is used, which, strictly speaking, does not apply to these conditions. The hook and all such members, as well as bridge pins, are short, thick beams to which the usual theory of bending does not strictly apply.

4 Again, one will find that in bridge specifications, the regular working fibre stresses in pins are permitted to be at least 50 per cent greater than in the tension members of the truss; that is, one may have a working stress of perhaps 14,000 lb. in bars, and a fibre stress in tension of 18,000 or 20,000 lb., sometimes even 24,000 lb. in pins. This is due to a fact I have already mentioned, that as a matter of accurate analysis, the common theory of flexure should not be used in connection with such members; but there is nothing else to be done.

5 That again brings me back to the same point made in connection with concrete beams. The proper procedure is to settle upon some sensible working formula, just as we do in connection with the pins in bridges, make tests of such members, and deduce from these tests such empirical quantities as may be properly used in the formula, so as to make the results of the analysis in that way conform to safe and sensible practice.

A. L. Campbell.[1] Table 2 of Professor Rautenstrauch's contribution shows an excellent agreement between actual test conditions and the results obtained by the formula which is the basis of his discussion.

2 A much simpler formula is used by the writer for similar computations. A crane hook or the frame for a punch is really a tension member with an exaggerated eccentric load. The maximum unit stress in such a tension member may be proved equal to

$$f_t = \frac{W}{A}\left(1 + \frac{l(1-e)}{R^2}\right)$$

using the author's notations. The radius of gyration, R, is equal to $\sqrt{\frac{I}{A}}$. Applying this formula to the frame shown in Fig. 4 gives $f_t = 7600$ lb. per sq. in. This stress is 90 per cent of that given by the more complex formula.

[1]The Solvay Process Co., Detroit, Mich.

Frank I. Ellis. While the paper, together with the article in the American Machinist to which it refers, covers very fully the design of hooks, giving results which agree remarkably with actual tests, its application to shear housings is not quite clear.

2 We note primarily, that in the derivation of his formula the writer has assumed the entire area to be in tension, i. e., the neutral axis to lie entirely without the section. While this condition is almost universally correct in hooks, it will seldom be encountered in shear housings, but still it appears to have important bearing on the form of the equations.

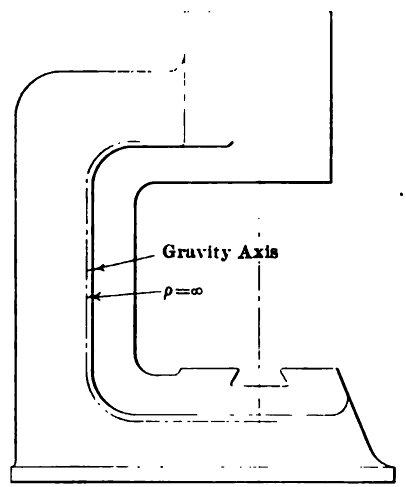

Fig. 1 Frame with Infinite Radius

3 Another point which is not quite clear, but is a matter of great importance, is the assumption of the value of ρ, the radius of curvature of the gravity axis of the section. In the case of a hook, this of course is obvious, but in machine members, such as shear housings, this seems far from being the case. For instance, in a housing of the general form of sketch shown in Fig. 1 herewith, we would have an infinite value of ρ. This would reduce the formula to a case of simple tension, which is obviously incorrect, giving stresses that would be very much less than would be obtained by actual test. On the other hand, if we consider an extreme case as in Fig. 2, where the value of ρ is very small, the stress as calculated by the formula

would be very much in excess of what could possibly exist in the actual casting.

4 The example of a shear housing which Professor Rautenstrauch has chosen as an illustration appears to be at variance with our experience. The stress calculated by the new formula is almost three times that obtained by the usual methods of computation. In our experience, cast-iron shear housings in which the calculated stress is 3,000 lb. per sq. in., never break except through defects in the casting, a condition which could hardly exist if the actual stress were in the vicinity of 9,000 lb.

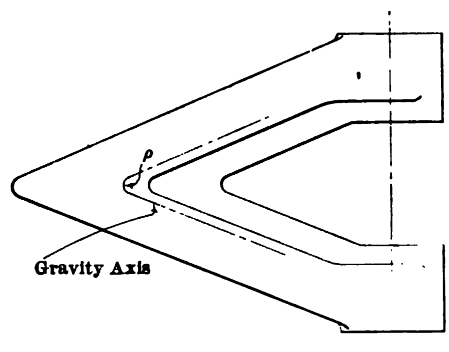

Fig. 2 Frame with Small Radius

E. J. Loring.[1] The figures in the paper show such striking discrepancies from these obtained by the usual methods of calculation that this analysis of the problem merits the most careful consideration. These results clearly show that the stresses, and particularly the maximum stress in a curved piece under the combined direct and bending load to which such hooks and gap frames are subjected, cannot properly be deduced from the simple combination of direct and bending stresses as determined by ordinary analysis from the stresses in a single plane, but may be influenced to a greater extent by conditions outside of the section plane, such as the relations connecting that plane with those nearby on either side.

2 This difference between straight and curved members arises from a different distribution of stress due to the variation of length of fibres at different parts of the section as taken between similar adjacent sections.

[1] Loring Speed Gauge Co., 76 Highland Ave., Somerville, Mass.

3 The usual deduction for stress in straight members commonly applied to this problem assumes that:

a Planes remain planes after bending.
b Strain is proportional to the distance from the neutral axis.
c Stress is proportional to strain and therefore that the stress is proportional to the distance from the neutral axis.

4 The assumption (*c*) that stress is proportional to strain is true only as referring to unit strain, as long fibres will yield more under a given stress than shorter ones. In the case of straight members the adjacent minimum sections are parallel and the elementary fibres therefore all of equal length, and the assumption may be applied. In the case of a curved member, which I would define as one in which the locus of the centers of gravity of the minimum cross sections is a curved line, these sections are not parallel, but radiate from a center of curvature so that the fibres are not of the same length throughout the section, and a correction must be made for the variation of the length of fibre before this assumption can be applied. This point has generally been overlooked or considered negligible, and in this is to be found the explanation of the difference in results. I might add that this exemplifies the danger of applying a formula to conditions which it was not intended to represent.

5 I am not certain that I can agree with the author in the use of the theory of lateral contraction in the analysis. I cannot at this moment see why it is any more necessary in the case of the hooks tested than, for example, in the case of the test bars from which he deduced the fibre stresses. Taking only the common assumptions, with the correction for the length of fibre, as above noted, it is possible to obtain results in very close agreement with those given in the formula recommended by Professor Rautenstrauch. In place of the usual straight-line diagram of stress on the section these assumptions give the stress at any point as varying according to

$$\frac{y}{1+\frac{y}{\rho}} \quad \text{or} \quad \frac{y\rho}{\rho+y} \qquad \text{(See Fig. 1)}$$

using the symbols of the paper, and from this may be determined the important fact that for the case represented by the hooks, where the line of application of the load contains the center of curvature, the

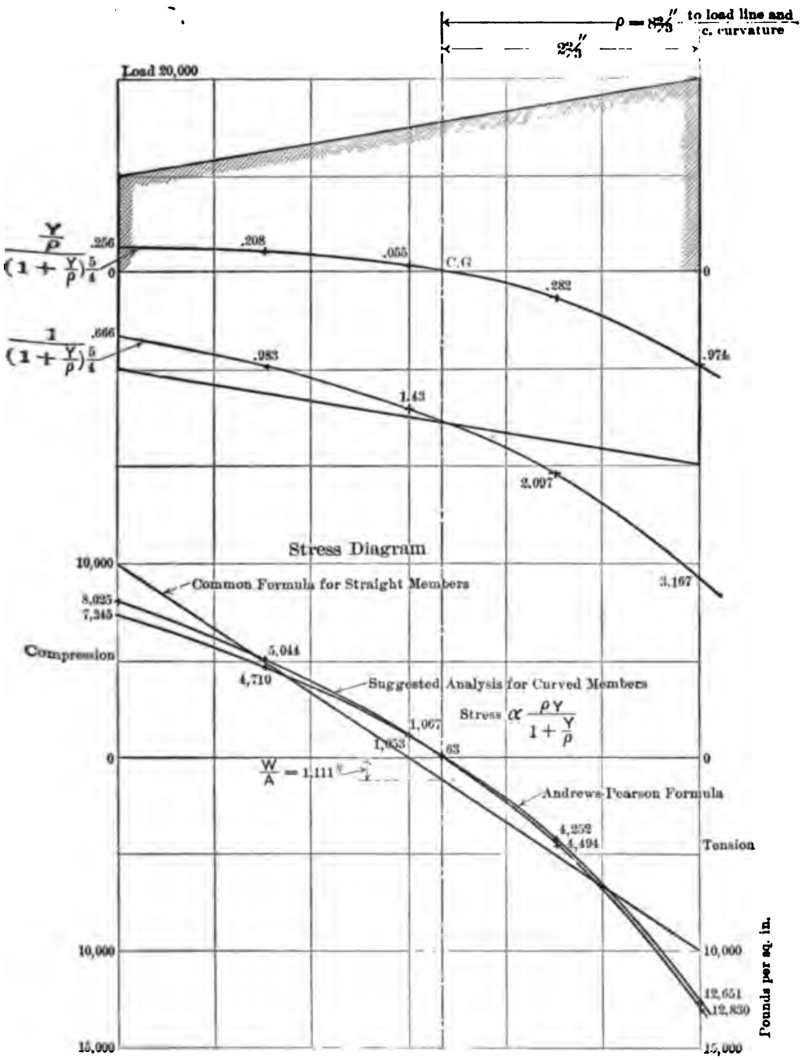

Fig. 1 Diagrams for Trapezoidal Sections of Straight and Curved Members for Equal Intensity of Stress

neutral axis contains the center of gravity. In other words, instead of the stress at the gravity axis being equal to the distributed stress as is true for straight members, the stress at this point in a member with this degree of curvature is zero, and this represents the manner in which the stress "piles up" toward the inner edge of a curved member. This condition of stress at the center of gravity would be represented in the analysis of the paper by the condition $\gamma_1 = 1 + \gamma_2$. The empirical formulae recommended give $\gamma_1 = 1 + 1.1\ \gamma_2$ and the data on the hooks give a variation from $\gamma_1 = 1 + 1.17\,\gamma_2$ to $\gamma_1 = 1 + 0.88\,\gamma_2$ with an average of $\gamma_1 = 1 + 1.015\ \gamma_2$ so that it will be seen that this is approximately true by Professor Rautenstrauch's analysis; that the gravity axis is the neutral axis for this degree of curvature just as it is for transversely loaded beams.

6 I have applied the variation of stress

$$\frac{y}{1 + \frac{y}{\rho}}$$

given above to the solution of an assumed section and find that the stresses and their manner of variation are substantially identical, for this particular case at least, with those given by the author's method. I believe that an analysis can be made along this line that will give results very close to those shown and be more generally workable. The differential expressions for the net stress on the section and the moment of the stress are similar to those for a beam with the addition of a factor

$$\frac{1}{1 + \frac{y}{\rho}}$$

$$W \text{ varies as } \Sigma \frac{y\,\mathrm{d}A}{1 + \frac{y}{\rho}} \qquad Wl \text{ varies as } \Sigma \frac{y^2\,\mathrm{d}A}{1 + \frac{y}{\rho}}$$

It may perhaps be possible to deduce some general expression to be used as a factor of correction for curvature to be used with the usual methods.

7 The effect of the curvature is less, the greater the ratio of radius of curvature to the depth of section. In the case of hooks this means greater strength where the contour of the inner edge is elliptical

instead of circular, so that the curvature at the most strained section is less. As the curvature tends to "pile up" the stress toward the inner edge, greater strength may be had by giving the hook a closer approximation to a T-section, by which means the metal is massed better where the stresses are abnormally high. It would also appear that a high gap is stronger than a low one for the same depth, since a lesser degree of curvature is possible.

8 I must disagree with the statements in Par. 2 except as limited to curved members; also with the statement in Par. 8 that γ_1 and γ_2 are constants for all sections of similar form, except it be modified to say "of similar form, curvature and load distance."

9 In determining the maximum stress by the method which the author has proposed, the function γ_2 is the most important factor, and this function is obtained from the difference in area of two derived curves; the difference is small and the less the difference the greater the maximum stress. It would seem that there is great opportunity for inaccuracy in determining this factor. It also appears to me to be simpler to take

$$\gamma_2 = \frac{-\Sigma\left(\dfrac{1}{1+\dfrac{y}{\rho}}\right)^{\frac{5}{4}}\dfrac{y}{\rho}\,\mathrm{d}A}{A}$$

as originally stated, for the purpose of the computation, rather than to use the value derived from it, of

$$\gamma_2 = \gamma_1 - \frac{\Sigma\left(1+\dfrac{y}{\rho}\right)\,\mathrm{d}A}{A}$$

for the reason that having the quantities for the determination of γ_1 for various points of the section, that is, the values of $\left(1+\dfrac{y}{\rho}\right)^{\frac{1}{4}}$, it will be simpler merely to multiply these by the respective distances from the gravity axis, plot the curve and integrate for the net area, rather than to proceed by raising the denominator to a new power and passing through all the processes anew.

10 It is stated that the standard section selected for the computation of constants for the empirical formula is not the most economic from the standpoint of equal tension and compression stresses. This is

true even if the member is straight, in which case, considering the trapezoid only and omitting the curved ends, the maximum stress in compression is 85 per cent of the maximum stress in tension. All other parts remaining the same, for equal intensities of stress in the edges for a straight member, the half width of the narrow edge should be 0.095 r, as may be very readily demonstrated. The geometrical relations for the correct proportions of a trapezoidal section for equal intensity of stress in a straight member are so exceedingly simple that I want to give them here, particularly since, so far as I know, they have never been published. This relation is that the sides extended intersect at a distance from the far or narrow edge equal to the distance of the load line from the near or wide edge, and for the solution of this case we have

$$d = \sqrt[3]{6\,ky\frac{F}{f}}$$

where d = depth of section
y = distance of load line from the near edge
F = load
f = maximum stress at the near edge or far edge (equal)
and k is a design constant = ratio of depth of section to width of far edge.

11 For the case of equal stresses in a curved member of trapezoidal section with center of curvature on the load line, a similar relation may be deduced from the analysis that I have here suggested, but is not quite so simple: the point of intersection of the sides is given by the following construction: Lay off on the axis of symmetry and toward the far edge a distance from the *near* edge equal to the distance from the near edge to the center of curvature and load line. If this distance is greater than the depth of section, equal stresses may be had. If this distance is equal to the depth of section, i. e., if the point thus laid off is on the far edge, equal stresses require a triangle with this point as the apex. If the point is beyond the far edge, divide the distance to that edge in thirds; then the stresses are equal when the sides extended intersect at the nearer point of division, one-third of this distance from the far edge.

12 It will be noticed that this construction gives the radius of curvature for this limiting case equal to 1.33 times the depth of section[1] instead of 1.75 as given by Professor Pearson. I have investi-

[1] Or depth of section equal to gap depth.

gated this case for both degrees of curvature by the method involving the lateral contraction and find that using the formulae given by Professor Rautenstrauch the curvature of 1.33 times the depth, measured to the gravity axis, gives a stress on the inner edge of 1.091 times that on the far edge. A similar operation for curvature of 1.75 as recommended by Professor Pearson, by his own method gives by my computations a ratio of stress of 0.912. A sharp triangular section such as this is, however, of little or no importance in actual construction, and the method of determining the proportions which I have given will, I think, be found to be of much more general application. I am unable to state at the present time whether a section having equal intensity of stress on the two edges is or is not the most economical of material; but presumably it is.[1]

PROF. C. E. HOUGHTON. The agreement between the elastic limit as calculated by the proposed formula and that as derived from the tests is, to say the least, wonderfully close, and the wide variation between the experimental values and those calculated by the use of a theory that has been in common use for many years leads one to ask "Why are there not more failures in crane hooks?"

2 Objection has been made to the tests because the hooks were not loaded beyond the elastic limit. This seems to the writer to be a mistake. What the engineer is mostly interested in is the effect of loads that produce stresses within the elastic limit, since the great majority of the formulae used for the calculation of stresses are based on theory that no longer holds true after the elastic limit has been exceeded.

3 Professor Burr has pointed out that the simple theory of flexure does not apply to curved members and Mr. Gabriel notes that stiffness and not strength is the controlling factor in many of the open-side machine frames. May not the fact that cast iron is used in the majority of such frames be another reason why the flexure formulae cannot be expected to give correct results? The well-known fact that the physical properties of any cast iron vary with the rate of cooling, and that the tensile strength and modulus of elasticity are not constant at all depths from the surface of any cast-iron member, but vary throughout any given section, leads one to ask "Is it not more reasonable to use the simpler formulae in the calculations for

[1]Since writing the foregoing, Mr. Loring has found that the method suggested by him for the determination of the stresses—or a very similar one —is given in some detail in Hütte, from some German source dating 1902.

strength and to provide against possible errors by that useful and elastic term—the factor of safety?"

H. Gansslen.[1] The author's tests prove the correctness of Andrews and Pearson's new formula for figuring crane and coupling hooks. All the experimenters, however, seem to have limited themselves to these hooks, for which the formula appears to have been gotten out. Hook's law of the direct proportionality between stresses and strains also underlies the new formula and the fact that this law holds practically good on wrought-iron, steel and similar materials would to some extent explain the good agreement of the results of tests and calculations by means of the new formula.

2 The author points out that the formula is applicable to punch and riveter frames. To generalize thus I believe is hardly wise at present, as all the various formulae for figuring curved beams are more or less empirical and each of them is naturally proved to be true for a certain limited field of calculations only. Hook's law does not hold true for copper, cast-iron, bronze, stones, artificial and natural, etc., and this law giving the modulus of elasticity as constant is the basis of the formula.

3 Engineers know that the old formula for figuring a curved member in the same way as a straight beam gives factors of safety too small, but that we are now underestimating the stresses in the throat of punch press frames $8500 \div 2450 = 3\frac{1}{2}$ times is surely saying much.

4 However, there is no use disputing the new formula in so far as tests have verified it and it is to be hoped that the author will have the opportunity of entering other fields of research besides that of crane hooks. That of press frames would be a desirable one.

5 I have not come across a case where a punch press frame figured in the usual, but wrong, way could have been $3\frac{1}{2}$ times underestimated, roughly considered by comparing the pressure exerted with the general behavior of the frame.

6 The old theory of flexure as applied to and compared with tests of cast iron has shown its inapplicability and this should make us all the more cautious in adopting the new formula for cast-iron press frames before having on hand the results of tests that would justify us in so doing.

John S. Myers.[2] The author's presentation on the design of curved machine members and his article in the American Machinist

[1] Mechanical Engineer, 404 Fisher Bldg., Chicago.

[2] John S. Myers, 2456 Almond St., Philadelphia.

of October 7, 1909 dealing exclusively with crane hooks, seem to indicate that the new theory is applicable to punch and riveter frames of the type shown in Fig. 1, where the throat is semi-circular, being struck with a radius having its center at O. In order, however, to find the radius of curvature of the gravity axis of the principal section it would seem necessary to plot points such as A, B, C, D, E, draw a curve through them, then, by trial, find the center O' of a circular arc which will pass through C and most nearly fit the curve for points intermediate between B and D.

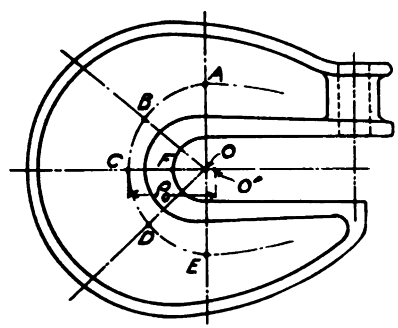

FIG. 1 FRAME WITH SEMI-CIRCULAR THROAT

Curve $ABCD$ represents the gravity axis of the section. Point O is the center of the throat radius. Point O' is the center of a circular arc which approximately coincides with the gravity-axis curve for points between B and D.

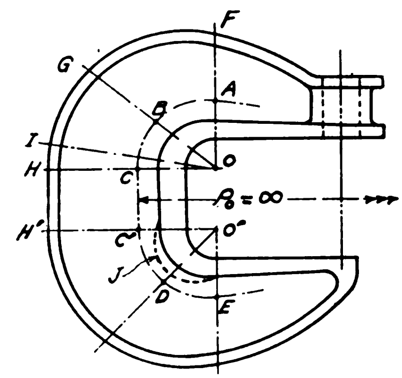

FIG. 2 FRAME WITH WIDER GAP THAN FIG. 1

Curve $ABCC'DE$ represents the gravity axis. Between points C and C' this curve becomes a straight line; hence $\rho_0 = \infty$

2 If the above is consistent with the assumptions upon which the theory is based, it will be seen that the point O' is not necessarily coincident with O, and that to find the value of ρ_0 a layout must be made and the gravity axis of several sections determined. It is also seen that ρ_0 is not strictly a function of the throat radius nor is it equal to $OF + CF$ as one would at first suppose. This adds more complication to the problem, which is already vexatious.

3 Again, such frames are not always made with the throat struck with a single radius; in fact, this is the exception rather than the rule for a large class of machines, which have a wider "gap" to accommodate the work and are more like that shown in Fig. 2. Here the curve representing the gravity axis is a straight line between

points C and C', in consequence of which $\rho_0 = \infty$ and it would therefore seem that the new theory did not apply to this portion of the frame. Now, if this be the case, and we design that portion of the frame between OH and $O'H'$ according to the old theory of straight beams, but design section OI according to the theory of curved beams under discussion, it would appear from an inspection of the results given by Professor Rautenstrauch that section OI should have about three or four times the flange area of section OH. Of course the metal at the corners could be thickened, as indicated by the dotted line at J, but it would be out of the question to double or treble the usual flange thickness, which is what the new theory seems to indicate as necessary.

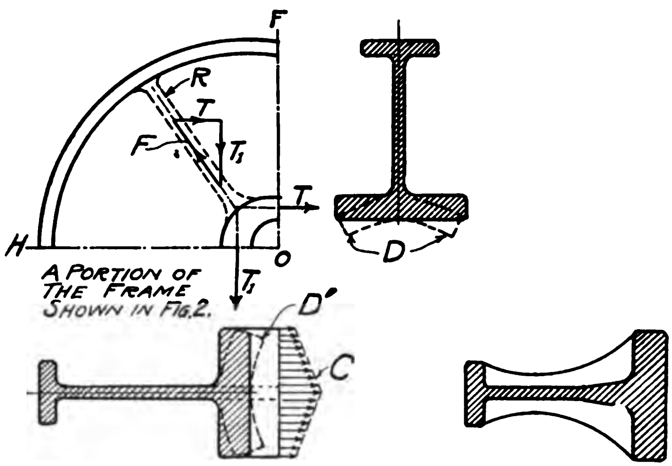

FIG. 3 FIG. 3a

FIG. 3 SHOWING HOW THE RAPID TRANSITION OF STRESSES INDUCES LOCAL STRESSES. FIG 3a. PROPOSED SECTION

4 It would be very interesting to know how the new theory could be properly applied in such a case; whether, for instance, it is entirely applicable at the section OG but gradually merges into the old theory at sections OF and OH; or whether it has not, as yet, been sufficiently developed to be generally applicable to sections other than those at right angles to the line of action of the force.

5 Generally speaking, a structural engineer never puts in curved tension or compression members because he knows that force either travels in straight lines or else produces bending strains; but the average designer of machinery seems to delight in curved ribs, bent levers, and the like. The average mechanical draftsman makes

layouts as if he held the opinion that force travels along a curved rib in a manner somewhat similar to water flowing in a pipe and that it will, therefore, follow any devious or sinuous course in which he may choose to distribute the metals. Most C-frames seem to be designed on the foregoing assumption and, while it is an exceedingly difficult piece of mental gymnastics to follow the mathematics of the new theory, it is, however, quite easy to see that there are stresses induced in curved ribs which are usually ignored.

6 To illustrate the foregoing, Fig. 3 shows that portion of the frame of Fig. 2 which lies between lines OF and OH. Now, let T and T_1 represent the total tensions in the flanges on sections OF and

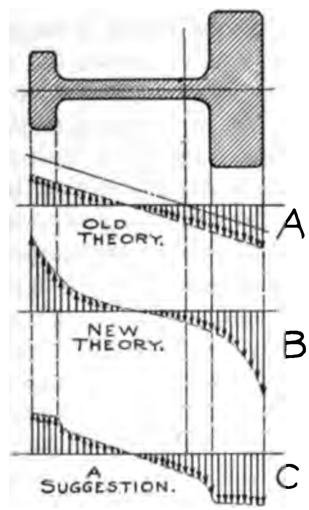

Fig. 4 Distribution of Stresses Under Different Theories

OH respectively. By combining T and T_1 graphically it is seen that a resultant force, F, must, in some manner, be supplied to establish equilibrium. The most direct way of supplying such a force is by the addition of a rib as indicated by the dotted lines at R which will distribute part of F into the web and deliver part of the force at the compression flange where there is a smaller, opposing resultant force. In the absence of any such rib the necessary force must be supplied by the web, partly through a local bending and distortion of the flanges as indicated by the dotted lines at D and D' and partly by a concentration of stresses towards the central portion of the flanges as indicated at C, this concentration being a direct result of the deformation at D'.

7 In supplying a rib R, if it was intended to carry the entire force F it would be necessary to make it about $1\frac{3}{4}$ times the average thickness of the flanges, but since the web can readily take half, or more than half, of the load, it would seem that a rib of $\frac{5}{8}$ or $\frac{1}{2}$ of the flange thickness, narrowed down at the center as shown in Fig. 3a would be entirely sufficient, especially if the web be judiciously thickened and liberal fillets used.

8 As I understand the new theory it does not recognize any such concentration of stresses as indicated at C in Fig. 3 but, on the contrary, assumes a more rapid concentration towards the extreme fibres in a manner somewhat similar to that shown at B in Fig. 4. Now in view of the close accord between the new theory and the results of Professor Rautenstrauch's experiments, I am quite ready to believe that diagram A represents quite closely the actual conditions for straight beams of solid section, and that diagram B represents the most plausible theory for curved beams of solid section; but that for beams composed of heavy flanges and a light web the probable distribution of stresses is more nearly like that suggested by diagram C, and that so far as the curved form of the beam is concerned, it is not the curve of the neutral axis we are interested in but the curve of the flanges, and that this results in local bending and concentration of the stresses as already pointed out.

9 I have no well formulated theory to advance in explanation of my belief in a distribution of stresses like that indicated by diagram C but have sufficient faith in it to calculate sections of this nature by the very simple process of considering the stress to be uniformly distributed over the flange area and entirely neglecting the web; then at points where there is rapid transition of stresses, supplying ribs, thickening up the web and allowing a lower flange stress and liberal fillets. This procedure may sound crude to a scientific man, but it has, at least, ease of application in its favor and may yet be shown to be actually more scientific than the more laborious methods usually pursued. As yet, I have not had the temerity to apply this method to large work but would like to have the opinion of those who have had experience along these lines.

THE AUTHOR. The test reported by Professor Lanza is interesting, but I do not feel justified in replying without a review of the entire data on the experiment. The point made by him in Par. 5 in regard to deflections, is somewhat misleading. I did not propose in my experiments to determine the relation of total deflections to the maximum stress in the hook, but rather to find the load at which the total deflection ceased to follow the straight-line law. Since the total deflection is dependent on the deflection of all the sections, it is rational to suppose that when any variations occur they are due to the fact that the "fibres" in the most strained section have been stressed beyond the elastic limit. This is all we wish to know. The most strained section is without doubt the main horizontal section. The examination of the bending moments in other sections is of no value in these determinations.

2 Referring to Mr. Gabriel's remarks: I regret that so many designers persist in applying the formulae for determining maximum intensity of stress beyond their limits of application. No computations can be made to determine *ultimate breaking strength* and I see no reason why anyone should be surprised that there is a disagreement between the "results of computations" and the results of test. I did not choose to consider the matter of rigidity, which the title of the paper would lead one to believe should be included. Rigidity is, of course, a controlling factor in die work.

3 Mr. Henderson's remark that his practical experience with hooks leads him to believe that a rather greater strength exists than can be expected from the Unwin formula, qualified by his report of certain tests, would lead one to believe that he has made use of Unwin's formula outside of its field of application. Unwin's formula indicates nothing beyond the elastic limit. There exists no method of analysis which enables us to determine the relation between the load on the hook and the resulting maximum intensity of stress when that stress is beyond the elastic limit of the material. In reply to the statement that "the ultimate strength interests us just as much as the elastic limit," I would say that I believe designers will be treading on much safer ground when they confine themselves to proportioning parts with a factor of safety based on the elastic limit rather than the ultimate strength.

4 Mr. Ellis says in the second paragraph "We note primarily that in the derivation of his formula the writer has assumed the entire area to be in tension, i. e., the neutral axis to lie entirely without the section. While this condition is almost universally correct in hooks, it will seldom be encountered in shear housings." No such assumption is made, nor is it universally correct in hooks. I believe that Mr. Ellis is also mistaken in his remarks on the particular form of the equation when ρ is infinite. When ρ is infinite the case is not that of simple tension but rather as expressed by Unwin's formula.

5 Mr. Loring's explanation of the two analyses, I regret to say is incorrect. Both analyses are founded on a determination of the relation between *unit* stretch and intensity of stress, but the real difference is found in the methods of evaluating the unit stretch. The older formula gives the *unit* stretch as

$$\lambda_y = \lambda_\beta + \frac{y'}{\rho'}$$

while the newer analysis gives

$$\lambda_y = \lambda_\beta + \frac{\frac{y'}{\rho'} - \frac{y_0}{\rho_0}}{1 + \frac{y_0}{\rho_0}}$$

where

λ_y = unit stretch of any fiber a distance y' from the gravity axis.
λ_β = unit stretch at gravity axis.
ρ' = radius of curvature at gravity axis after stretching.
ρ_0 = same before stretching.
y_0 = modified y' after stretching.

The newer analysis retains terms of the same order of magnitude as λ_y and therein lies the difference. The theory of lateral contraction is rationally applied in this analysis, its application being unnecessary to the test piece, since direct measurement of stress is made.

6 Par. 2 in the paper is obviously limited to curved members. The similar form referred to in Par. 8, includes the radius of curvature. The method in the paper for determining γ_1 and γ_2, I believe will be found more convenient than those proposed by Mr. Loring.

7 Professor Houghton will agree with me that a more correct analysis for straining action will permit a more intelligent use of the factor of safety.

8 Mr. Myers is quite correct in his remarks on the value of ρ_0. The analysis, however, does apply to the case of straight beams where $\rho_0 = \infty$, for which case it reduces to the form of the Unwin formula. The formula has not as yet been sufficiently developed to determine its usefulness in establishing proportions for other than those sections at right angles to the load. The difficulty of determining the stretch on sections at an angle to the load will leave this problem unsolved for some time. It is, however, rational to suppose that the flange on oblique sections should be thickened, but to what

extent has not yet been determined. In regard to the behavior of a T-section, I would state that Professor Pearson has found experimentally that it is subjected to the same laws as a solid section. This indicates that the suggestion of Mr. Myers in Fig. 4 can hardly be accepted.

9 I judge from Professor Burr's remarks that he discredits the analysis by Professor Pearson on the basis that it is founded on the common theory of flexure, that is, it is not applicable to beams of very great depth compared with the length. I believe that if Professor Burr had given more thought to the matter he would not have made this statement. In view of the experimental results obtained by myself and others in verification of the theory and the lack of any data in verification of Professor Burr's statement, I am still inclined to believe that Professor Pearson's analysis is correct.

No. 1253

TESTS ON A VENTURI METER FOR BOILER FEED

By Prof. C. M. Allen, Worcester, Mass.
Member of the Society

A reliable and accurate hot-water meter has been in demand for a good many years. The principle of most of the cold-water meters, where there are moving parts in the water, is not at all adaptable for hot-water work. A hot-water meter for boiler-feed purposes must stand not only the variation in temperature but also considerable variation in pressure, and quite often it has to stand a certain amount of water-hammer, this depending somewhat upon the style and condition of pump used. The Venturi meter, having no moving parts to get out of order and being of material which will stand the ordinary corrosive effects, should make a reliable hot-water meter.

2 The object of these tests was to determine the accuracy of a Venturi meter to be used for measuring boiler feed under a great variety of conditions. The plan was to make a complete series of tests upon a small Venturi meter under all the probable conditions that would ever be met in boiler room practice. The tests were made under varying temperatures and velocities; under varying pressures, intermittent and steady; using a triplex power pump in good condition, and with one plunger out of commission, a duplex steam pump in good condition and in poor condition, and an injector.

3 The meter was installed in the steam engineering laboratory of the Worcester Polytechnic Institute and set up the way most convenient not only for weighing the water passing through the meter, but also for heating the water before it went through the meter, and pumping it in in various ways. The meter used was built by the Builders' Iron Foundry of Providence, R. I., and is what is ordinarily called a 2-in. meter, the upstream and downstream ends being 2 in. in diameter and the throat $\frac{3}{4}$ in. in diameter. The main part of the meter is of cast iron and the internal portions are lined with brass. Surrounding the upstream end and throat are annular chambers between

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the brass sleeve and the iron casing. Six holes are drilled through the brass lining into these annular chambers at about equal distances around the circumference, in order to give the actual pressure heads in the meter at both throat and upstream end. From the outside of these annular chambers were pipe connections to a manometer tube which consisted of a glass U-tube containing mercury. There were the necessary valves and pet-cocks to manipulate the meter, blowing out the air whenever it accumulated. The general layout of the apparatus is shown in Fig. 1.

4 The apparatus was set up so that the meter could be supplied from a 1¼-in. metropolitan injector, a 4½-in. by 2¾-in. by 4-in. duplex pump or 4-in. by 5¾-in. triplex power pump; or from a pressure tank supplied from the city mains or by a large duplex pump. These pumps were arranged to take their suction from a pit 12 ft. long, 6 ft. wide and 4 ft. deep, directly beneath the Venturi meter. A 1-in. steam line was put in to heat the water. There being about 300 cu. ft. in the supply pit, a very even temperature could be maintained. In order to keep the discharge from the pumps constant, a suction well was supplied, kept at constant level by an additional pump from the main pit. The discharge from all the pumps used was carried up a vertical 2-in. pipe, at the top of which was an air chamber 4 in. in diameter and 3 ft. long, with a valve so inserted that it might be cut out whenever desired. From this vertical pipe ran a line containing the Venturi meter, a thermometer-well for determining temperatures, and a valve for throttling water in order to get any desired pressure. At the end of this line was a swinging end that discharged into either of two 5000-lb. weighing tanks.

5 The first tests were made with cold water in order to determine the coefficient. These were made with steady pressure, securing a very constant flow. Water was run through the meter until the conditions had become constant. One tank was weighed while the water was being discharged into the other. The tests were started by diverting the discharge into the weighing tank and taking the time. Readings of the Venturi meter were taken every thirty seconds for low velocities and every minute for higher. The tests were ended by diverting the discharge into the other tank, taking the time and weighing.

6 When hot water tests were made, it was found that a certain amount of water evaporated; evaporation tests were therefore made, which proved this amount to be a negligible quantity.

7 In order to compare the workings of the meter under the various

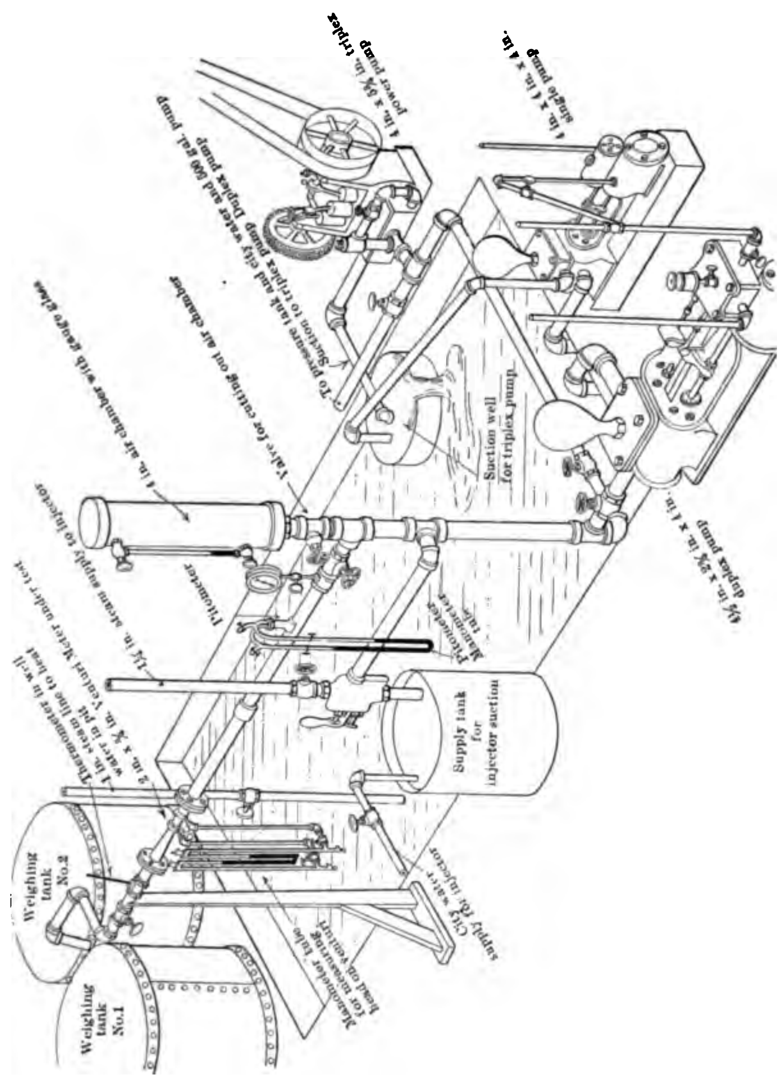


FIG. 1 LAYOUT OF APPARATUS, VENTURI METER TESTS

conditions, it was decided to determine the coefficients for this meter under these conditions. The discharge of the Venturi meter was figured from the regular formula, using a coefficient of one, and the actual weight obtained from the weighing tanks was divided by this value to obtain the real coefficient.

8 The following temperatures were used during the tests: 80 deg., 120 deg., 140 deg., and 180 deg. fahr. Water for these tests was supplied by the triplex power pump with different velocities through the meter. Tests were made at 140 deg. with and without the air chamber, the water being furnished by the triplex power pump; then at 140 deg. with one plunger of the triplex pump disconnected so as to produce fluctuations in the velocity and pressure of the water supplied to the meter.

9 In order to duplicate more nearly the conditions of boiler feed, an air chamber and check valve were placed in the pipe line in the downstream side of the Venturi meter (not shown in Fig. 1). Because of the air chamber at this end the pump fluctuations could pass through the meter to a much more marked degree than if the discharge was merely throttled by a valve. Under these conditions tests were run with water supplied by the injector, taking suction from a special supply tank; with the injector, however, the temperature of the water varied of necessity with the velocity through the meter.

CONCLUSIONS

10 The chief difficulty encountered in making this series of tests was in getting the true average readings of the manometer. With the higher velocities through the meter, the fluctuations could be easily dampened by closing the valves to the manometer tube, but with the lower velocities any error in reading was so large in proportion to the entire head as to make a considerable difference in results. It may be said, then, that the meter is not accurate for velocities of less than 10 ft. per sec. in the throat of the meter, which corresponds to a discharge of 0.03 cu. ft. per sec., or about 6140 lb. per hr., so that this meter would be best adapted for measuring water for a boiler plant of above 200 h.p. The coefficients are materially lower below 0.03 cu. ft. per sec. The principal feature shown by the cooler water tests (80 deg. fahr.) is the low value of the coefficients of the meter, the average, excepting the values for velocities below 10 ft. per sec., being 0.978. This coefficient might have been expected, however, as in a small meter the ratio of area of cross section to surface is much lower.

11 These experiments clearly show that the meter is as accurate for hot water as for cold. The maximum error in discharge, as figured from manometer deflections using the mean coefficients for that temperature, is as follows: 80 deg., 1.39 per cent; 120 deg., 1.5 per cent; 140 deg., 1.9 per cent; 180 deg., 0.82 per cent. The average error is well within 1 per cent.

12 Of the pumps tried with the meter, the triplex gave by far the best results, and it may be confidently stated that the Venturi feed-water meter would give very satisfactory results in a plant using the power pump. Even with one plunger disconnected, the maximum variation was only 2.4 per cent.

13 In tests with the injector, the weighed calculation from the Venturi formula, using the mean coefficient, shows variations from actual weight of 3 per cent. The average error is inside 2 per cent.

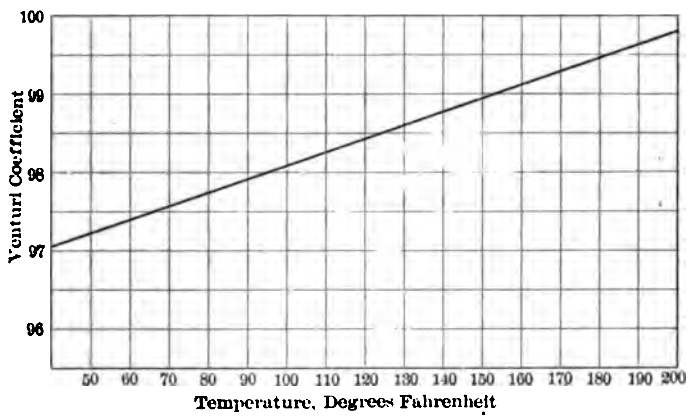


FIG 2 CURVE SHOWING VARIATION OF VENTURI COEFFICIENT WITH RISE IN TEMPERATURE

14 The results from 4½-in. by 2¾-in. by 4-in. duplex pump show up better than might be expected as this pump was in very poor condition, the piston rods being so worn that the pump took air through the stuffing boxes at the water end.

15 These tests represent the worst conditions which would be met with in boiler feed. The pump would start building up pressure, then pause until the pressure had fallen. The check valve, which was located only about 6 in. downstream of the Venturi meter, was opening and closing constantly. For a discharge of more than 0.03 to 0.04 cu. ft. per sec., the coefficients were very consistent.

16 It must be remembered in considering these tests that the Ven-

turi meter itself probably worked more accurately than the tests would indicate, as every change in velocity through the meter is accompanied by its corresponding change in head on throat and upstream end, and that a continuous recording device attached to the meter would probably cut down the error considerably. It is the opinion of the experimenters that the Venturi meter is a very reliable form of hot-water meter, provided the proper size is used.

17 These tests were conducted for the most part by George Y. Lancaster, a post-graduate student in mechanical engineering at Worcester Polytechnic Institute, and these results are taken from a thesis submitted by him.

DISCUSSION

F. N. Connet. I notice also that the results obtained were not very satisfactory when pulsations were present and when the throat velocity was less than 10 ft. per sec. The three reasons for this seldom if ever exist in actual Venturi meter installations:

a The instrument used in the test was a mercury U-tube or manometer, containing but little more than a pound of mercury. The inertia of the mercury was therefore small and the mercury levels were unsteady. In an actual installation a registering instrument is generally used which contains almost 100 lb. of mercury, the mere inertia of which has a decided "damping" effect.

b The graduations on a manometer scale are quite close together at low throat velocities. At 10 ft. per sec. throat velocity, the difference of mercury levels is only 1½ in. In the registering instruments the movements are increased by a lever so that accurate readings are facilitated.

c During the tests described in the paper, the globe valves in the two pressure pipes were partially closed to minimize the mercury level fluctuations, and in all probability the valve discs were slightly loose on the valve stems. This therefore allowed the discs to behave like check valves and permitted a freer flow in one direction than in the other; consequently incorrect mercury levels would result.

2 The only reason for not always obtaining accurate results with the Venturi meter for boiler feed is the presence of severe pulsations in velocity due to the action of the feed pump. The most accurate results can be obtained when the feed pumps are of the centrifugal

type and many such pumps of the two-stage or three-stage turbine variety are now in successful use. The pulsations which are due to the action of the water plungers or to defective valve action in a reciprocating pump, make it necessary to place a rather large air chamber directly on the pump, or on the feed line as close as possible to the pump. If placed on the feed line, it should not be connected on a tee set in the line but it should be so arranged that all of the water will pass through it. Furthermore, the cross section of such an air chamber should be large and the arrangement should be such that the surface of the water will rise and fall with each stroke of the pump. There should be a gage glass on the side of the air chamber so as to insure the presence of a sufficient vacuum of air. These precautions will render the velocity of the water sufficiently uniform to obtain accurate results with the Venturi meter.

3 The correction necessary for difference in temperatures with the Venturi meter is not as great as with mechanical meters, for the reason that the Venturi meter itself automatically compensates for one-half of the difference in specific gravity. In other words, if the water be hot and the specific gravity 2 per cent less than that for which the meter was calibrated, a correction of 1 per cent is automatically made by the meter and therefore a further correction of only 1 per cent is necessary, whereas, with a mechanical meter depending upon volumes, a correction of 2 per cent would have to be made if the readings were desired in pounds. The reason for this difference between the two types of meters is that the flow through the Venturi meter is proportional to the square root of the Venturi head and is not directly proportional to it.

4 There are at least three better ways to throttle one or both of the pressure pipes than by using globe valves. The first and perhaps the best way is to use a capillary tube, say $\frac{1}{8}$-in. inside diameter by two or three feet long. The second way is to use a needle valve which is similar to a globe valve, but without a loose valve disc and with a long tapered point directly on the valve stem. The third way is to use a cock instead of a valve. Any of these methods of throttling combined with an ample air chamber permits accurate Venturi meter readings at throat velocities as low as 2.8 ft. per sec. This extends the range of the meter from its maximum capacity down to one-thirteenth of the maximum.

5 Although a manometer, because of its portability and simplicity, is particularly well adapted to the making of short boiler tests, it nevertheless is not automatic and it shows the rate of flow only at

the moment of observation, and if this rate fluctuates considerably from minute to minute, it becomes necessary to take very frequent readings. For this reason an instrument has been perfected which has two dials, one for indicating the rate of flow and the other for continuously recording this rate upon a circular chart paper. A special planimeter enables the charts to be measured so as to obtain the total quantity of water. This planimeter multiplies the factor of velocity by the factor of time and the product, of course, represents quantity. This type of recording instrument is largely used for meters 4 in. and smaller in diameter but for larger size meters the users generally prefer a three-dial instrument of the integrating type in order that the total quantity of water may be read directly upon a revolution counter without the aid of the planimeter.

Clemens Herschel. Professor Allen's paper shows, by tests properly and skilfully made, that the meter is reliable for hot-water and boiler-feed service, and is new and unique as reproducing in tests of the meter the curious conditions to which a boiler-feed water meter is subjected. But for this feature the tests would have been only a repetition of other tests already made. Not that such repetitions are not desirable, especially when made as accurately and with the scope and purpose of those given in Professor Allen's paper. Further series of tests on Venturi meters of all sizes, are in fact still called for in the interests of exactitude. But they can only in a general way confirm, not discover.

2 The point to be considered is, that several thousand Venturi water meters are now in use, the world over. They are the embodiment of the action of one of the laws of nature, and are but little dependent on a correction by coefficients. They have been tested in various sizes, from ½-in. to 10-ft. main pipe diameter, and operate exactly alike in all these sizes. They are also used to meter gases, brine and chemicals, and, as we see from the paper, to meter hot water. It is indeed a curious circumstance, that while the inventor and the manufacturers of the Venturi water meter never expected to see many of these meters of less than 12-in. diameter used in practice yet the demand for hot-water boiler-feed meters has exceeded in value that of all the other sizes, for certain periods.

Sanford A. Moss. I understand from Par. 7 that the discharge of the Venturi meter was figured on the basis of cold water with standard density in all cases, and that the theoretical effect of change of density was not taken into account in the formula. This would mean that Professor Allen's curve takes account of the effect of density changes, as well as all other changes. The actual formula used, and a sample of the calculations, might be a desirable addition to the paper.

2 Assuming that the above interpretation is correct, Professor Allen's curve shows that the actual flow in pounds per hour, with a given pressure, increases as the density decreases, due to rise of temperature. Is this not surprising? Theoretically, flow should decrease with the square root of the density. Of course change in the orifice friction coefficient, due to change of density, temperature, etc., might occur to such a great extent as to overbalance effect of density change. The actual orifice friction coefficient would then have a greater upward slope than in the chart so as to be over 98 per cent at 200 deg. Orifice friction coefficients for all density conditions and all fluids are usually the same for velocities occurring in practice, which are always above the "critical velocity" where fluid adjacent to a wall is stationary and where viscosity is a factor. Thus the orifice coefficient for air is the same as for water, even though the density is decreased about 800 times.

F. N. Connet. If I understand Dr. Moss correctly, he states that the quantity decreases as the density increases. With the Venturi meter this depends upon the character of the graduations. If the units are cubic feet the readings *decrease* in proportion to the square root of the increase of density, but if the units are pounds the readings *increase* in proportion to the square root of the increase of density. One is exactly the reverse of the other.

Geo. A. Orrok. I note that Professor Allen has obtained results for the coefficient of the Venturi meter similar to those given by Clemens Herschel in his paper presented before the American Society of Civil Engineers, December 21, 1887, the lower values of the coefficient appearing at a velocity of about ten feet per second.

2 The New York Edison Company for some years has been using Venturi meters for the measurement of water. We find them accurate and very convenient. For the last three years we have been using them in the testing of our boilers, having conducted a series of check experiments to determine the variations with temperature. Our condition is considerably better than Professor Allen's, since we use centrifugal feed pumps and consequently have a steady reading on the manometer.

3 In cases where we have both weighed and measured the feed water our results were remarkably close. On a 7-hr. test, where about 170,000 lb. of water was fed to the boiler, the meter exceeded the weighing by 631 lb., or approximately 0.37 of one per cent. In another test, in which nearly 200,000 lb. was fed, the difference was about 0.47 of one per cent. I believe the meter readings are more nearly correct than the weighing, as there was considerable opportunity for evaporation from the tanks in which the weighing was done.

The Author. I agree with Mr. Connet in regard to the throttling of the water in the pipes leading to the manometer. I believe the needle valve, or a fairly long pipe of small diameter, would be a decided improvement over the globe valves which were used in these experiments. We had not discovered that the movement of the end of the globe valves affected the reading, but Mr. Connet has had a good deal more experience along these particular lines, and I am perfectly willing to believe that this is true and that these fluctuations could be materially cut down and yet give the true mechanical average. This is what we are looking for, and it is a good deal better than using maxil mum and minimum readings and then obtaining the arithmetical average. The mechanical average obtained by means of throttling is certainly more accurate because we do not know how long the maximum deflection continues, relative to the minimum.

2 For the benefit of Mr. Moss, I would state that the density at different temperatures was considered. The following is a sample test giving an idea as to how computations were made:

If W = actual weight of water from weighing tank, then

$W = 60 w a C t \sqrt{2gh}$
w = weight per cu. ft. at the temperature
a = area Venturi throat
C = Venturi coefficient
t = time in minutes
h = Venturi head

$$C = \frac{W}{60\ w\ a\ t \sqrt{2\ gh}}$$

$$C = \frac{W}{1.48\ u\ t \sqrt{h}}$$

DATA OF TEST

Time 3:40 −3:51; duration 11 minutes.	lb.
Weight of tanks at beginning........................	1158
Weight of tanks at end............................	5369
	4211
Deduct for tank calibration.	20
	4191
Add for evaporation..............................	2
Total water................................	4193
Mean mercury deflection.......................	17.24 in.

$h = 1.05 \times 17.24 = 18.1$ ft.

$\sqrt{h} = 4.25$

w for temperature of $137 = 61.43$

Weight $= 1.48 \times 11 \times 61.43 \times 4.25 = 4250$

$C = \dfrac{4193}{4250} = 0.986$ coefficient of Venturi meter

No. 1255

THE PITOT TUBE AS A STEAM METER

By Prof. Geo. F. Gebhardt, Chicago, Ill.

Member of the Society

Steam meters may be conveniently grouped in two general classes, which, for lack of more suitable names, may be designated as *a*, series meters, and *b*, shunt meters.

2 The series meter is an integral part of the piping, the entire mass of fluid to be measured passing through the apparatus. The St. John's and Venturi meters are the best known of this class. In the former the *volume* of fluid passing is determined by the rise and fall of a weighted plug valve and in the latter the *velocity* of flow is determined by the well-known principles of the Venturi tube. Both are indicating instruments and show only the rate of flow.

3 In the shunt meter only a portion of the steam to be measured is diverted through the apparatus, the velocity of flow through the shunt being an indication of that in the main pipe. In this class one or more small openings ½ in. or less in diameter suffice for attaching the apparatus to the pipe. One instrument suitably calibrated may answer for any size of pipe. The Pitot tube forms the basic principle of practically all meters of this class.

4 It is the object of this paper to describe a number of applications of the Pitot tube for steam measurements as constructed and tested at the Armour Institute of Technology.

5 The Pitot tube was first used by its inventor, Pitot, in 1730, in the measurement of the flow of water and since then has been successfully used in measuring the flow of many fluids and all true gases. Considerable difficulty has been experienced, however, in its application to vapors condensable under normal conditions of operation; and so far as the writer knows, no commercially successful instrument is on the market.

6 Many of the instruments are interesting laboratory devices and are of considerable value for experimental investigations; but on

Presented at the Annual Meeting, New York, (December 1909), of The American Society of Mechanical Engineers.

account of the great number of variables involved, fall short of being practical commercial instruments. Fig. 1 illustrates the most common and the least accurate application of the Pitot tube for measuring the flow of condensable vapor in a pipe. S is the static nozzle at right angles to, and D the dynamic nozzle facing the current. U is an ordinary manometer partially filled with mercury. When there is no flow the mercury in columns N and W will be on the same level and the upper portions will be filled with condensed vapor. When there is a flow the mercury will be depressed as indicated and the difference in height H of the mercury columns in the two tubes will be

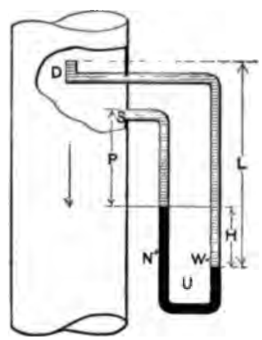


FIG. 1 PITOT TUBE WITH MERCURY MANOMETER

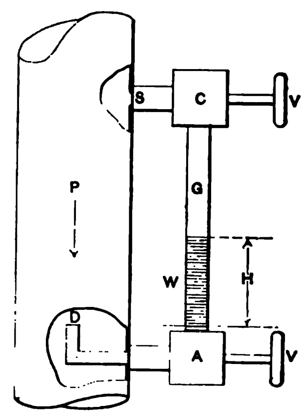


FIG. 2 SIMPLE GAGE-GLASS METER WITH SELF-ADJUSTING WATER COLUMN

a measure of the velocity of flow in the main pipe. On account of the great density of mercury and the variation in height of the condensed vapor above the mercury this application of the Pitot tube has very little value scientifically or commercially. For example: With dry steam at 100-lb. gage, a velocity of 8000 ft. per min. would give a depression H of only 1 in. and an error of 1/100 in. in measuring H would mean an error of 40 ft. per min. in the velocity.

7 In Fig. 2 is shown the original apparatus designed by Prof. R. Burnham of the Experimental Department of Armour Institute of Technology and the writer, in which the water of condensation is used as a self-adjusting column in place of mercury. This embodies the basic principle of many of the meters constructed later.

8 Referring to Fig. 2, A and C are two ordinary water gage cocks

and G an ordinary glass tube. Gage C is connected to the static nozzle S and gage A to the dynamic nozzle of a Pitot tube. The height of water H varies as the square of the velocity of steam flowing through pipe P and automatically adjusts itself to the variations in velocity; thus, for decreasing velocities the water in glass G discharges itself through tube D until the water column H balances the velocity pressure in pipe P, and for increasing velocities condensation from the upper part of the instrument accumulates and the water column H rises until a balance is effected for the higher velocity.

9 The velocity of flow is determined by the well known equation:

$$V = c\sqrt{2gH} \tag{1}$$

in which

- V = maximum velocity of flow, ft. per sec. Dynamic nozzle D is inserted in middle of pipe where the velocity is a maximum.
- c = coefficient determined by experiment.
- g = acceleration of gravity = 32.2.
- H = height of a column of steam equal in weight to water column H.

Equation (1) may be expressed:

$$V_1 = 139\,c\sqrt{h\frac{d_w}{d_s}} \tag{2}$$

in which

- V_1 = maximum velocity, ft. per min.
- d_w = weight of 1 cu. ft. of water in gage glass G.
- d_s = weight of 1 cu. ft. of the steam or mixture in pipe P.
- h = height of column H in inches.

The weight of steam flowing per hour may be determined by substituting the proper quantities in equation (2), thus:

$$W = 58\,acr\sqrt{h\,d_s d_w} \tag{3}$$

in which

- W = weight of steam flowing, lb. per hr.
- a = area of the pipe, sq. in.; other notations as in (2).
- r = ratio of the mean velocity to the maximum.

10 Equations (2) and (3) are general and are applicable to any size pipe and any pressure and quality of steam. For a given size of pipe, say 3 in. (extra heavy), and a given pressure of steam, say 70 lb. gage, equations (2) and (3) assume the following simple forms:

$$V = 2435\,cr\sqrt{h} \tag{4}$$

$$[illegible] = 1292\,cr\sqrt{h} \tag{5}$$

11 Tests with pipes 1 to 6 in. in diameter gave r a value of 0.79 to 0.84. For a 3-in. extra heavy pipe this value was 0.82 and remained practically constant for all velocities and pressures (atmospheric to 100 lb. gage.) Substitute this value of r in (4) and (5),

$$v = 1996\ c_1 \sqrt{h} \tag{6}$$

$$w = 1060\ c_1 \sqrt{h} \tag{7}$$

in which

v = actual mean velocity, ft. per min.
w = actual weight of steam flowing, lb. per hr.
c_1 = a coefficient determined by experiment.

Coefficient c_1 varied from 0.8 to 1.2 for the simple instrument in Fig. 1.

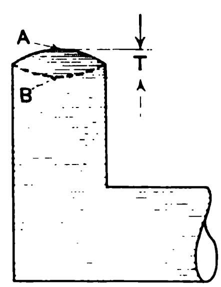


Fig. 3 Influence of Surface Tension on Height of Water in Dynamic Nozzle

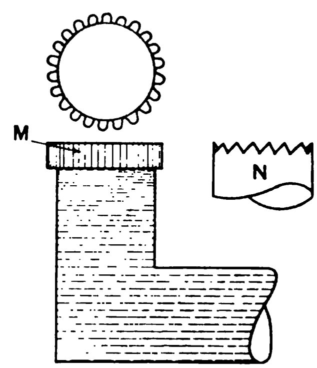


Fig. 4 Dynamic Nozzle with Arrangement for Maintaining a Constant Level of Overflow

12 Variation in the value of c_1 was found to be due to

a Fluctuation in height T (Fig. 3), at the end of the dynamic tube due to surface tension. With a plain tube this variation amounted to as much as 0.25 in.
b Variation in density of water column.
c Capillarity in gage glass G.
d Aspiration in static tube at high velocities.

13 (*a*) A number of devices were constructed for eliminating the variation in height of T in the dynamic nozzle but all proved inefficient, except that shown in Fig. 4. By serrating the tube as indicated in N, Fig. 4, and surrounding it with a corrugated ferrule M, thereby forming a series of capillary tubes, the variation was practically eliminated, amounting to but 0.02 in.

14 (*b*) The variation in density of the water column is an inherent defect of this type of meter and cannot be remedied in this simple form of apparatus. The experiments gave a range in temperature

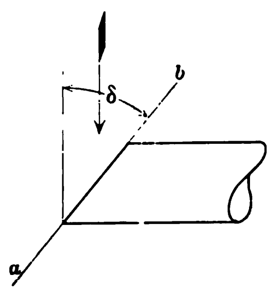


Fig. 5 Static Nozzle Corrected for Aspiration

of 150 to 300 deg. fahr. resulting in a maximum possible error of 6 per cent. For high velocities the fluctuation in temperature is negligible but for low velocities the range may be considerable.

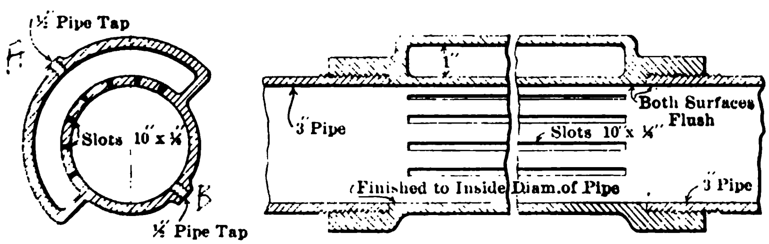


Fig. 6 Device for Determining Aspiration Effect

15 (*c*) Capillarity in the gage glass increases as the diameter of the glass decreases and may be considerable in tubes of small bore. With a $\frac{3}{4}$-in. tube it amounts to 0.05 in., hence its influence is negligible for high velocities.

16 (*d*) Aspiration in the static tube is appreciable only with

velocities above 6000 ft. per min. It may be entirely eliminated by beveling the tube as indicated in Fig. 5. In this device the dynamic and aspiration effects neutralize each other and only the true pressure is recorded. The aspiration effect was determined by means of the apparatus illustrated in Fig. 6. It consists of a special fitting containing a chamber in communication with the main pipe, but so constructed that the velocity of flow in the chamber is practically eliminated. The difference in pressure between the two openings A and B is due to aspiration. Further experiments are necessary to show whether any fixed angle is applicable to all velocities.

17 A simple construction, as illustrated in Fig. 2, with ½-in. pipe

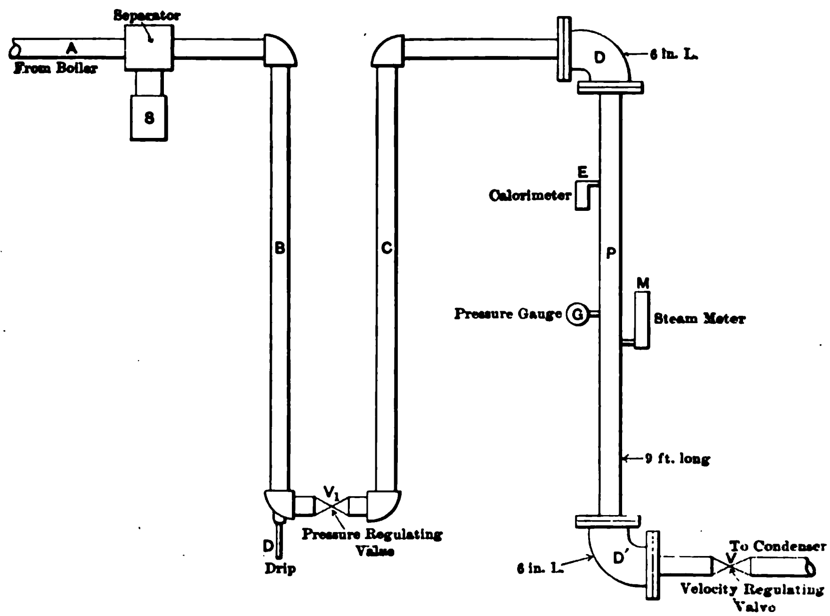


FIG. 7 DIAGRAMMATIC ARRANGEMENT OF PIPING FOR TESTING STEAM METERS

connections and ¾-in. gage glass, fitted with dynamic tube, as illustrated in Fig. 4, and static tube as in Fig. 5, is an accurate means of indicating the true velocity of flow and the actual weight of water discharged for all velocities above that corresponding to 1½ in. of water. For lower velocities any error in reading is so large in proportion to the entire head as to make considerable difference in results. The scale may be graduated to read velocities in feet per minute and the water rate in pounds per hour.

18 The results of a few tests of this simple device are given in Table 1, and the arrangements of piping for conducting the tests is shown diagrammatically in Fig. 7. The results are also plotted in Fig. 8. The curves give the weights as calculated from equation (7) and the small circles the weights as determined from the condensed

TABLE 1 TEST OF SIMPLE GAGE GLASS METER

STEAM PRESSURE 70-LB. GAGE. STEAM DRY. 3-IN. EXTRA HEAVY PIPE.

h	Velocities Feet per Minute			Weights Pounds per Hour			
	Actual Mean	Maximum by Meter	Ratio	Actual	Meter	Difference	Error
[illegible]	2860	3528	1.23	1516	1545	29	1.88
[illegible]	3890	4685	1.20	2062	2052	10	0.48
[illegible]	4530	5480	1.21	2400	2400	0	0.00
[illegible]	5360	6570	1.22	2852	2878	26	0.91
[illegible]	6170	7405	1.20	3270	3240	30	0.92
		Average	1.21				

[illegible] = 0.82. Coefficient of Meter = 0.82 × 1.21 = 0.992 or practically unity.
Equation for meter: $V = 1996\sqrt{h}$, and $W = 1060\sqrt{h}$.

steam. It will be noted that coefficient c_1 is unity and no calibration is necessary. Tests on 1-in., 2-in., 3-in., and 4-in. standaro pipe gave a coefficient practically of unity. The tests tend to show that for a given size of pipe and a constant pressure and quality of steam the actual mean velocity and weight of water may be accurately determined by equations:

$$v = 139\sqrt{h\frac{d_w}{d_s}} \qquad (8)$$

$$w = 58a\sqrt{hd_w d_s} \qquad (9)$$

This is strictly true for continuous flow only. Interrupted flow, as in connection with reciprocating engines, creates a fluctuating water column and it is difficult to obtain the mean readings. For engines making over 100 strokes per minute the height of water is practically constant but for lower speeds the fluctuation increases with the decrease in speed. By suitably throttling the lower valve V, water column H_s may be made to assume a fairly approximate mean value for speeds as low as 20 strokes per minute in engines taking steam full stroke.

19 The limitations of the simple gage glass meter for commercial purposes are:

a It is purely an indicating device and readings must be taken frequently to obtain average results.

b A scale graduated for a given set of conditions is accurate

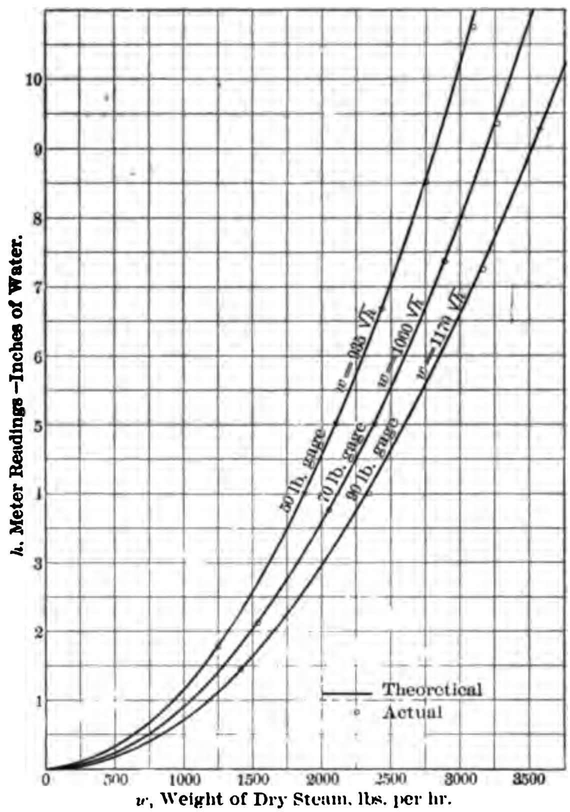


FIG. 8 TEST OF SIMPLE GAGE-GLASS METER, 3-IN. EXTRA HEAVY PIPE

only for these conditions, the degree of accuracy varying with the fluctuation in steam pressure, change in quality of the steam and variations in temperature of the water column. A convenient form of chart for a given size of pipe and for a wide pressure range is illustrated in Fig. 9. The chart is wrapped around the drum, as in Fig. 10, and set to correspond to the given pressure. The height of

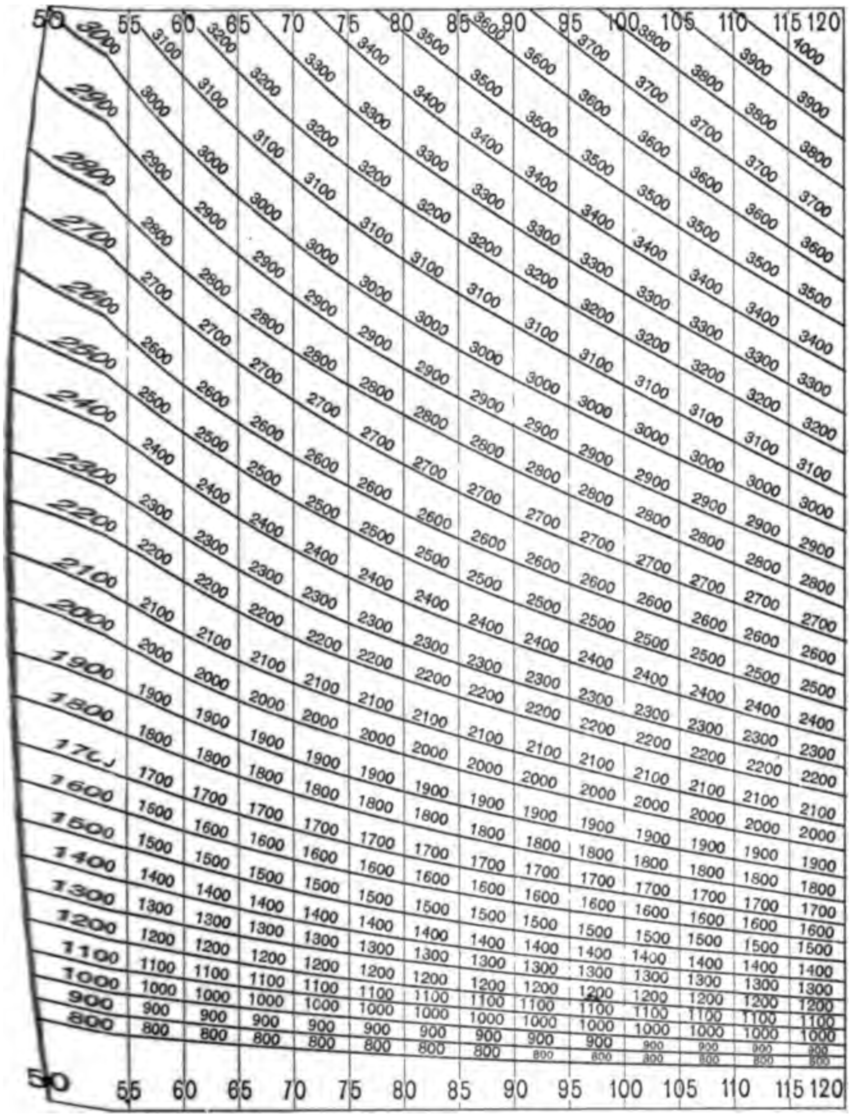


FIG. 9 TYPE OF CHART GIVING WIDE PRESSURE RANGE FOR GIVEN SIZE OF PIPE

the water column transferred to the chart by a su pointer gives at once the weight of steam flowing.

c Rapid increase in flow may cause the water in the gage to be blown out requiring several minutes for suff condensation to collect and balance the velocity he

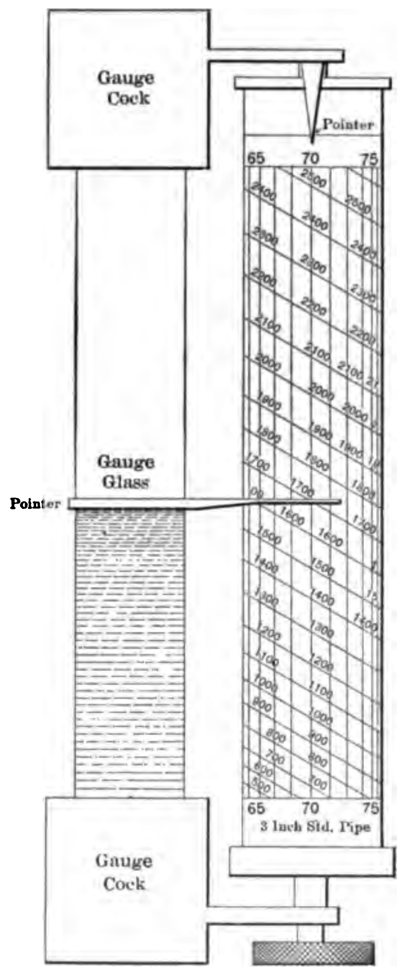


FIG. 10 METHOD OF MOUNTING CHART

d Inaccuracy for velocities below that corresponding 1½-in. water column, or roughly, 2000 ft. per min. for sures over 70-lb. gage.

e Cannot be used for measuring the flow of highly superheated steam except for practically constant flow and constant degree of superheat. With highly superheated steam a condensation chamber must be fitted to upper gage cock *C*, Fig. 2.

20 Fig. 11 shows a modification of the simple gage glass meter with simple pipe connection, which requires the pipe line to be tapped in only one place. It may be set at any angle with the horizontal, thereby increasing the sensitiveness of the readings.

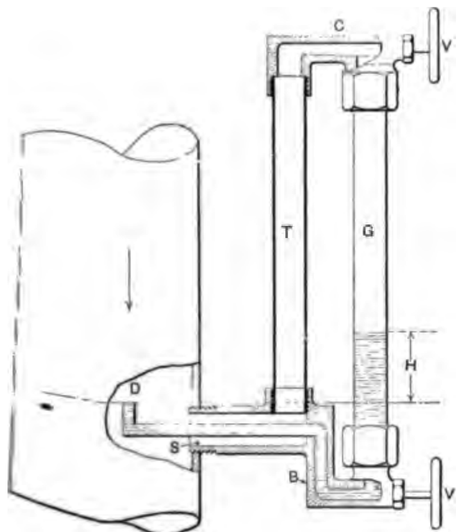


FIG. 11 GAGE-GLASS METER WITH SINGLE PIPE CONNECTION

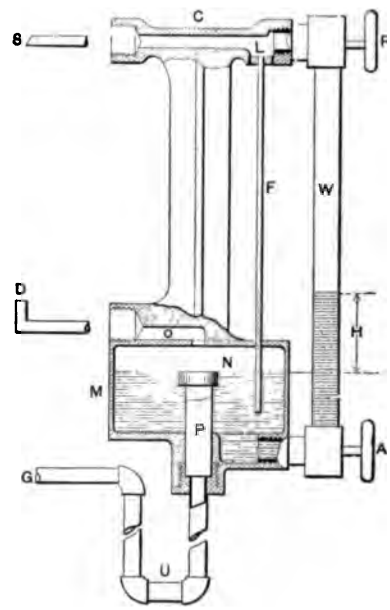


FIG. 12 GAGE-GLASS METER WITH DR DYNAMIC TUBE

21 Fig. 12 shows an application of the gage glass meter in which many of the defects of the simple gage glass device are eliminated. The error due to the variation in water level at the end of the dynamic nozzle is entirely eliminated by making the tube a "dry" tube; i. e., water of condensation is not returned through the dynamic nozzle. The temperature of the water in the gage glass is practically constant, all condensation from the static end being discharged through drain *F* to the chamber below, and all water carried over by the dynamic end being discharged into pipe *P* directly to overflow *G*. The area

and volume of chamber M is so large compared with that of glass W that sudden variations in flow do not materially affect the level of water in M and cannot blow the water out of glass W. The only defect in the instrument is the error due to capillarity in glass W, which, as stated before, amounts to but 0.05 in. for a $\frac{3}{4}$-in. tube.

22 The operation is as follows: Velocity pressure is transmitted through tube D and opening O, into the body of chamber M. This pressure acting on the surface of the condensation in the chamber forces water into glass W until a balance is effected. Condensation is discharged continuously through pipe P and water seal U of the

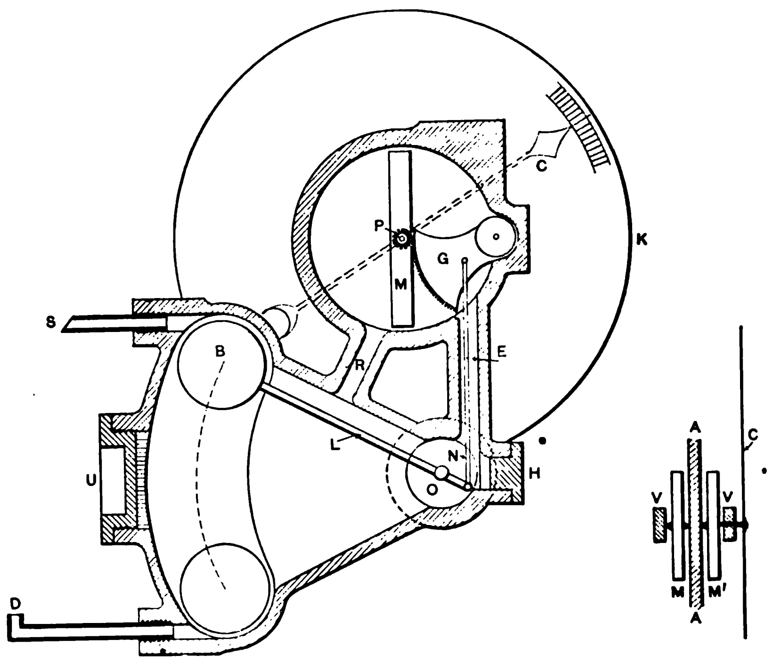


FIG. 13 METER WITH SELF-ADJUSTING WATER COLUMN FOR LOW VELOCITIES

main pipe. Tests of this meter gave practically theoretical results for all velocities ranging from the equivalent of a $\frac{1}{2}$-in. water column to 10-in.

23 Fig. 13 shows an application of the self-adjusting water column for very low velocities, 2000 ft. per min. and under. Glass ball rises and falls with the water column, transmitting its motion through levers L and N, gear sector G and pinion P, to permanent magnet, M. The latter transmits its motion to magnet M' outside the casing.

Pointer C is fastened to magnet M'. Thus any motion of ball B is multiplied by pointer C and indicated on dial K. Magnets M and M' are independently mounted on pivot bearings, M within the casing and subjected to steam temperature, the other outside the casing. Motion of M is transmitted magnetically to M' through the casing, thus doing away with stuffing boxes. The relative positions of magnets and casing are illustrated in the lower corner of Fig. 13. On account of the angularity of the connecting links and the frictional resistances, small as they are, the dial graduations cannot be conveniently calculated but must be calibrated by experiment.

24 This instrument is very sensitive, indicating velocity changes of 200 ft. per min. When connected directly to the pipe it is subject

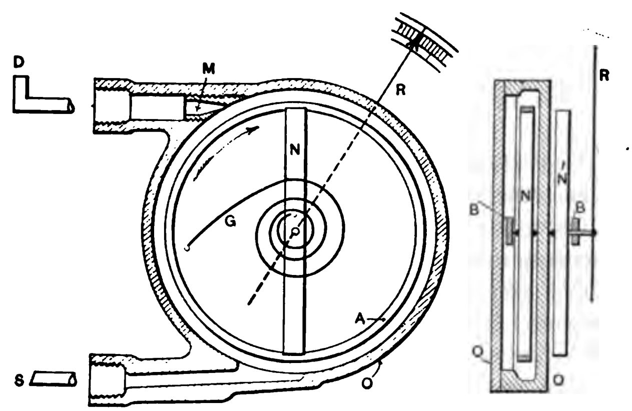


FIG. 14 INDICATING IMPULSE METER

to all of the errors of the simple gage glass meter and is altogether too sensitive for accuracy.

25 All of the devices described above are simple indicating-mechanisms, and with the exception of the one illustrated in Fig. 13, cannot conveniently be made autographic.

26 For commercial purposes a steam meter should be autographic, or better still, integrating. The ideal meter is one which shows at a glance the weight of steam flowing for any given period of time and which may be read as one does a watt-hour meter.

27 Fig. 14 shows an application of the Pitot tube for indicating, autographically, the weight of steam discharged and differing basically from those just described. A permanent magnet N forms the spoke of a small aluminum wheel A. Rotation of wheel A is resisted

by spiral spring G. D and S are dynamic and static nozzles, respectively, of a Pitot tube. The velocity head discharges a small jet of steam through nozzle M and exerts a force on the periphery of wheel A, tending to rotate it about its axis. The angular rotation increases with the velocity. The motion of wheel A is imparted through the medium of magnets N and N' to pointer R. By means of a suitable clock-work the angular movement of wheel A may be autographically transferred to a chart giving a continuous record of the weight of steam flowing.

28 By permitting the wheel to rotate and by connecting magnet N' to a series of rotary dials an integrating or total output mechanism is readily effected.

29 Experiments are now being conducted with the autographic and integrating devices just described, but sufficient data are not yet available as to their respective merits.

DISCUSSION.

Prof. W. B. Gregory. The author has apparently developed a practical instrument of real value. However, it seems to the writer that the device for determining aspiration effects can not be relied upon to make determinations of any value.

2 I would like to ask Professor Gebhardt if he has used static openings about 1/16 in. in diameter drilled at right angles to the axis of the pipe? Extensive experience with the Pitot tube as a device for measuring the velocity of water has taught me to avoid irregularities in a pipe, due to special fittings or other causes, when the static pressure is taken from the walls of the pipe. An unobstructed length of straight pipe is absolutely essential to accurate work.

3 The desirability of finding the correct static pressure is apparent as it seems probable that one constant would apply to reduce velocity at the center to mean velocity, in any and all sizes of pipe. The experimental determination of the correct angles for the static nozzle, as shown in Fig. 5, would then be avoided.

Walter Ferris. The remarks of Professor Gregory in regard to the special fitting for finding the effect of aspiration remind me forcibly of an experience a few years ago with both a Venturi meter and a Pitot tube for measuring water. Perhaps the conclusions at which I arrived at that time may be suggestive, although possibly not of direct application in the case of a steam meter.

2 Until quite recently, that is, within a few years, I think it has been assumed that it was necessary, in the use of the Pitot tube, to have a static tube close to the dynamic tube, or at least at the same distance from the walls of the conduit. I believe that William Monroe White, six or seven years ago, made some experiments demonstrating that the velocity head taken from the impact side of a pitot tube is correct, whatever the shape of the nozzle, so long as it is a surface of revolution. Thus the nozzle may be either cylindrical, or a converging or diverging cone, and the dynamic head will be correctly indicated, any variations in the coefficient of the pitot instrument as a whole being due to the shape or location of the static opening.

3 In the Venturi meter, we find that the static pressure is always taken from the walls of the conduit, where the velocity may not be over half the maximum velocity, and yet the results from the Venturi meter are invariably correct to within one per cent, if conditions are favorable. Therefore a dynamic nozzle, which is a surface of revolution, combined with a static nozzle terminating in the wall of the conduit (as in the venturi meter) should together form a Pitot instrument which is correct to the formula, and needs no calibration. This seems to indicate that for a Pitot instrument to measure the flow of water it is not necessary to take the static head and the dynamic head in regions of the same velocity, and that the true average static pressure will be indicated through intervening velocities, and correctly registered, even when the piezometer is located in a region of low velocity. From this I infer that in this steam meter sufficiently small static openings in the true smooth wall will probably give correct results as they do in the water meter, although I have no experimental data with which to confirm this opinion.

A. R. Dodge. I would like to take exception to a statement in Par. 6: "On account of the great density of mercury and the variation in height of the condensed vapor above the mercury, this application of the Pitot tube has very little value scientifically or commercially." The General Electric Company has developed a steam meter, both of the indicating and recording types and has built several hundred of these meters using mercury and condensed vapor above the mercury. This condensed vapor automatically remains at a constant head.

2 Recently three recording meters, selected at random out of a lot of fifty, showed a maximum error of less than two per cent. Ninety per cent of the readings were within one per cent on the three meters, which had an automatic pressure correction and also a temperature

correction. These meters can be used on any size of pipe, from 2 in up to 36 in., the 36-in. pipe, of course, being for atmospheric conditions of steam. These steam meters we have found to be valuable in improving the consumption of steam in our various plants.

3 We have also experimented with several of the types described in this paper in which mercury is not used and have found them excellent in many respects, but the use of mercury is not at all objectionable.

The Author. Static openings, about one-sixteenth of an inch in diameter, drilled at right angles to the axis of the pipe, showed no aspiration effects at velocities up to 15,000 ft. per min. (the maximum obtained during the tests) but are unsuitable for the appliances described. It is the author's intention to develop a simple meter which can be constructed of standard fittings and which may be attached by tapping the pipe in the ordinary way. Such an application necessitates the projection of the static nozzle beyond the inner surface of the pipe, an arrangement which causes serious aspiration. With a standard ½-in. nipple projecting ½ in. beyond the inner surface of a 3-in. pipe an aspiration effect corresponding to 10 in. of water was noted at a velocity of 12,000 ft. per min. (pressure 100-lb. gage). At a velocity of 6000 ft. per min. the aspiration amounted to 1½ in. of water. It was for the purpose of neutralizing this aspiration that the static nozzle was cut at an angle, as indicated in Fig. 5.

2 Mr. Ferris' remarks are in accordance with experiments conducted by the author, but, as stated above, a static opening terminating with the inner wall of the conduit is not applicable to the instruments in question. Fig. 1 illustrates such a static opening, but in the actual construction the nozzle projected ½ in. beyond the inner surface.

3 Mr. Dodge's experiments with the use of mercury as an indicating medium are of considerable interest, in that they show the development of a practicable and accurate steam meter which is little known to the general engineering public.

No. 1256

EFFICIENCY TESTS OF STEAM-TURBINE NOZZLES

By Prof. Frederick H. Sibley, University, Ala.
Member of the Society

T. S. Kemble,[1] Cleveland, O.
Non-Member

n 1905 a series of tests was begun at Case School of Applied Science veland, O., to determine the proper proportions and the efficien- of steam-turbine nozzles for given steam conditions. The final s, from which the results given herewith are derived, were made n apparatus designed by T. S. Kemble, of the Chase Machine Co., Cleveland. The writers spent about two years on these tests, through the generosity of the company and the facilities afforded Case School, were able to procure apparatus of considerable pre- on.

THEORY OF NOZZLES

The nozzle should be so constructed that the expansion of the m will take place between the given initial and terminal pressures wholly within the nozzle, the steam filling it completely at all tions. The available heat energy of the steam less that lost by tion will then be all converted into kinetic energy and the effi- cy will be a maximum. By efficiency we understand the kinetic rgy of the jet per unit mass, divided by the available heat energy he steam per unit mass.

To find the correct relation between the length and the ratio hrust to muzzle area of the nozzle, that the efficiency may be a imum, is the problem to be determined experimentally. Let

[1] T. S. Kemble, Experimental Engineer for the Chase Machine Company, eland, O.

resented at the Annual Meeting, New York, (December 1909,) of The rican Society of Mechanical Engineers.

W = weight of steam flowing in pounds per second.
V = velocity of the jet in feet per second.
M = mass.
F = reaction in pounds.
g = acceleration due to gravity at Case School = 32.16015
K = kinetic energy of the jet in foot-pounds or B.t.u.
E = total available heat energy of the steam.
Efficiency = $K \div E$.

4 From Mechanics we have: $K = \frac{1}{2}MV^2$ and $M = W \div g$. If $W = 1$ lb., then

$$K \text{ (the B.t.u.)} = \frac{V^2}{2g \times 778}$$

also

$$V = \frac{Fg}{W} \quad \text{for any flow}$$

F and W are the factors to be determined by experiment.

METHODS SUGGESTED

5 Three methods may be suggested for determining the efficiency of steam nozzles.

- *a* By measuring the force of the jet when allowed to impinge on an external surface.
- *b* By investigating the character of the jet with a search tube inserted axially in the nozzle.
- *c* By measuring the reaction of the nozzle when a jet of steam is flowing through it.

6 The first method involves complications which tend to cast some doubt upon the results obtained. The force upon the external surface may be modified by the character of the surface, by eddying and steam friction, and by the distance traveled by the jet after leaving the nozzle and before it reaches the surface. When the surface used is a flat plate perpendicular to the axis of the jet, the force may even vary from a maximum to a negative value according to the relative location of the plate and nozzle.

7 The second method was tried in a series of experiments to find the pressure in the jet at various sections of the nozzle. A search tube was inserted axially in the nozzle and the relation between pres-

sure and flow was compared with the theoretical relation, as calculated and plotted from the steam tables. The first search tube used was $\frac{1}{16}$ in. in outside diameter and $\frac{1}{64}$ in. in inside diameter. One end was closed and the other connected to a mercury column. A $\frac{1}{64}$-in. hole was drilled through the search tube at right angles to its axis. The

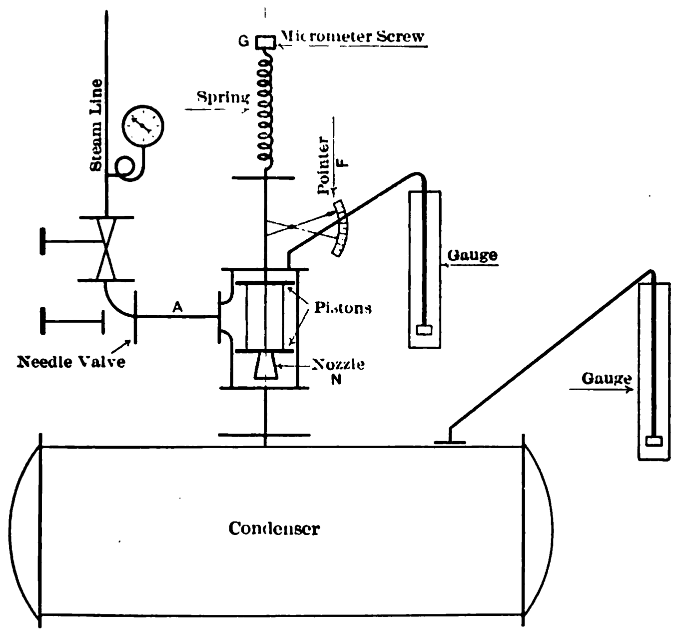


FIG. 1 DIAGRAM OF PISTON APPARATUS

readings with this tube varied greatly under identical conditions and a tube with a larger bore was tried, which gave results much more consistent but not nearly accurate enough to determine the efficiencies at the various sections of the nozzle.

8 The third method provided for the determination of the reaction of the jet in the nozzle, and apparatus was constructed for this purpose, differing in detail as follows:

a By fastening the nozzle into the outer face of one of a pair of rigidly connected pistons suspended in a cylinder. Steam entering between the pistons and flowing out through the nozzle would produce a measurable reaction.

b By using a flexible steel tube suspended freely by one end and having the nozzle attached to a chamber at the other end with its axis perpendicular to the axis of the tube. Steam flowing downward through the tube and out of the nozzle would cause the tube to deflect with a measurable force.

PISTON METHOD

9 Difficulty was experienced with the piston apparatus, diagram of which is shown in Fig. 1, due mainly to the static friction of the

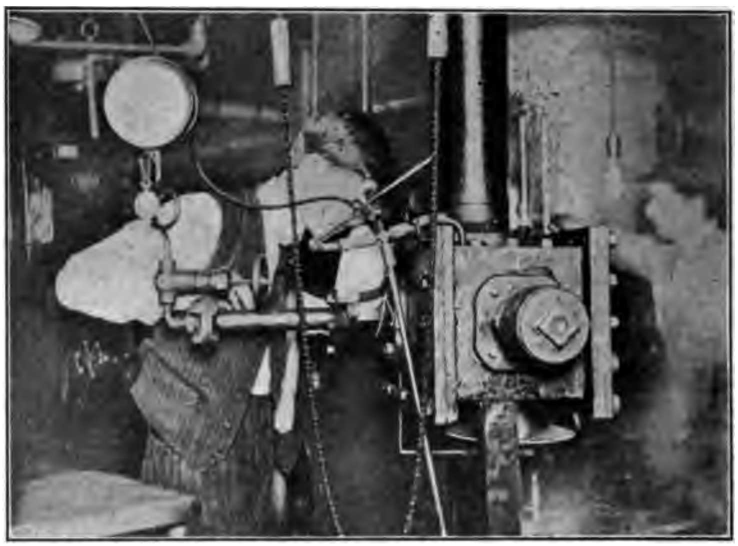


FIG. 2 FLEXIBLE TUBE APPARATUS FOR MEASURING REACTION OF JET

pistons when under pressure and to leakage past the pistons when they were reduced in diameter to lessen the friction. Modification of this apparatus to overcome these difficulties would have required almost complete rebuilding and the flexible-tube apparatus was therefore tried and with such success that it was not considered necessary to return to the old one.

FLEXIBLE TUBE APPARATUS

10 The apparatus shown in Figs. 2 to 4 was designed with a special view to combining the search-tube and reaction methods, by

which means we hoped to obtain results more accurate than would result from either method alone.

11 The tests were conducted under a pressure of 155 lb. gage. There was about 50 deg. fahr., superheat at the boiler, which was connected with the testing apparatus by about 55 ft. of 5-in. pipe and 25 ft. of 4-in. pipe.

12 In the main steam pipe was located an angle needle-valve operated by a sprocket wheel and chain which made it possible to hold the nozzle feed pressure very constant. From this point the

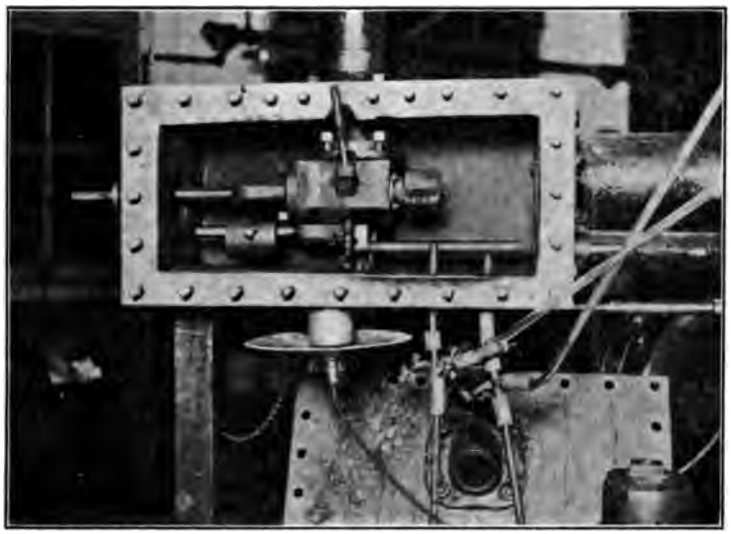


FIG. 3 INTERIOR OF BOX CONTAINING FLEXIBLE TUBE APPARATUS

steam passed downward through the tube *A* to the chamber *B*, (Fig. 4) thence through the nozzle *C* into the box *D*, and on through the passage *E* to the condenser.

13 The upper end of the tube *A* was screwed into a diaphragm on the lower flange of the angle valve. At its lower end it supported the chamber *B* which was allowed to move between stops restrained only by the stiffness of the tube and of the spring *F*. The motion of the chamber *B* was indicated by a needle which multiplied the motion about ten to one. The spring *F* was operated by a micrometer nut and screw and was calibrated *in place* by known weights hanging on a flexible wire cable which extended from the back of the chamber in the

line of the nozzle axis and down over a ball-bearing sheave. The tube A, the chamber B and the nozzle, were all enclosed in the vertical pipe P and the box D, and the vacuum surrounding them was greater than that in the condenser owing to the "augmenter" action of the steam jet entering the passage E to the condenser.

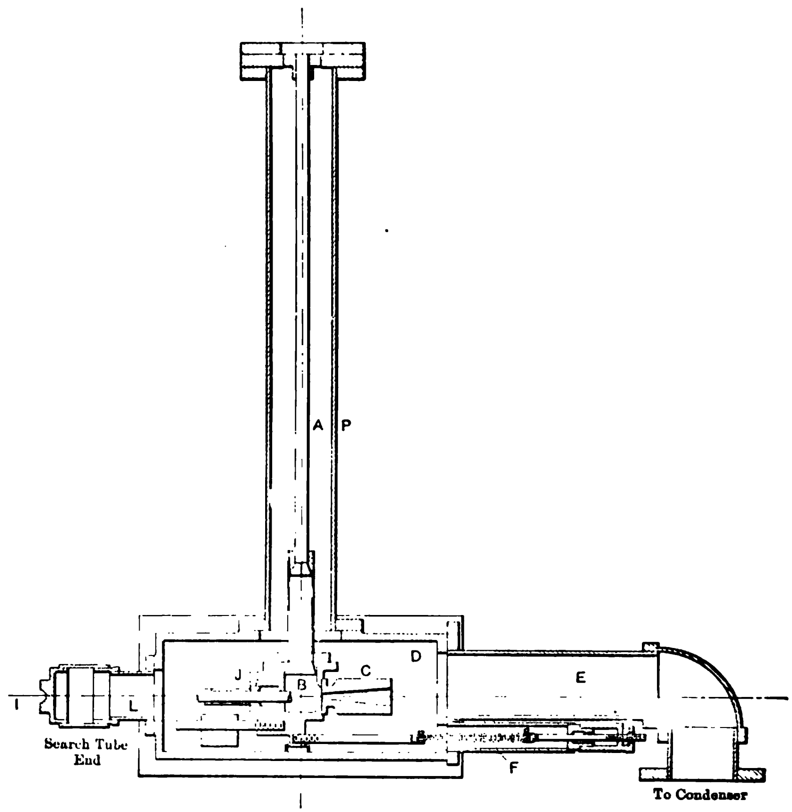


FIG. 4 SECTION OF FLEXIBLE TUBE APPARATUS

14 The initial temperature of the steam was shown by a thermometer inserted in a well in the chamber B and observed through a glass in the box D. A steam gage was connected by a flexible tube to the chamber B. The vacuum in the condenser and in the box D was shown by mercury columns, and another column was joined by a flexible connection to a hole drilled as near as possible to the muzzle of the nozzle and perpendicular to the wall. All these connections

to the mercury columns were of glass tubing with rubber couplings which allowed the moving parts to swing freely without friction and made it easy to observe any accumulation of moisture. By breaking the connection air could be let through to dry them quickly.

FORMS OF NOZZLE TESTED

15 The exact dimensions of the nozzles are shown in Fig. 6 and Fig. 7. They were all of machinery steel, bored taper, and had entrances rounded off with a hand tool to approximately ⅛-in. radius. All nozzles except No. 15 and No. 18 were bored smooth and polished. Nos. 10, 11 and 12 were identical except as to length and angle of divergence. No. 18 was like No. 11 except that it was finished rough on the inside between the throat and muzzle, the finishing chip being

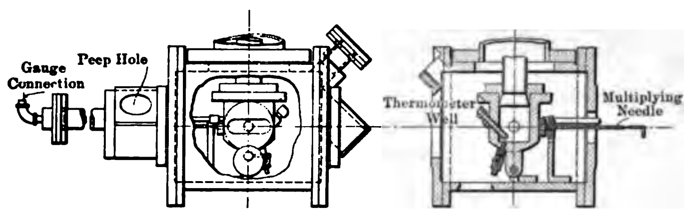


FIG. 5 END VIEWS OF SEARCH-TUBE END AND OF CONDENSER END

made with a threading tool having an angle of 120 deg., the feed being 90 turns per inch. No. 14 and No. 15 were identical except that whereas No. 14 was bored taper, No. 15 was made in halves, and after being bored taper these halves were separated and milled longitudinally with a 90-deg. cutter, the cut beginning just beyond the throat and running deeper toward the muzzle, where the section became square, with the same area as the muzzle of No. 14. No. 16 was like No. 14 but had a larger throat area so that a needle point could be introduced to give the same net area as No. 14. No. 9 and No. 13 are search-tube nozzles made with throat and muzzle areas large enough so that the net areas, with the search tube in place, were equal to the net areas of the corresponding "reaction" nozzles; that is the nozzles whose efficiency was determined by reaction alone without the use of the search tube. No. 9 corresponds to Nos. 10, 11, 12 and 18. No. 13 corresponds to Nos. 14, 15, and 16. The dimensions of the nozzles are given in Tables 1 and 2.

FLOW TESTS

16 Numerous tests were made to determine the rate of flo.. of steam through the various nozzles. Fig. 8 shows the results of these tests plotted to a scale of pounds flow per hour. The variations in flow are probably due principally to moisture in the steam, and to

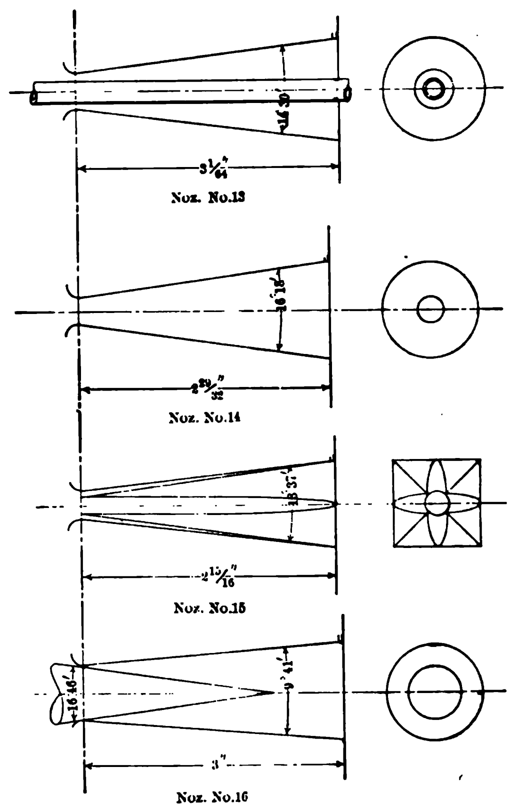
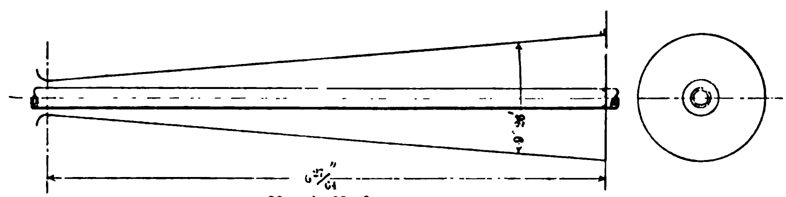


Fig. 6 Forms of Nozzles Tested

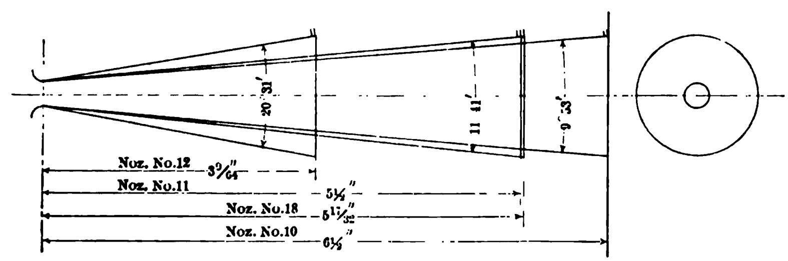
some extent to leakage from the water to the steam side of the condenser. The condenser was tested and at no time showed a leak exceeding two pounds per hour. There was sometimes a trace of superheat at the nozzle entrance and this increased with an increase in the volume of flow. At pressures of less than 145 lb., moisture was probably always present. For this reason the values used in the calculations were the mean flow values for 145-lb. pressure, and a trifle less than the mean for the lower pressures. It is to be regretted that we were unable to procure a calorimeter of sufficient accuracy for our purpose.

TABLE 1 DIMENSIONS OF NOZZLES IN FIG. 6

Nozzle No.	Gross Diameter Throat inches	Gross Diameter Outlet inches	Net Area Throat sq. ins.	Net Area Outlet sq. ins.	Length inches	Angle of Divergence
13	0.3949	1.156	0.0734	1.0005	3 1/16	14° 30′
14	0.3038	1.128	0.0725	0.9993	2 13/16	16° 18′
15	0.3038	1 in. sq.	0.0725	1.0000	2 11/16	13° 37′
16	0.625	1.1315	0.0725	1.0055	3	9° 41′



Nozzle No.9



Nozzles No.10, No.11, No.12, and No.18

Fig. 7 Forms of Nozzles Tested

TABLE 2 DIMENSIONS OF NOZZLES IN FIG. 7

Nozzle No.	Gross Diameter Throat inches	Gross Diameter Outlet inches	Net Area Throat sq. in.	Net Area Outlet sq. in.	Length inches	Angle of Divergence
9	0.3940	1.4505	0.0728	1.6033	6 27/64	9° 26′
10	0.3039	1.4240	0.0726	1.5926	6 1/8	9° 53′
11	0.3039	1.4231	0.0725	1.5907	5 1/8	11° 41′
12	0.3056	1.4241	0.0733	1.5930	3 9/64	20° 31′
18	0.3039	1.4225	0.0725	1.5893	5 11/32	11° 36′

17 In Fig. 8 the results are given in pounds per hour for the four initial pressures. Each small circle represents the result of one flow test of from 15 min. to 30 min. duration. The vertical dotted lines represent the flow values that were used in the efficiency calculations. The flow values for the nozzles of Fig. 6 are a little higher than the others, as is shown in the upper part of the diagram. The diagonal lines simply connect together the results found in the same test. For example, the five circles along the lowest line of the chart represent the values found for nozzle No. 9 on January 17 and 18, 1908. The vertical position of the observation point is of no significance, each initial pressure being placed higher up on the diagram than the preceding one, as a matter of convenience.

SPRING CALIBRATIONS

18 The accuracy with which the reactions were determined depended largely on the care taken in calibrating the springs. These calibrations were first made with the spring in place in the apparatus, and at the room temperature of about 85 deg. fahr. The readings were taken while gradually loading the spring to 25 lb. and then unloading it, repeating the operation a great number of times and taking the average extension under any given load as the true extension for that load. The average extension under a load of 25 lb. was 1.4875 in., or 0.0595 in. per lb. In cooling off, tube A, Fig. 4, seemed to warp a little, so that after about two minutes there was a decrease in the initial extension of the spring of about 0.0025 in. As it took about two minutes to shut off the steam and get the initial extension after each reaction reading, the spring extensions were all corrected by this amount.

19 Another factor which affected the spring calibration was the change in temperature of the spring itself. A thermometer was inserted in the spring casing and the spring calibrated at various temperatures by observing the temperature and extensions simultaneously. From these temperature calibrations a correction factor $l\,[0.0002423\ (t_2 - t_1)]$ was obtained and used to correct the reactions found by using the factor 0.0595 in. per lb.

20 After the thermometer had been inserted in the spring casing it was noted that as the reaction test progressed the temperature at first increased and then remained nearly constant regardless of moderate changes in initial and terminal pressures. This was due to the fact that the box D stood open and cooled off between tests and then

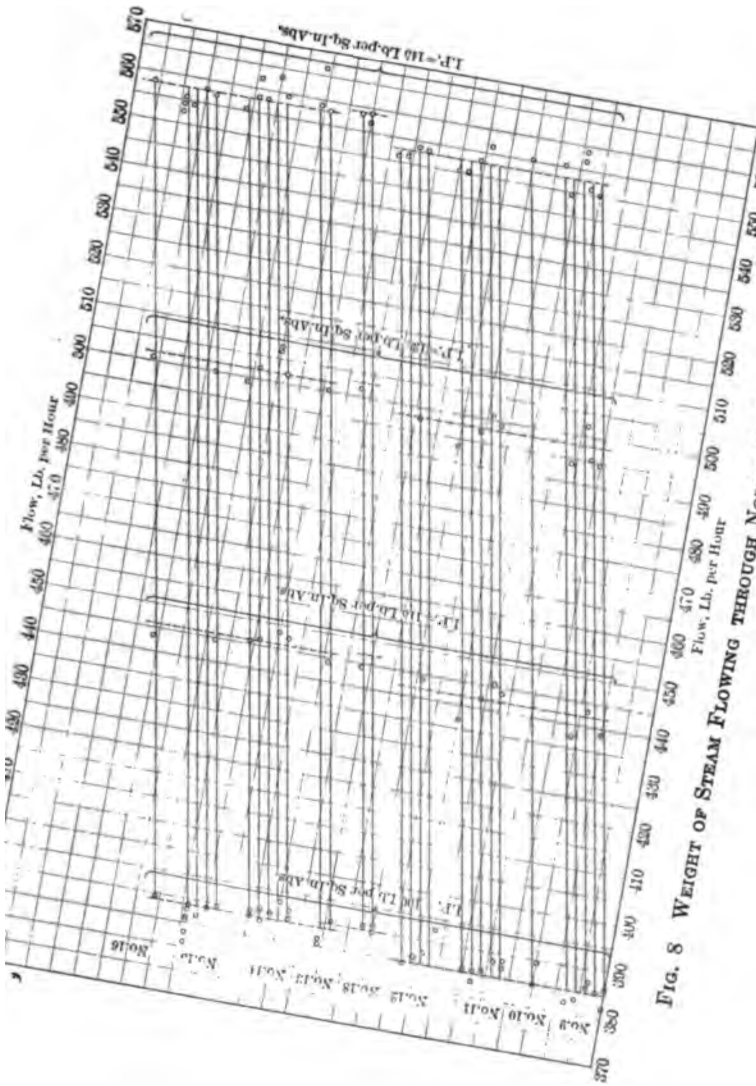


FIG. 8 WEIGHT OF STEAM FLOWING THROUGH NOZZLES ON DIFFERENT DATES

warmed up gradually when the steam was turned on. The average temperature in the spring casing was about 135 deg. fahr., or 50 deg. higher than the room temperature at which the original calibration was made. All the reactions found before this thermometer was in place were corrected on the assumption that the spring temperature was 135 deg. fahr. While this did not eliminate the error due to the fact that the temperature increased during the first part of each test, it did bring the average quite close to what it should be.

21 Corresponding readings of spring extensions at the beginning and end of tests were lower and higher respectively than the average extension. This was due to the above-mentioned difference in spring temperature. When the spring temperature was read simultaneously with the spring extension this difference disappeared.

SEARCH-TUBE TESTS

22 After completing the flow tests and the spring calibrations one other factor remained to be determined before the reaction tests could be made. This was the determination of the pressure at the muzzle of the nozzle.

23 *The reaction of any nozzle is equal to the summation of all the components, parallel to its axis, of the pressures within the nozzle and in the chamber from which it leads.* If the pressure of the medium surrounding the nozzle and the chamber is equal to that in the plane of the muzzle, then the reaction as shown by the pull on the spring is the true reaction. If the pressure of the surrounding medium is greater than that in the plane of the muzzle it will decrease the apparent reaction, and if the pressure of the surrounding medium is less than that in the plane of the muzzle it will increase the apparent reaction. The amount of such increase or decrease will be equal to the difference in the unit pressure multiplied by the area of the muzzle. The true reaction of the nozzle is equal to the pull of the spring plus or minus this pressure difference.

24 The demonstration of this proposition possibly differentiates these experiments from those heretofore published, as the writers do not know of any other case where the combination has been used in this manner.

25 The muzzle pressure was found by using the search tube with nozzles No. 9 and No. 13. The search tube here used was a selected piece of cold-drawn Shelby tube ¼ in. in outside diameter and $\frac{3}{16}$ in. inside diameter, with six holes $\frac{1}{32}$ in. in diameter drilled perpendicular to the axis. The outside of the tube was polished to micrometer

measurement. The chamber B was rigidly connected to the back wall of the box D by the distance piece at J (Fig. 4). The rear end of the search tube was encased and supported by a tube at L which had on its outer surface a thread fitted with a micrometer nut, and passed through the distance piece holding the search tube in the axis of the nozzle.

26 The holes in the search tube were located in the same plane as the hole in the wall of the nozzle. One gage was connected to the box D. another to the hole in the wall of the nozzle, and a third to the rear end of the search tube. Simultaneous readings of these gages were taken with varying pressures in the box. These readings were plotted in Fig. 9 and Fig. 10.

EXPLANATION OF RESULTS, FIGS. 9 AND 10

27. The diagonal lines represent the box pressures, or external terminal pressures which were varied from 0.5 lb. to 2 lb. absolute. The circles represent the pressures at the rim of the nozzle and the dots those at the center of the nozzle. These are connected by vertical lines as an aid in following the chart. The results are plotted for the four initial pressures as found on different dates. The dotted horizontal lines represent the muzzle pressures used in determining the true reaction. For example, on May 22 and 23, with an initial pressure of 100 lb. under a box pressure of 0.8 lb. absolute the terminal pressure at both rim and center of the nozzles was 0.7 lb. absolute; 0.649 was the average terminal pressure used in the calculations.

28 To illustrate further, the data for nozzle No. 13, April 15 and 16, 1908, with an initial pressure of 145 lb. per sq. in., are as follows:

Box pressure..........	0.384	0.84	1.33	1.67	1.71	1.81	1.995	2.315	2.80
Rim pressure..........	1.665	1.67	1.665	1.67	1.68	1.685	1.725	1.865	2.23
Search pressure.......	1.71	1.715	1.72	1.705	1.705	1.72	1.725	1.73	1.73

The three pressures in each column were read simultaneously and recorded.

29 The box pressure into which the nozzle discharged is represented arbitrarily by the diagonal line AB, in Fig. 10. The point c on this line shows the box pressure 0.84 lb. per sq. in. given in the table.

30 The corresponding rim pressure 1.67 lb. per sq. in., (that given by the gage connected with the hole in the wall of the nozzle at the muzzle) is plotted as a circle vertically above the point.

31 The corresponding search pressure 1.715 lb. per sq. in., (that obtained by means of the search-tube with the holes in the same plane as the hole in the wall of the nozzle) is plotted as a dot vertically above the point *c*.

32 The first tests seemed to indicate a higher pressure in the center of the stream than in the rim. Later a leak in the search-tube connections was discovered and repaired and the tests repeated, with the result that there was practically no differences between the pressures at the rim and center of the jet. It was therefore assumed that such differences as had occurred were due to the leaky search-tube. As it was difficult to keep the search-tube connections perfectly tight, it was decided to use the rim readings.

33 The tests from which Fig. 9 and Fig. 10 are plotted show that, after the leak in the search-tube had been eliminated, the pressure at the rim and that at the center of the stream did not differ by an appreciable amount. They also show that the pressure within the nozzle did not vary as the external pressure was decreased below that in the muzzle of the noazzle, and that as the external pressure was raised above the muzzle pressure, it crept in gradually along the wall of the nozzle, but did not affect the pressure at the center of the stream within the limits shown.

REACTION TESTS

34 In making reaction tests the following routine was observed: The barometer was read with every observation, and corresponding corrections made so that the initial absolute pressure used should be 145, 130, 115 and 100 lb. per sq. in. This was for convenience in making computations. One observer maintained a constant initial pressure by manipulating the needle valve in the steam pipe A second observer maintained a constant vacuum in the box *D* by manipulating the valves leading to the pump and condenser. A third operated the micrometer screw which registered the spring extension and thus held the multiplying-needle opposite an index at the center of its travel. A fourth and sometimes a fifth man read gages, and one man was generally occupied in moving about behind the observers to check observations.

35 As the reaction of the jet forced the chamber *B* back against the tension of the spring, the micrometer screw was worked forward until the tension in the spring balanced the reaction and the multiplying-needle indicated that *B* was swinging freely in the central

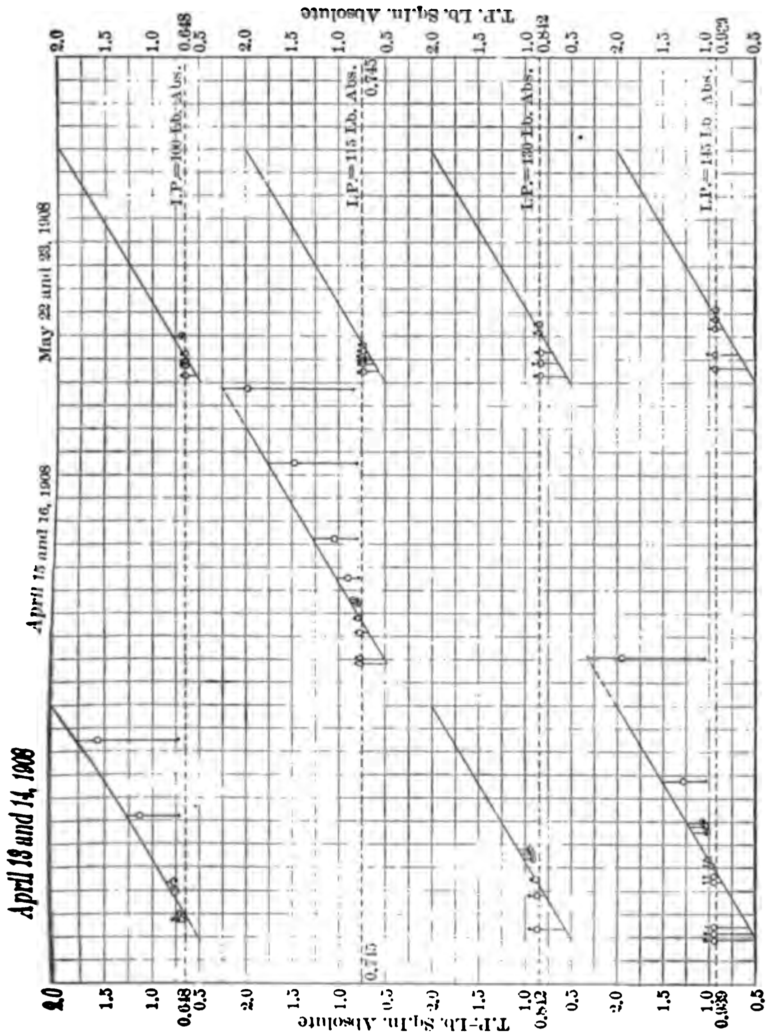


FIG. 9 EFFECT ON TERMINAL PRESSURE OF NOZZLE, PRODUCED BY VARYING PRESSURE IN BOX IN WHICH NOZZLE WAS SUSPENDED

THIS IS FOR NOZZLE NO. 9, WHICH CORRESPONDS TO NO. 11.

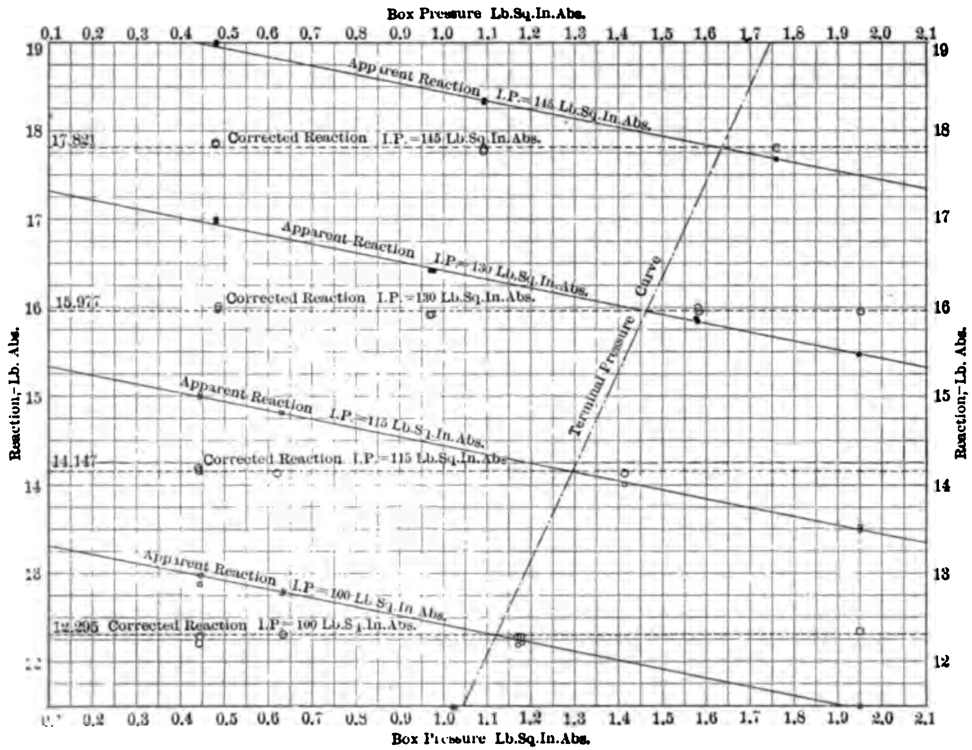


FIG. 10 EFFECT ON TERMINAL PRESSURE OF NOZZLE, PRODUCED BY VARYING PRESSURE IN BOX IN WHICH NOZZLE WAS SUSPENDED

THIS IS FOR NOZZLE NO. 13, WHICH CORRESPONDS TO NO. 14.

position. When the needle remained quiet for an instant in this position the observer gave a signal and the thermometer and gages were read simultaneously. After each reading the steam was shut off, and the position of the micrometer screw with the needle in the central position was noted. The difference between this position and that when the steam was flowing gave the elongation of the spring due to the reaction.

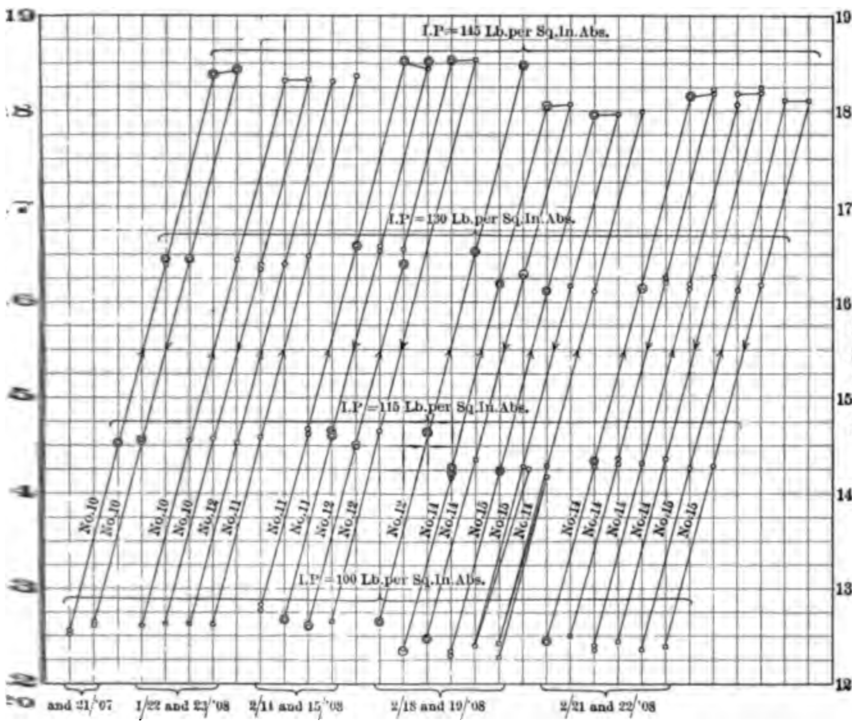


FIG. 11 PRELIMINARY REACTION TESTS

36 It will be noted that friction has been entirely eliminated in this apparatus, except for the trifling amount due to the movement of the multiplying-needle.

CALCULATION FOR EFFICIENCY

37 A series of preliminary reaction tests was run on all of the nozzles with the result that nozzles No. 11 and No. 14 were selected as

representative for future tests. The results of the preliminary tests upon these nozzles are given in Fig. 11. Exhaustive tests of long duration were then made upon nozzles No. 11 and No. 14 with the results shown in Fig. 13 and Fig. 14, and in Tables 3 and 4. Fig. 12 shows the comparative results of certain of the tests.

38 In Fig. 11 and Fig. 12 the reactions in pounds absolute are grouped together horizontally. The reactions are shown for each of the four initial pressures. The vertical scale is four lines to the pound. The horizontal position of the points has no significance, the

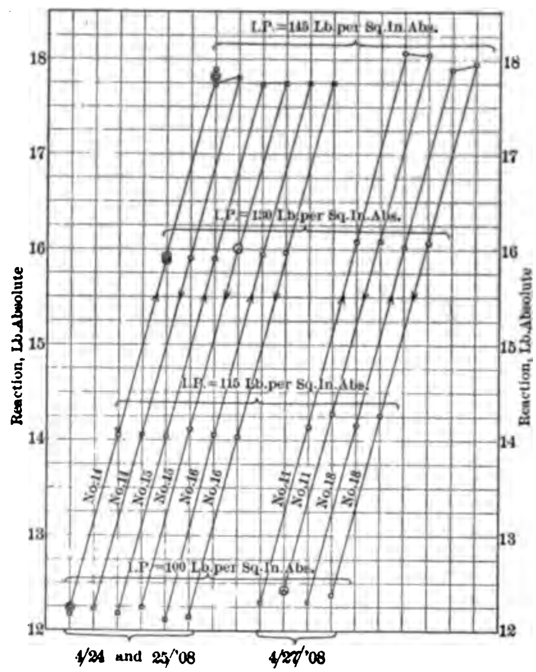


FIG. 12 COMPARISON OF TESTS ON NOZZLES 14, 15 AND 16 WITH THOSE ON NOZZLES 11 AND 18

initial pressures being placed diagonally over each other as a matter of convenience. The diagonal lines connect the reactions of a single nozzle at the various pressures. The arrowheads indicate whether the reactions shown on that line were taken when the pressure was increasing or decreasing. The circles represent the actual reactions in pounds as plotted from the tests. Two circles occurring together indicate that two independent readings of the reaction were taken at the same time.

39 Fig. 13 shows graphically the result of a complete series of reaction tests on nozzle No. 11, which is practically the same as nozzles Nos. 9, 10, 11, 12 and 13. The vertical scale is 4 lines to the pound, while the horizontal scale is 20 lines to the pound. The full diagonal lines connect together the observed reactions in pounds under a

TABLE 3 COMPUTATION OF EFFICIENCY, NOZZLE NO. 11

I. P. Abs.	T. P. Lb. per Sq. In. Abs.	Reaction	Flow Lb. per Hr.	Flow Lb. per Sec.	Vel.	B.t.u.$_1$	B.t.u.$_2$ Table	Eff. %
145	0.929	18.134	553	0.1536	3796	288.0	317.4	90.75
145	1.029	17.974	553	0.1536	3763	283.0	312.5	90.55
145	0.829	18.294	553	0.1536	3830	293.1	322.7	90.84
145	0.929	18.134	558	0.1550	3762	282.9	317.4	89.13
145	0.929	18.134	548	0.1522	3831	293.3	317.4	92.41
145	0.929	18.234	553	0.1536	3817	291.2	317.4	91.75
145	0.929	18.034	553	0.1536	3776	284.9	317.4	89.75
130	0.832	16.244	498	0.1383	3776	285.0	315.4	90.36
115	0.735	14.351	442	0.1228	3759	282.4	313.6	90.03
100	0.638	12.45	385	0.1069	3744	280.1	311.5	89.91
100	0.738	12.29	385	0.1069	3696	273.0	304.6	89.63
100	0.538	12.61	385	0.1069	3792	287.4	319.5	89.94
100	0.638	12.45	390	0.1083	3696	273.0	311.5	87.62
100	0.638	12.45	380	0.1056	3793	287.5	311.5	92.29
100	0.638	12.55	385	0.1069	3774	284.6	311.5	91.36
100	0.638	12.35	385	0.1069	3714	275.6	311.5	88.47
			Assuming 2 per cent moisture					
145	0.929	18.134	553	0.1536	3796	288.0	311.9	92.35
130	0.832	16.244	498	0.1383	3776	285.0	310.0	91.93
115	0.735	14.351	442	0.1228	3759	282.4	308.2	91.62
100	0.638	12.45	385	0.1069	3744	280.1	306.1	91.51

B.t.u. are given only to the nearest tenth, and for this reason efficiencies are not accurate in second decimal place.

B.t.u.$_1$ = equivalent of kinetic energy of jet in B.t.u.

B.t.u.$_2$ = available heat energy of steam.

varying box pressure. The horizontal dotted lines connect together the same reactions after being corrected for the difference between the pressure in the box and in the muzzle of the nozzle. The dotted diagonal line represents the pressure in the muzzle of the nozzle as found with the search tube and is plotted to the same horizontal scale as the box pressure. When the box and terminal pressures are the same the apparent and corrected reactions are the same, as is shown by the intersection of the three lines. As the box pressure increases above that point where the terminal and box pressures were the same the apparent reaction falls below the true reaction, and as the box pressure falls below that point where the terminal

TABLE 4 COMPUTATION OF EFFICIENCY, NOZZLE NO. 14

SHOWING EFFECT OF ERROR IN DETERMINATION OF TERMINAL PRESSURE FLOW OR REACTION

I. P. Abs.	T. P. Lb. per Sq. In. Abs.	Reaction	Flow Lb. per Hr.	Flow Lb. per Sec.	Vel.	B.t.u.₁	B.t.u.₂ Table	Eff. %
145	1.632	17.821	558	0.1550	3698	273.2	289.8	94.28
145	1.732	17.721	558	0.1550	3677	270.2	286.7	94.23
145	1.532	17.921	558	0.1550	3718	276.3	293.0	94.30
145	1.632	17.821	563	0.1564	3665	268.4	289.8	92.61
145	1.632	17.821	553	0.1536	3731	278.2	289.8	95.99
145	1.632	17.921	558	0.1550	3718	276.3	289.8	95.34
145	1.632	17.721	558	0.1550	3677	270.2	289.8	93.22
130	1.46	15.977	502	0.1394	3685	271.3	288.2	94.15
115	1.288	14.147	446	0.1239	3672	269.5	286.5	94.07
100	1.116	12.295	389	0.1081	3659	267.6	284.7	93.99
100	1.216	12.195	389	0.1081	3630	263.3	280.3	93.92
100	1.016	12.395	389	0.1081	3689	272.0	289.3	94.01
100	1.116	12.295	394	0.1094	3613	260.8	284.7	91.62
100	1.116	12.295	384	0.1067	3707	274.6	284.7	96.45
100	1.116	12.395	389	0.1081	3689	272.0	284.7	95.53
100	1.116	12.195	389	0.1081	3630	263.3	284.7	92.47
				Assuming 2 per cent moisture				
145	1.632	17.821	558	0.1550	3698	273.2	284.7	95.97
130	1.46	15.977	502	0.1394	3685	271.3	283.1	95.84
115	1.288	14.147	446	0.1239	3672	269.5	281.5	95.74
100	1.116	12.295	389	0.1081	3659	267.6	279.7	95.67

B.t.u. are given only to the nearest tenth, and for this reason efficiencies are not accurate in second decimal place.

$B.t.u._1$ = equivalent of kinetic energy of jet in B.t.u.

$B.t.u._2$ = available heat energy of steam.

and box pressures were the same, the apparent reaction rises above the true reaction. As an example, the terminal pressure at 115 lb. initial pressure is approximately 0.7341 lb. When the box pressure was increased to 0.89 the apparent reaction was 14.10 lb. To this apparent reaction was added the correction factor (difference in

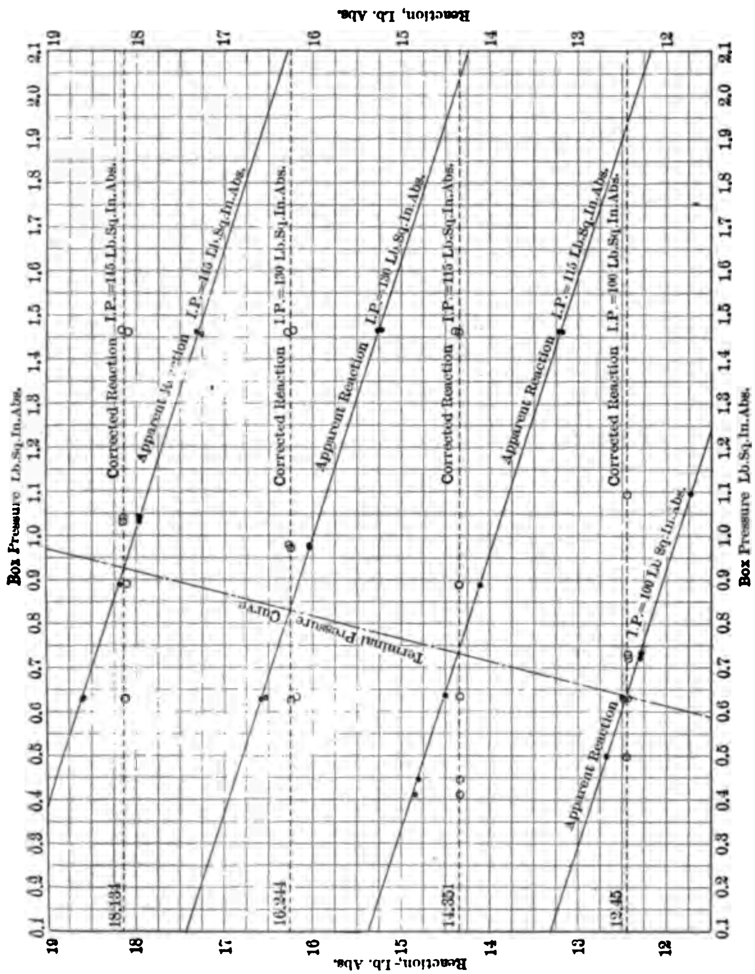


FIG. 13 REACTION TESTS OF NOZZLE NO. 11 WITH VARYING BOX PRESSURE

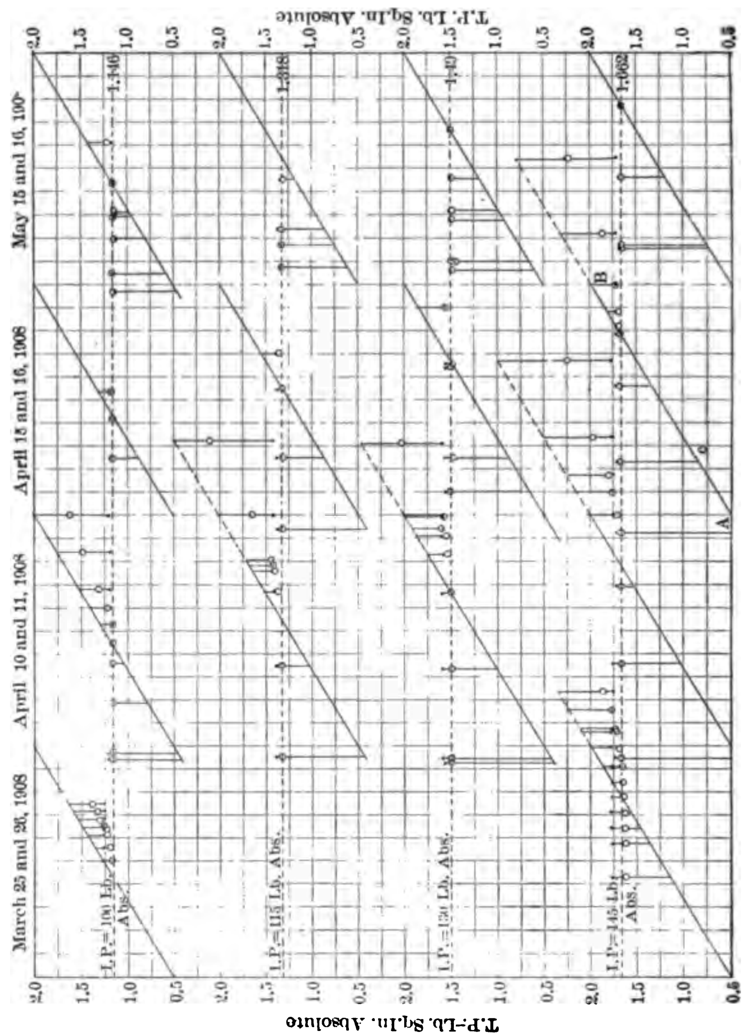


Fig. 14 Reaction Tests on Nozzle No. 14 with Varying Box Pressure

pressure × area of muzzle) to give the corrected reaction 14.351. Fig. 14 is like Fig. 13, the values being for nozzle No. 14.

40 The method of calculation may be illustrated by the following from nozzle No. 14 with an absolute initial pressure of 145 lb. per sq. in. The terminal pressure (See Fig. 10) is 1.662 lb. per sq. in., and deducting 0.03 this becomes 1.632 lb. per sq. in. The reaction (Table 4) is 17.821 lb. The flow is 558 lb. per hr. or 0.155 lb. per sec.

$$\text{Velocity} = V = \frac{\text{reaction} \times g}{\text{flow (lb. per sec.)}} = \frac{17.821 \times 32.16}{0.155} = 3697.6 \text{ ft. per sec.}$$

$$\text{B.t.u.}_1 = \text{kinetic energy of jet} = \frac{V^2}{2g \times 778} = \frac{3697.6^2}{2 \times 32.16 \times 778} = 273.22$$

$$\text{B.t.u.}_2 = \text{available energy (from steam table)} = 289.8$$

$$\text{Efficiency} = \frac{\text{B.t.u.}_1}{\text{B.t.u.}_2} = \frac{273.22}{289.8} = 0.9428 \text{ or } 94.28 \text{ per cent}$$

41 If the terminal pressure had been determined as 1.732 and no other factor changed, the true reaction as calculated would have been 17.721 and the resulting efficiency would be 94.23 per cent.

42 If the flow had been determined as 563 lb. per hr. with no other change of values the efficiency would have figured 92.61 per cent.

43 A reaction of 17.921 without other change would have given an efficiency of 95.34 per cent.

44 If we assumed 2 per cent of moisture, the efficiency would figure as 95.97 per cent.

45 The efficiency was also calculated by the search-tube method, by first plotting curves (similar to those shown in Fig. 15) showing the relation between pressure and rate of flow per unit area of section, with adiabatic expansion and with various percentages of friction loss. The pressure and flow found by experiment were then plotted on this chart and the efficiency determined graphically by comparison.

46 The chart in Fig. 15 was designed for finding the nozzle efficiency by the search-tube method. The vertical scale is 20 lines to the pound. The horizontal scale is 10 lb. per hour per line. The chart shows the relation between the flow in pounds per square inch of section and the pressure in any section of the nozzle, assuming adiabatic expansion for the lower curve and 5 per cent loss of

heat for the upper one. The data for plotting these curves were obtained from the steam tables. By plotting the observed values of pressure and flow upon these sheets we are able to obtain a graphic solution for efficiency. For example, in the case of nozzle No. 9 the terminal pressure at 100 lb. initial pressure was found to be 0.638 lb. per sq. in. and the corresponding flow in pounds per square inch of section is 238 ±. From the chart, assuming adiabatic expansion, under terminal pressure of 0.638 the flow would be 248 lb. and 238/248 is approximately 96 = the efficiency.

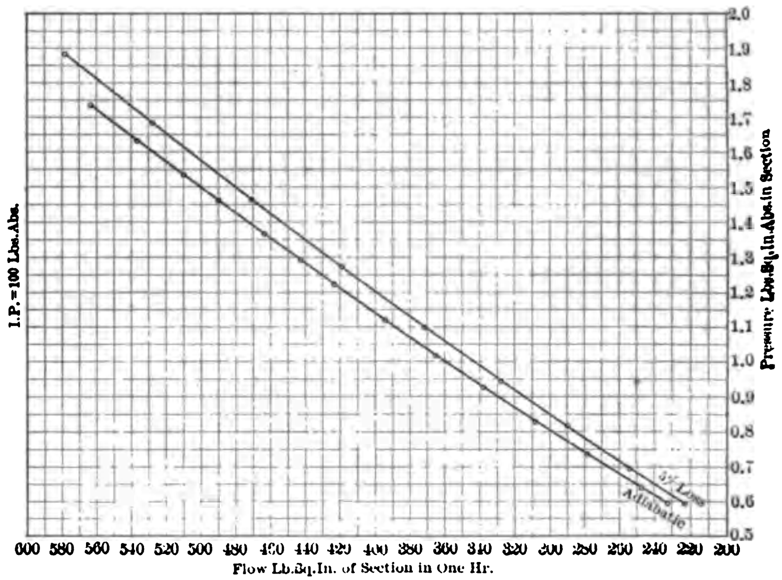


Fig. 15 Pressure-Flow Chart

RESULTS AND CONCLUSIONS

47 Table 5 gives a summary of the calculations for nozzles No. 9, 11, 13 and 14, and Table 6*a* summary showing what would be the effect of error in the observed values.

48 The efficiencies for nozzles No. 9 and No. 13 were obtained graphically by the method described in Par. 46. The discrepancy between the efficiencies calculated for the search tube and the reaction nozzles is due principally to the great difference of the effect in the two methods of calculationca used by a small error in terminal

TABLE 5 SUMMARY OF CALCULATIONS OF EFFICIENCY FOR NOZZLES
Nos. 9, 11, 13 AND 14

Nozzle No.	I. P. Lb. per Sq. In. Abs.	T. P. Lb. per Sq. In. Abs.	Flow Lb. per Hour	Reaction Pounds	Theoret. B.t.u. per Lb.	Calcul. B.t.u. per Lb.	Theoret. Vel. Ft. per Sec.	Calcul. Vel. Ft. per Sec.	Eff. %
9	100	0.638	385	....	....	....	....	....	96.1
	115	0.735	442	....	....	....	....	....	96.3
	130	0.832	498	....	....	....	....	....	96.0
	145	0.929	553	....	....	....	....	....	95.5
11	100	0.638	385	12.45	311.5	280.1	3948	3744	89.9
	115	0.735	442	14.351	313.6	282.4	3962	3759	90.0
	130	0.832	498	16.244	315.4	285.0	3973	3776	90.4
	145	0.929	553	18.134	317.4	288.0	3985	3796	90.7
13	100	1.116	389	....	....	....	....	....	98.3
	115	1.288	446	....	....	....	....	....	97.9
	130	1.46	502	....	....	....	....	....	97.5
	145	1.632	558	....	....	....	....	....	97.1
14	100	1.116	389	12.295	284.7	267.6	3774	3659	94.0
	115	1.288	446	14.147	286.5	269.5	3796	3672	94.1
	130	1.46	502	15.977	288.2	271.3	3798	3685	94.2
	145	1.632	558	17.821	289.8	273.2	3808	3698	94.3

TABLE 6 EFFECT OF ERROR IN OBSERVATIONS

Nozzle No.	Observation	Error of Observations	Corresponding % Error in Efficiency	
		+	+	−
9	Terminal Pressure	0.1 lb. per sq. in.		8.5 to 14.0
11	Terminal Pressure	0.1 lb. per sq. in.		0.03 to 0.3
13	Terminal Pressure	0.1 lb. per sq. in.		5.4 to 9.2
14	Terminal Pressure	0.1 lb. per sq. in.		0.02 to 0.07
9	Flow	5 lb. per hr.	1.0 to 1.6	
11	Flow	5 lb. per hr.		1.6 to 2.4
13	Flow	5 lb. per hr.	1.1 to 1.5	
14	Flow	5 lb. per hr.		1.67 to 2.46
9	Dryness Factor	2%	0.9 to 0.6	
11	Dryness Factor	2%		1.6
13	Dryness Factor	2%	0.9 to 0.7	
14	Dryness Factor	2%		1.7
11	Reaction	0.1 lb.	1.0 to 1.45	
14	Reaction	0.1 lb	1.05 to 1.54	

Note.—An error of +0.1 lb. would be caused in the calculated reaction by an error of +0.1 lb. per sq. in. in the box pressure-reading of No. 14 or by an error of + 0.0628 lb. per sq. in. in the box pressure-reading of No. 11.

pressure. An increase of only 20 lb. per sq. in. over the values used would cover the discrepancy. Corrections for any slight condenser leak which may have existed would decrease the flow values and bring the calculated efficiencies closer together.

49 The "calculated velocity" for No. 11 and No. 14 is obtained as described in Par. 40, from the reaction; and this was obtained from the series of interconfirmatory tests plotted in Fig. 13 and Fig. 14.

50 The terminal pressures chosen were the minimum observed values as determined with the search tube in the muzzles of the nozzles. The dryness factor was assumed as 100 per cent. Assuming a 2 per cent moisture would make the calculated efficiencies for No. 13 and No. 14 very nearly equal, and very materially reduce the difference between No. 9 and No. 11.

51 In consideration of the above, taken in connection with Tables 5 and 6, we may assume that the values 91.5 per cent for No. 9 and No. 11, and 95 per cent for No. 13 and No. 14, are probably within 2 per cent of the true efficiencies. No. 15 and No. 16 show a trifle less reaction than No. 14 but the flow also appears to be a trifle less, and there is not sufficient ground for assuming any difference between the efficiencies of these three nozzles. Neither is there any appreciable difference in Nos. 10, 11 and 12. No. 18, with a greater flow and less reaction than No. 11, shows an efficiency of about three per cent less. Since no appreciable difference in efficiency is shown either with a variation in cone angle from 9 deg. to 20 deg., or with such variations in contour as shown in nozzles No. 15 and No. 16, smoothness of finish would appear to be a much more important factor than the exact contour.

52 The specific volume of the steam can be calculated from the data in Table 5 and is greater than would be calculated by assuming adiabatic expansion, because the dryness factor has been raised by the friction.

53 In Tables 3 and 4, lines 1, 8, 9 and 10 contain the same values as those given in Table 5; and in the other lines assumption is made of certain values other than the observed values, to show what would be the effect of error in observation, as is illustrated in Par. 40 to 44 inclusive and tabulated in Table 6.

54 The values in the last column of Tables 3 and 4 are given to the second decimal place, to show what would be the effect of certain errors in observation. In Table 5 they are given to the first decimal place, to avoid conveying the erroneous impression that some of the values as computed from observations were absolutely identical.

DISCUSSION

PROF. J. A. MOYER. The methods used in these tests are obviously much more accurate than the impact plate devices used by Lewicki in his experiments with De Laval nozzle and by others who have conducted similar investigations more recently.

2 The high efficiencies obtained may be surprising to some who have not followed the latest developments in the designing of steam nozzles. Results of this investigation confirm in general the results given by Steinmetz[1] and by the writer showing that the efficiency of a well-designed nozzle for relatively large, as well as for small, limits of pressure will be above 97 per cent.

3 However, in one respect the investigation is not as complete as it was hoped it would be. There are not enough data to determine the effect on the efficiency of varying the length of a nozzle: that is, nozzles of different lengths, but with the same taper or angle of divergence, should be compared. However, the statement is made in the paper that there is no appreciable difference in the efficiencies of nozzles 10, 11 and 12, which, however, do not have the same taper, but have the same *areas* at the throat and at the mouth. It is probable that all of these nozzles were longer than they should be to obtain the highest efficiencies. More data are needed about the best length of the nozzle for a given expansion. Lewicki's experiments cover the two extremes: nozzles which are obviously too short, and those which resemble in proportions the ones used in this investigation.

4 The error due to moisture in the steam could not readily be determined, and while it is probably not large, yet this uncertainty might have been avoided by using superheated steam. The reaction in a nozzle due to the flow of superheated steam is apparently constant for a varying amount of superheat. This can be shown by the usual thermodynamic equations for flow and velocity—which determine the impulse force of a jet— and by the experiments of Lewicki[1] on the flow of superheated steam through De Laval nozzles. It should be observed however, that when these tests were started, Knoblauch and Jakob had not yet published the values which we are now using for the specific heat of superheated steam, and for this reason alone it was desirable to avoid the use of superheated steam.

[1] The Journal; Am. Soc. M. E., May 1908, p. 628.

[1] Mitteilungen über Forschungsarbeiten, Heft 12, Zalentafel 9 (*c*). Verein deutscher Ingenieure, 1904.

5 It has not been mentioned by the authors of this paper that their method can be used to calculate the *apparent* efficiency of any nozzle for any initial and final pressures. By measuring the areas of a nozzle at the throat and at the mouth or "muzzle," the expansion ratio in a nozzle is determined, and by means of empirical equations, due to Zeuner and others,[1] the ratio of the corresponding initial and final pressures giving the highest efficiency, can be obtained. This ratio of pressures would correspond to the condition in these tests where the terminal and box pressures are the same.

6 If the ratio of the initial to the final pressure has been determined, either of these pressures can be readily calculated if the other is known. For example, if by measurement of the mouth and the throat areas, the expansion ratio of the nozzle is found to be, say, 3, then the ratio of the initial to the final pressure must be nearly 13.3 for the maximum efficiency of the jet discharged from it. For this nozzle, therefore, with an initial pressure of 200 lb. absolute, the final pressure should be 15 lb. absolute. From the equations given in Par. 4 of the paper, the theoretical reaction can be readily calculated from the available energy corresponding to the pressure limits. The change in reaction due to final pressures different from those for which a nozzle is designed is, then, according to the method presented here, the product of the area of the mouth of the nozzle, times the difference between the correct final pressure for the nozzle—in this case 15 lb. absolute—and the pressure in the box, or in practice the pressure inside the casing of a stage of a turbine. Since reaction and velocity are directly proportional—*with constant flow*—the apparent velocity of the jet will be increased or decreased in the same proportion as the reaction is increased or decreased.

7 In actual practice, however, this does not occur. It has been observed that if a nozzle is used which does not expand the steam sufficiently, there is not nearly so much loss in the velocity of the jet as when the nozzle is too wide at the mouth and "over-expands" the steam. In other words, it has been found that a nozzle which is about 25 per cent too large in area at the mouth, will give to the jet only 90 per cent of the theoretical velocity, while one which is too small by the same percentage will give within 2 or 3 per cent of the maximum efficiency obtainable with the pressures best suited. All this involves something which is not taken into consideration in these reaction experiments; and for that reason, the results obtained

[1] J. A. Moyer, The Steam Turbine, p. 40–41.

by this method with varying back-pressures may possibly be misleading.

PROF. C. C. THOMAS. I have for years been interested in this line of investigation, and am glad to see this contribution. In Par. 23, the corrections which the authors make to the observed reactions seem to me to be somewhat open to question. Aside from this fact, I cannot quite see the theory upon which the corrections are based; the fact that the pressures vary considerably in all but perfect nozzles, from the center to the walls, and that very considerable irregularities of flow are found in nozzles, makes me doubt the necessity for making these corrections to the observed reactions.

STRICKLAND L. KNEASS. The tests appear to cover ordinary straight-tapered nozzles, as follows: 1 in 6, 1 in 5.77, 1 in 5, 1 in 4 and 1 in 3. In several cases the net areas vary slightly from these ratios owing to the displacement of the cylindrical search tube, but the ratio of the throat area to the outlet area is practically the same for all nozzles, so that the results relate chiefly to the effect of the length of the tubes and the friction upon the walls.

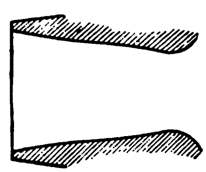


FIG. 1 SUGGESTED FORM OF NOZZLE

2 From Table 5 it would appear that nozzle No. 13, which has a taper of approximately 1 in 4, gives a much higher efficiency between 100 lb. and 145 lb. absolute, than nozzles Nos. 9, 11 and 14, with tapers ranging from 1 in 3½ to 1 in 6. As far as the knowledge of the writer extends, there is no logical reason for this result, and he would attribute the higher percentages to greater precision in the experiments rather than to any inherent efficiency in the 1-in-4 nozzle.

3 The correct contour of a nozzle for the discharge of an elastic fluid is still a moot question. After an extended series of experiments between the years 1888 and 1891 with steam nozzles of various tapers, the writer offered the suggestion that the section should be in the form of a reversed curve, somewhat as shown in Fig. 1 here-

with. This curve was based on the theory that the acceleration should be constant during the passage of the steam through the nozzle, and that the areas at the several sections should be unit distances apart. These sections were calculated with due allowance for the change in the specific volume of the steam during expansion. The results obtained seemed to confirm this theory and were compared with the discharge from straight-tapered nozzles in a paper read before the Engineers' Club of Philadelphia in 1891. The writer's opinion was further corroborated by F. Hodgkinson before the Engineers' Society of Western Pennsylvania in 1900. In view, therefore, of published experiments upon nozzles of special contour for which advantageous results were claimed, it is surprising that the authors of this paper did not increase its value by widening the scope of their experiments, instead of confining their tests to the oldest and possibly less efficient form of tube.

4 Referring again to the experiments of the writer, his conclusions covered the general theorem that there was little difference in the efficiency of the straight-tapered nozzle, so long as the terminal pressure of the steam within the tube was the same as that of the medium into which it flowed, and, further, that the terminal velocity would be the same under this given condition whether the taper were 1 in 6, 1 in 5, or 1 in 3. This opinion seems to be sustained by the authors, although the results are not satisfyingly definite, because different terminal pressures were used with each initial pressure and the table does not contain the terminal pressures within the nozzle, so that the comparison cannot be made with the pressure of the final medium.

5 It is desired that this point be emphasized, for a slight difference between these two pressures has an important effect upon the results. It is thought that a more exact method of determining the relative efficiency would have been to modify the length so that the terminal internal and external pressures would always be the same, for when an attempt is made to introduce minus or plus reaction for correction, doubt is thrown upon the result. This is especially obvious to any one who by careful observation of the flow of steam through and from nozzles of different proportions, has noted the unstable equilibrium of the jet when the terminal pressure of the medium exceeds that within the end of the nozzle. Some of the minor discrepancies may be charged to this item and the writer is somewhat skeptical as to the accuracy of the results obtained in practice when calculated under the theorem given in Par. 23.

6 It would have been interesting if the authors had recorded new data relative to the action of the steam within the nozzle and determined the terminal specific steam volume. The writer maintains that the specific gravity of steam at different sections of the nozzle does not correspond to that calculated by the thermo-dynamic equation, and therefore would be glad to have the authors state if the velocity of the steam, as given in Tables 3 and 5, is equal to the specific volume based upon the adiabatic equation, divided by the cross-sectional area.

7 A test of this kind should give the initial condition of the steam. The authors state that a thermometer was placed in a well at the rear of the nozzle, but there are no figures in the table giving the percentage of moisture. An objection to the construction of the apparatus can be offered in the liability of condensation of steam in the vertical flexible supply pipe. The steam flows downward under pressures varying from 100 lb. (328 deg. fahr.) to 145 lb. (356 deg. fahr.) and is surrounded by steam at a pressure of 28 in. vacuum (100 deg. fahr.) so that a certain amount is sure to be condensed.

The Authors. It appears to be a generally accepted fact that under-expansion in the nozzle is preferable to over-expansion. Stodola's Theory of Steam Shock and his search-tube experiments point very decidedly in this direction. Reaction experiments may even *appear* to indicate that under-expansion in the nozzle is in some cases preferable to using the theoretically correct ratio. This may also be true; but if the theory advanced in Par. 23 is correct, it is impossible to accept the results of any purely reaction experiments as giving a definite answer to this question; and where the pressure in the muzzle of the nozzle is not taken into account, all the results may be in error.

2 Of course, it is possible to calculate the muzzle pressure by theoretical and empirical formulæ; but if we are to rely upon theoretical formulæ there is no object in conducting tedious and expensive experiments. Moreover, empirical formulæ on this subject are at least liable to be based in part upon reaction tests which have not taken into proper account the pressure in the muzzle of the nozzle. Also when the nozzle discharges into a pressure which is considerably greater than the theoretical muzzle pressure, violent fluctuations occur within the nozzle itself, so that the formulæ do not apply and the results of reaction tests may become very misleading.

3 Par. 23 has been called in question from both the theoretical and the practical standpoint, so that a more extended consideration may not be out of order.

4 The first statement, "The reaction of any nozzle is equal to the summation of all the components, parallel to its axis, of the pressures within the nozzle and in the chamber from which it leads," can scarcely be questioned.

5 The net accelerating force F (Par. 4) which produces the velocity actually present in the muzzle of the nozzle may be divided into two parts. One part (call F_f) is a summation of components of the forces with which the internal walls react against the pressure of the steam. The second part is a force due to the pressure of the steam in the muzzle, and acts in opposition to the first.

6 Let F_m be this second part, P_m the muzzle pressure, and A the muzzle area. Then

$$F_m = P_m A$$

and

$$F = F_f - F_m = F_f - P_m A$$

Let R be the "true reaction of the nozzle," i. e., the force which is equal and opposite to F. Then

$$R = F = F_f - P_m A \tag{1}$$

7 The apparent reaction (called R_a) is the force with which the nozzle actually pulls or pushes in the direction opposite to the steam flow during the test. The apparent reaction of any nozzle is equal to the summation of the components parallel to its axis, of all the pressures, both internal and external, upon the walls of the nozzle and of the chamber from which it leads.

8 That part which is due to the internal wall pressure is equal to F_f. The external pressure acts, in the direction of flow of the jet, upon an area which is greater than that upon which it acts in the opposite direction, the difference being the area of the muzzle.[1]

9 Let P_e be the external pressure. Then

$$R_a = F_f - P_e A \tag{2}$$

Combining (1) and (2) we have

$$R = R_a = A (P_m - P_e) \tag{3}$$

10 The rest of Par. 23 accords with these equations.

[1] Gages connected to various points within the box showed that the external pressure did not vary in different parts of the box by as much as 0.01 lb. per sq. in. It must be remembered that the nozzle and the chamber from which it leads are here suspended within the box into which the jet discharges.

11 It is evident from this that any acceleration or retardation of the jet beyond the muzzle (due to the pressure into which it is discharged or to any other cause) cannot affect the true reaction, and that so long as the conditions within the jet are stable so that the muzzle pressure can be properly determined, there is no danger of being misled except by a failure to make the corrections.

12 When the pressure into which the nozzle discharges is considerably greater than the theoretical muzzle pressure, such violent flucuations ensue as to make all corrections impracticable, and the reaction tests under these conditions become worse than useless because they are misleading. The criticism by Professor Thomas is well founded with regard to such cases; but does not apply to the tests reported in this paper for the reason that these were all made under conditions which did not disturb the stability of the jet within the nozzle.

13 The fact that the corrected reactions shown in Fig. 13 and Fig. 14 lie in a horizontal line, i. e., are equal, is a further evidence that the theory upon which they are based is correct, also of the fact that the jet within the nozzle remained in very stable equilibrium, and that the creeping in of the external pressure along the internal wall had no practical effect, while the box pressure varied within the limits shown.

14 To show further the form of error involved in the failure to use these corrections, apparent and true reactions have been taken from Fig. 13 and Fig. 14, and the accompanying table computed.

Nozzle	t. p.	Flow Lbs. per sec.	Box t. p.	Reac.	Vel.	B. t. u.$_1$	B. t. u.$_2$ Table	Eff.
11	145	.1536	0.929	18.134*	3796	288.0	317.4	90.75
14	145	.1550	1.632	17.821*	3698	273.2	289.8	94.28
11	145	.1536	1.632	17.01†	3561	253.5	289.9	87.43
14	145	.1550	0.929	18.52†	3843	295.1	317.4	92.96
11	100	.1069	0.638	12.45*	3744	280.1	311.5	89.91
14	100	.1081	1.116	12.295*	3659	267.6	284.7	93.99
11	100	.1069	1.116	11.69†	3517	247.2	284.7	86.81
14	100	.1081	0.638	12.77†	3799	288.4	311.5	92.59

*Apparent and true.
†Apparent.

Data obtained from nozzles No. 11 and No. 14, with the box pressure equal to that in the muzzle of the nozzle, are given in lines 1 and 2. These velocities and efficiencies are the same as those given in Table 5, and require no correction for terminal pressure.

15 For line 3 the apparent reaction is taken for nozzle No. 11 with a box pressure which would be correct for No. 14, and the apparent velocity and efficiency of No. 11 are calculated from that basis.

16 For line 4 the apparent reaction for nozzle No. 14, with a box pressure which would be correct for nozzle No. 11, is similarly used.

17 It was found in the experiments plotted in Fig. 9 and Fig. 10, that the pressure conditions within the nozzle remained stable and practically constant with such variations from the proper box pressure for each nozzle. Also, by applying the corrections called for in Par. 23 it is found that these values reduce to the same values as those obtained in lines 1 and 2, showing that the velocity and efficiencies of the jets as they reached the muzzles were not affected by the changes in box pressure.

18 The acceptance of the uncorrected values would therefore imply an assumption that in nozzle No. 14, with an initial pressure of 145 lb. and a terminal pressure of 0.929 lb., the jet attained a velocity of 3698 ft. per sec. in the nozzle, and that after leaving the nozzle its velocity jumped to 3843 ft. per sec., and that in nozzle No. 11, with an initial pressure of 145 lb. and a terminal pressure of 1.682 lb., the velocity of the jet after leaving the nozzle dropped from 3796 ft. to 3561 ft. per sec.

19 The efficiencies calculated from the apparent reactions, if accepted in this form, would show that No. 14 is better than No. 11, not only for its own proper terminal pressures, but for the terminal pressures found in the muzzle of No. 11 as well. It may be that such is the case; but there is considerable probability of arriving at erroneous conclusions if it is assumed arbitrarily, without having first been proved by very careful experiments which are not in any manner dependent upon the assumption for their accuracy. There certainly is no basis for making such an assumption from these data as it has no bearing whatever upon the subject.

20 Previous to the time when this series of tests was begun, there had been considerable investigation of nozzles with cone angles up to 12 deg.; but the action of steam in nozzles of greater cone angle had not received the same degree of attention. It was therefore decided to use nozzles with divergence angles of from 9 to 20 deg., it being then thought that this upper limit might be beyond the value for highest efficiency.

21 Another set of nozzles tested contained one with a cone angle of 24 deg. 30 min., which seemed to show an equal efficiency with those of smaller angle. This set was made of babbitt metal, was not

perfectly smooth and was somewhat worn with long-continued use, so that the results could not be thoroughly checked.

22 With the steam conditions given and the ratio of muzzle to throat area determined therefrom, the only point left for the designer is the general contour of the nozzle, including the shape of cross section, length and angle or angles of divergence. The two sets of nozzles shown in Fig. 6 and Fig. 7 were designed with this in mind, each set having a common ratio of areas; those of Fig. 7 differing among themselves only in length and consequent angle of divergence, or vice versa, and those of Fig. 6 differing only in elements of general contour, not including length.

23 Professor Moyer's statement that "nozzles of different lengths, but with the same taper or angle of divergence, should be compared," is not understood, unless he means to suggest that the whole field of different steam expansion ratios should have been investigated. This was not permitted because of limitations of time and other circumstances familiar to most investigators. Such an investigation would not serve to determine the proper length for a given steam expansion ratio, because the different nozzles would not be suited to the same steam conditions; but it would give the efficiencies for one angle of divergence with all the pressure ratios to which the various nozzles were adapted.

24 Each set contained one search-tube nozzle for use in determining experimentally the terminal pressure in the muzzle, to be applied in reaction tests on the rest of the nozzles in that set. The efficiencies of these nozzles, No. 9 and No. 13, as calculated by the search-tube method, are shown in Table 5; but they are not worthy of consideration except as an example of the inaccuracies almost certain to be involved in this method. The high efficiency given for nozzle No. 13 is not due to greater precision in the experiments, as Mr. Kneass suggests, but rather to the great error in the search-tube method of calculation, caused by a very small error in determining the muzzle pressure. In Table 6 it is pointed out that a "+ error" of only 0.1 lb. per sq. in. in determining the terminal pressure would cause a "− error" of from 5.4 to 14 per cent in the "search tube computed" efficiency of No. 9 and No 13.

25 These "search-tube computed" efficiencies are evidently responsible for Mr. Moyer's statement that efficiencies were here found as high as 97 per cent. Values obtained from reaction tests are lower, and it is upon these that the conclusions stated in Par. 51 are based.

26 No. 9 ("search-tube" nozzle) was made with a small angle of divergence, to be doubly sure that the steam should not leave the walls before reaching the muzzle.

27 Both the length and the ratio of areas in nozzle No. 10 were made to correspond as nearly as possible with those in nozzle No. 9 so that the terminal pressure found in the muzzle of No. 9 might be applied to reaction tests upon the former with the least possible error.

28 No. 11 and No. 12 were made shorter and with a greater cone angle but with the same sectional areas, in order to find out what difference, if any, this would make in efficiency.

29 No. 18 was finished rough for comparison with No. 11, upon which the greater number of tests had been made.

30 No 14 was used to determine the efficiency with a smaller expansion ratio.

31 No. 13 ("search-tube" nozzle) was made to correspond as nearly as possible with No. 14, so that the terminal pressure as determined in the former might be applied in reaction tests with the latter.

32 No. 15 and No. 16 were used to determine the effect of these very considerable variations in contour.

33 Other forms, such as shorter nozzles or those designed for uniform acceleration and upon other theories, may and probably do give just as good efficiency as those herein described. It seems doubtful, however, in view of the uniform results obtained with nozzles of such different contour as those covered by these experiments, whether it would be advantageous to use any form especially difficult to manufacture, unless it be for the purpose of controlling the shape of the jet as it strikes the moving blades of the turbine. This is very important, as it has a great effect upon the efficiency of action in the blades.

34 It is to be regretted, as stated in Par. 16, that we were unable to procure a calorimeter of sufficient accuracy for our purpose, but such great care was taken to maintain uniform conditions in the boiler room, and these conditions gave such repeated indications of the dryness of the steam at the nozzle entrance, that the probable error introduced is not serious.

35 As stated in Par. 11, the steam pressure was 155 lb. gage and the superheat about 50 deg. fahr. at the boiler. Steam was throttle to the required initial pressure just before entering the flexible pipe, with the result that the thermometer inserted at the nozzle entrance showed about 4 deg. superheat with 700 lb. flow per hr. and sometimes a trace of superheat with 500 lb. per hr. It is probably fair to assume

om this that the steam was dry when used with 145 lb. pressure at ie entrance to the nozzle, and that (in view of the greater throttling hich tends to offset the increased unit radiation from the pipes) iere was always less than 3 per cent of moisture present even with ressures as low as 100 lb. abs.

36 It may be stated in conclusion that a proper method of deter- iining the net effect of under and over-expansion in the nozzle would e as follows:

First: Make a set of nozzles of the same cone angle and finish with throats identical, and with muzzles of different areas.

Second: Determine accurately the proper terminal pressure and the true efficiency of each nozzle, by the method herein described, using a reaction apparatus in which static and moving friction has been eliminated.

Third: Find the push upon a set of turbine blades, using each nozzle discharging into its own proper terminal pressure and into the pressures which are proper for each of the other nozzles of the set.

Fourth: A comparison of the push exerted under these conditions, bearing in mind the "true efficiency" of each jet within the nozzles, will show the net effect of under and over-expansion.

No. 1257

AN ELECTRIC GAS METER

By Prof. Carl C. Thomas, Madison, Wis.
Member of the Society

The meter described in this paper is designed for measuring the rate of flow of gas, air or steam. The operation of the meter depends upon the principle of adding electrically a known quantity of heat to the gas and determining the rate of flow by the rise in temperature of the gas between inlet and outlet of the meter. This principle lends itself to the operation of a meter possessing the following characteristics:

a There are no moving parts inside the meter or in contact with the gas.

b The accuracy of the meter and its sensitiveness are independent of the rate of flow of gas, and of fluctuations in pressure and temperature.

c The meter may be used to measure gas at high pressure as well as at low pressure, and is independent of small fluctuations in pressure, such as those in the discharge from an air compressor or in the suction of a gas engine.

d The meter produces a continuous autographic record showing the rate of flow and its variation.

e Meters of comparatively very small size have very large capacity.

f The meter may be opened for inspection, for blowing out accumulated matter with an air blast, or for washing with gasolene, and it can be dismantled to any extent desired without interfering with the operation of the plant.

2 Fig. 1 shows the meter as constructed for gas or air measurement, and Fig. 2 shows the exterior of the meter, of which Fig. 1 is a section. The meter consists of two parts, first, the measuring element *A* (Figs. 1, 3 and 4), through which all the gas passes when the

Presented at the Annual Meeting. New York, (December 1909), of The American Society of Mechanical Engineers.

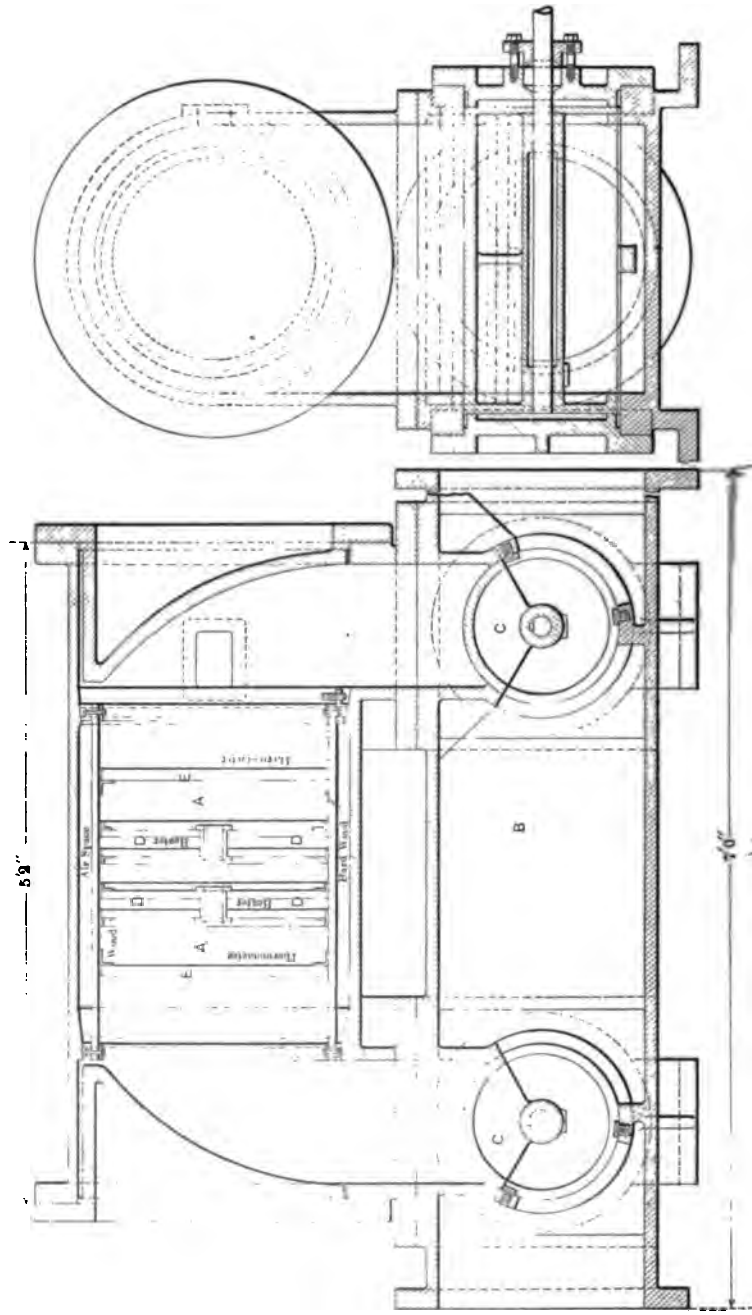
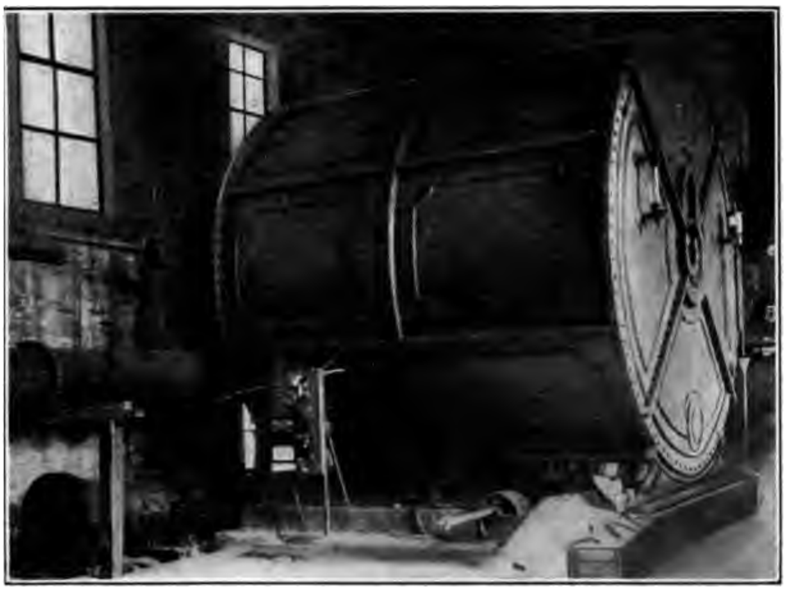


Fig. 1 Longitudinal and Cross Sections of Electric Gas Meter, Showing Construction for Measuring Air or Gas

ıeter is in operation; and second, a by-pass, B (Fig. 1), so arranged hat the meter can be readily cut off from the gas main by operation f the valves C, when it is desired either to operate without the meter or the purpose of inspecting or cleaning out, or to cut the meter out ltogether for any reason. In certain classes of gas work, rolling alves, such as are shown at C, have been found to give trouble, while in other classes of work they are satisfactory. The gate valves ustomarily used in gas work can be substituted for rolling valves s occasion requires, and the by-pass can be made up of ordinary ipe and fittings instead of being a part of the meter.



G. 2 View Showing the Comparative Size of the Electric Gas Meter (at the Lower Left-Hand Corner) and the Ordinary Wet Gas Meter of the Same Capacity

3 The meter consists of an electric heater D (Fig. 1 and Fig. 4), formed of suitable resistance material disposed across the gas passage in such a way as to impart heat uniformly and at a regular rate to the gas passing through the meter. The temperature of the gas is thus raised from that at entrance to some higher exit temperature, and the rise of temperature is measured and autographically recorded by means of the two electrical resistance thermometers E (Fig. 1 and Fig. 4), on the two sides of the heater.

4 These thermometers consist of wire wound upon vertical tubes so disposed as to come in contact with all the gas passing through the meter, thereby indicating the average temperature over the cross section of the gas passage. The fifteen tubes shown at the right of Fig. 1, and also shown in Fig. 3 and Fig. 4, extending in a vertical direction over the cross-section of the meter, support the resistance wire of the thermometers so as to afford a rugged construction. These

Fig. 3 Heater Unit and One of the Resistance Thermometers

thermometers are connected to a recorder (Fig. 2 and Fig. 5), which draws a line on a chart and thus indicates the difference of temperature between the two thermometers.

5 A typical diagram is shown in Fig. 6. This diagram represents a gas flow of from 90,000 to 85,000 cu. ft. per hr., taken during a portion of the day when the fluctuation in flow is small, but nevertheless

continuous. Every small fluctuation in quantity of flow is recorded on the diagram.

6 The diagram in Fig. 7 was made during a period in which the flow varied extensively, the smallest amount recorded being about 17,000 cu. ft. per hr., increasing to 45,000, then to 62,000, to 75,000, the record ending at a flow of about 32,000 cu. ft. per hr.

7 The record in Fig. 6 was made with a temperature difference of about 4 deg. fahr. between the two thermometers, and an energy input of approximately 2 kw. The energy input when the record

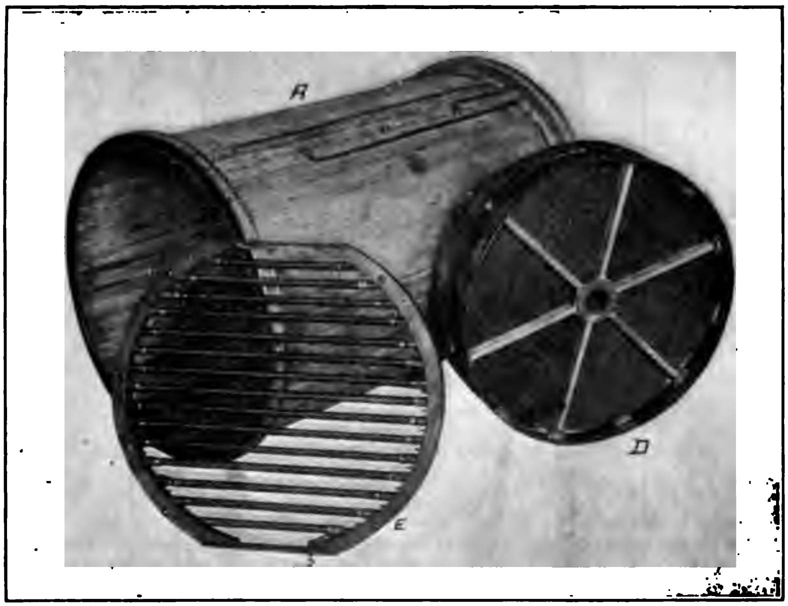


Fig. 4 Showing Construction of Heater and Thermometers

in Fig. 7 was made was approximately 1.15 kw. Fig. 6 is a typical record for a meter of normal capacity of 100,000 cu. ft. per hr., with an electric input of 2 kw.

8 The principle underlying the measurement of gas by this means is as follows: If gas is flowing through the heater at a given uniform and constant rate, and if heat is being supplied electrically, and imparted to the gas at a constant rate, a certain definite rise of temperature will be produced in the gas during its passage between the two thermometers and through the heater, and this constant difference

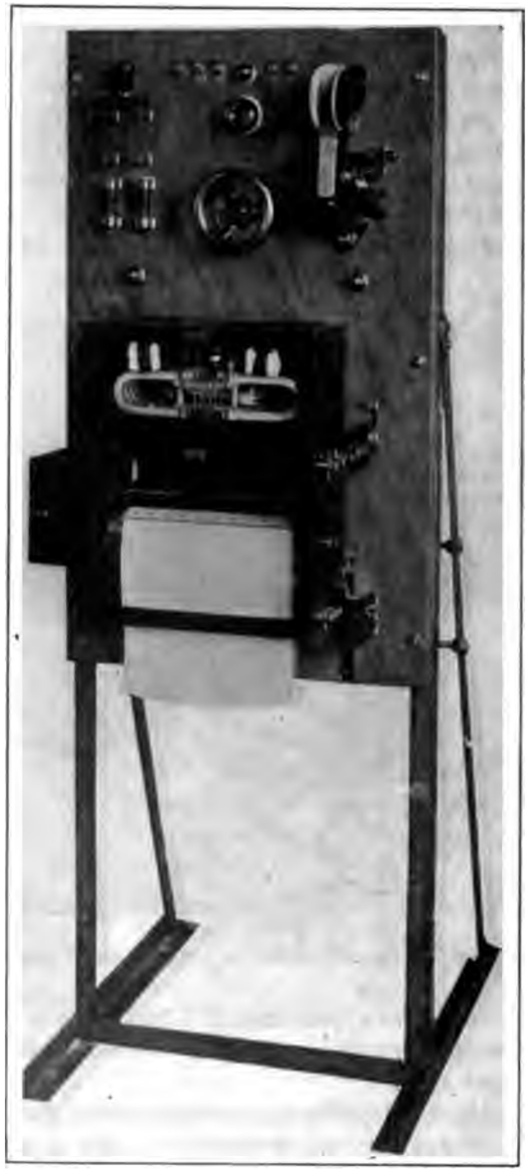


Fig. 5 Recording and Operating Instrument

of temperature will be maintained so long as the amount of gas passing per unit of time is constant. But if the quantity of gas passing per unit of time diminishes, the heat supplied at the same constant rate as before will raise the temperature of the gas by a greater amount than was the case when a larger quantity of gas was flowing and absorbing the energy liberated by the heater. Conversely, if the rate of flow increases, the energy being supplied to the heater and delivered to the gas will not be able to raise the temperature by as great an amount as when the rate of flow was less. The temperature difference produced by a known input of electrical energy thus forms a measure of the quantity of gas flowing through the meter.

9 The meter may be operated in either one of two ways, of which the first is as follows: the difference of temperature between inlet and

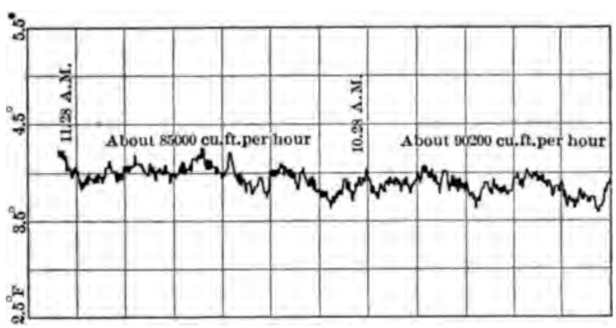


FIG. 6 AUTOGRAPH RECORD SHOWING GAS FLOW OF ABOUT 87,000 CU. FT. PER HR.

NOTE: THIS DIAGRAM WAS TAKEN UNDER APPROXIMATELY STEADY CONDITIONS OF FLOW DURING THE REGULAR OPERATION OF ONE OF THE PLANTS OF THE MILWAUKEE GAS LIGHT COMPANY. THE PAPER IN THIS CASE WAS TRAVELING AT A RATE OF 3 IN. PER HR. THE RECORDER CAN BE SET FOR ANY ONE OF THREE SPEEDS OF PAPER, 3 IN., 6 IN., OR 12 IN. PER HR. THE HIGHER SPEEDS ARE DESIRABLE AS THEY SMOOTH OUT THE CURVE OF TEMPERATURE DIFFERENCES. THE SCALE OF TEMPERATURE DIFFERENCES CAN ALSO BE GREATLY ENLARGED IF DESIRED.

outlet is kept constant, and the watts required to maintain this constant difference of temperature vary directly as the weight of flow. The watts input thus forms the measure of the weight of flow of air or gas, the watts being measured by a recording wattmeter, or in some cases by an integrating wattmeter. The fixed difference of temperature (about 5 deg. fahr.) is maintained by the action of a device made upon the same principle as the well-known autographic temperature recorders used in connection with resistance thermometers, but without the autographic part.

10 The mechanism which actuates the pen carriage in the autographic recorder is so arranged that when the carriage tends to depart from the straight-line path indicating a constant difference of temperature it automatically cuts in and out the resistance necessary in order to maintain the fixed difference of temperature. This variation of energy input is accomplished by a small motor-controlled rheostat mounted on the switchboard. Thus as the rate of flow of gas is increased, the temperature difference tends to decrease, and at once additional energy is introduced sufficient to heat the increased weight of gas so as to maintain the constant temperature difference. This method of operation is advantageous because it does not require the maintenance of a constant voltage on the line supplying the energy for heating the gas. The accuracy is thus independent of the small fluctuations in voltage generally found on electric supply circuits.

11 The second method of operation involves the use of the autographic temperature recorder, including the graphical part, the diagram from which, representing the variation of difference of temperature with constant energy input, gives the measure of the quantity of gas passing the meter. That is, the electrical resistance of the meter remains constant, and the meter is supplied with current at constant voltage, which results in constant energy dissipation in the meter. The difference of temperature between inlet and outlet then rises and falls according to the decrease or increase, respectively, of the rate of flow of gas.

12 The first method of operation mentioned is superior to this second method, inasmuch as the first is independent of any change which might take place in the electrical resistance of the material composing the heater. Operation by the second method requires that constant voltage be maintained across the line, and that the electrical resistance of the heater shall remain constant, or else that both watts input and temperature difference shall be recorded. In the experimental work of developing the meters it has been found convenient to use this second and more cumbrous method, but in meters at present under construction the first-mentioned method has been adopted, thus avoiding the necessity for either constant voltage or constant resistance, and resulting in simpler apparatus throughout. A record of the watts input is, by the method now used, all that is required for determining the flow of gas through the meter. The meters can be arranged to operate with either direct or alternating current, and the controlling device can be arranged to work with any desired voltage.

13 Fig. 2 shows, at the lower left-hand corner, an electric gas meter together with its autographic recorder and switchboard control. This electric meter is used for measuring all of the gas which was formerly passed through the large wet meter shown in the figure, and is of sufficient capacity to enable it to measure about three times the amount of gas for which the wet meter is suited. The electric meter was placed in this position between a 100,000-cu.ft. gas holder and the large station wet meter, for the purpose of calibrating the electric meter and comparing the results, based upon the rate of drop of the gas holder, with the readings of the wet meter. The curve obtained from the autographic recorder was thus interpreted by means of the calibration carried on in connection with the gas holder, the wet meter and a meter prover of the largest size made. It was found that the wet meter used in this case was exceedingly accurate. It had been carefully put in order and calibrated before these tests, and when operated at loads within its capacity, the readings were entirely reliable. The best evidence of this is given by the results used in plotting Fig. 8.

14 The specific heat of a given kind of gas appears to be very nearly constant, since those constituents which vary from time to time are not those which appreciably affect the value of the specific heat. But it is desirable to calibrate the meters with a gas having the same specific heat as the gas which it is intended to measure in a particular case. The specific heat of illuminating gas is very closely 0.020 per cu. ft. at mean atmospheric pressure and 60 deg. fahr. temperature, as shown in Fig. 8 and also by the following calculation based upon a fairly typical analysis. Such variation as commonly occurs in the relative amounts of the various constituents does not greatly affect the specific heat. The following calculation is for a gas at 760 mm. and 0 deg. cent.

	Vol. cu. ft.	Weight per cu. ft., lb.	Total Weight lb.	Specific Heat per lb.	Specific Heat per cu. ft.
CO_2	0.04	0.11637	0.004658	0.216	0.00100
C_2H_4	0.11	0.0741	0.00815	0.404	0.00329
O_2	0.001	0.08463	0.00085	0.217	0.00023
CO	0.331	0.07407	0.02450	0.245	0.00600
CH_4	0.1761	0.04234	0.00746	0.593	0.00442
H_2	0.303	0.00530	0.00160	3.409	0.00546
N_2	0.0389	0.07429	0.00289	0.244	0.00071
					0.02111

15 The specific heat of blast-furnace gas is practically the same as that of atmospheric air, and the same is true in a general way regarding producer gas. Thus, taking the following as an average analysis of blast-furnace gas, the specific heat is found to be 0.0192, while atmospheric air has a specific heat almost identical with this, or approximately 0.0191 per cu. ft. This is to be expected, since producer gas and blast-furnace gas consist principally of nitrogen and carbon monoxide.

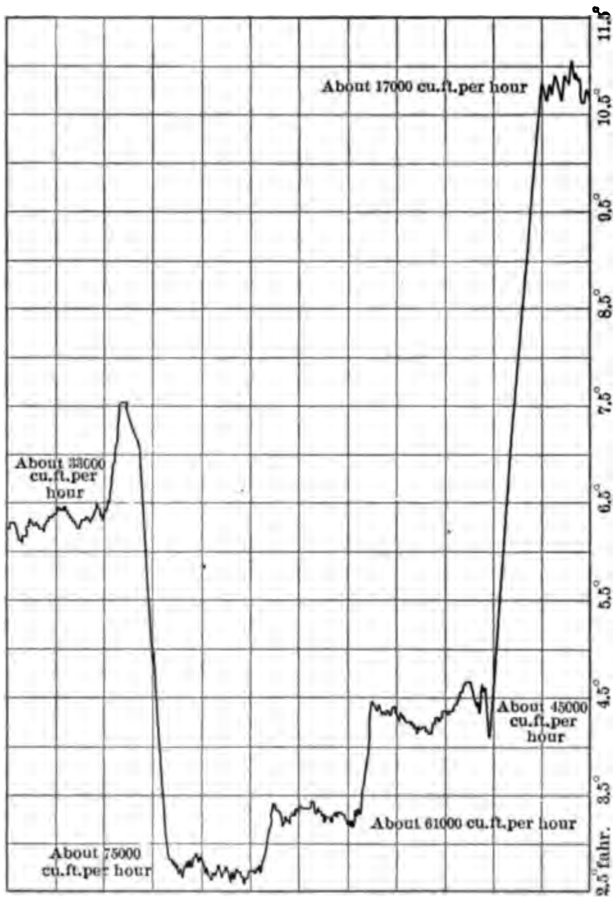
	Vol. cu. ft.	Weight per cu. ft., lb.	Total Weight lb.	Specific Heat per lb.	Specific Heat per cu. ft.
N_2	0.60	0.0743	0.0446	0.244	0.0109
CO	0.24	0.0741	0.0178	0.245	0.0044
CO_2	0.12	0.1164	0.0140	0.216	0.0030
H_2	0.02	0.0053	0.0001	3.409	0.0003
C_2H_4	0.02	0.0741	0.0015	0.404	0.0006
					0.0192

16 The meters have been calibrated with illuminating gas and with air. A certain amount of water vapor is carried with the gas or air passing the meter. This vapor forms part of the gas or air, and is heated just as are the other constituents. The rise of temperature caused by the heat added in the meter is only a few degrees, and consequently the water vapor does not experience a change of state. The temperature of the metal forming the electric heater rises only 15 or 20 deg. fahr. above the temperature of the gas. The question of latent heat of vaporization of the water vapor therefore does not enter into the considerations underlying measurement of the gas.

17 While calibration of the meters under actual conditions of service is depended upon to obtain quantitative results, yet these meters are of such a nature that the quantity of gas or air passing through them can be very closely calculated from a knowledge of the energy input and the specific heat of gas or air. This fact, that the quantity of flow can be quite closely calculated, independently of a calibration curve, makes it possible to check the accuracy of the readings obtained.

18 The development of this meter is a result of experiments which the writer has been making for some years to determine the specific heat of gases by heating them electrically. The performance of a properly constructed heater for this purpose proved to be so entirely regular that it was apparent that the quantity of gas flowing through it could be very accurately measured by the method now used in these meters. The problem is thus the reverse of the problem of

termining specific heat by measurement of the electrical energy cessary to heat the gas. It will be seen by reference to Fig. 1 that e whole process of heating the gas and of measuring the difference temperature between inlet and outlet, is accomplished in a relavely small space which is well insulated so far as heat losses are con-



3. 7 AUTOGRAPH RECORD SHOWING WIDE FLUCTUATIONS IN FLOW OF GAS

rned, since the heater and thermometers are contained in a casing ade of hardwood strips and separated from the metallic walls of e meter by an air space.

19 A typical calibration curve is shown in Fig. 8. The curve ows the degrees rise in temperature per kilowatt introduced when

any given rate of flow through the meter is taking place. It will be seen that this curve is asymptotic to the coördinate axes, because, when an indefinitely great amount of gas is being heated, any finite input of heat will produce only an indefinitely small rise of temperature; and on the other hand, when the amount of gas becomes indefinitely small, a finite input of heat will cause an indefinitely great rise of temperature. The calibration curves obtained are therefore rect-

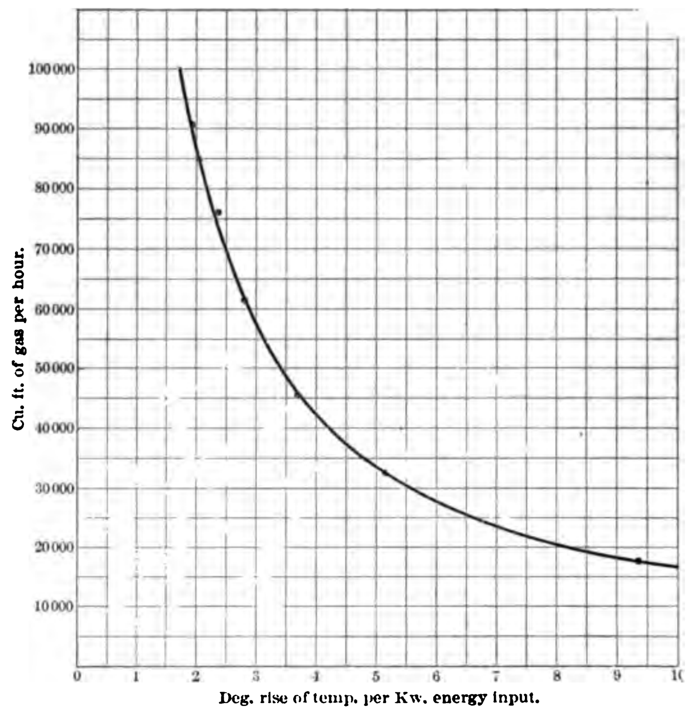


Fig. 8 Calibration Curve, See Appendix for Data

angular hyperbolas. The product of weight of gas multiplied by degrees temperature rise per watt introduced is a constant, and this constant, for a given kind of gas, takes the place of a calibration curve and renders it unnecessary to refer to a curve. The constant as shown by Fig. 8 is 170,000, showing a specific heat per cu.ft. of $\frac{3.412}{170}$ = 0.0201.

20 The accuracy of these meters is not affected by changes in

pressure of the gas or air, since the unit of measurement is that of weight rather than of volume; that is, the meter takes cognizance of the specific gravity, or the amount of "stuff" in a given volume of the gas. Also variation of temperature of the incoming gas does not affect the accuracy, because it is a difference of temperature, rather than a fixed temperature, upon which the measurement depends. The meter can be used for gas or air at either high or low pressure, and at either high or low temperature, provided the materials used in construction are suited to the conditions.

21 This method of measuring gas seems especially useful in connection with engines operated by gas from producers, blast furnaces, etc., and in measuring the discharge of gas or air from compressors, because the small and rapid periodic fluctuations of pressure, due to the suction of gas engines or to the discharge from compressors, do not interfere with the steady action of the thermometers. The time lag of the latter is sufficient to smooth out the curve of temperature variation, or of watts input, as the case may be, and true average results are thus indicated.

22 The temperature difference employed when operating with a constant difference, is approximately 5 deg. fahr. When a curve of temperature difference is employed, the temperature rise is from 4 to 5 deg. fahr. when the normal maximum amount of gas is flowing. This difference may be increased to 10 or 12 deg. when the rate of flow is greatly diminished, and at 100 per cent overload the temperature difference is from 2 to 2½ deg. On the autographic record one inch represents a temperature difference of one degree. The thermometers and recording device are such as to render the records accurate within 1 per cent. The minute fluctuations shown by the curves on Fig. 6 and Fig. 7 are produced by the constantly varying rate of flow in the gas mains. These can be "damped out" to any extent desired. The apparatus with which this record was taken was purposely made sensitive to minute fluctuations.

23 The electrical energy required to operate the meters is approximately 1 kw. per 50,000-cu. ft. hourly capacity. The curves shown in Fig. 7 represent variations of from 17,000 to 75,000 cu.ft. per hr., and were made with an energy input of approximately 1.15 kw. To provide for more gas and still have the record lie conveniently on the paper, it is only necessary to increase the energy input by manipulation of the rheostat hand-wheel on the switchboard.

24 The meters are so constructed that the heads can be easily removed and an air blast used for cleaning out the interior, or the

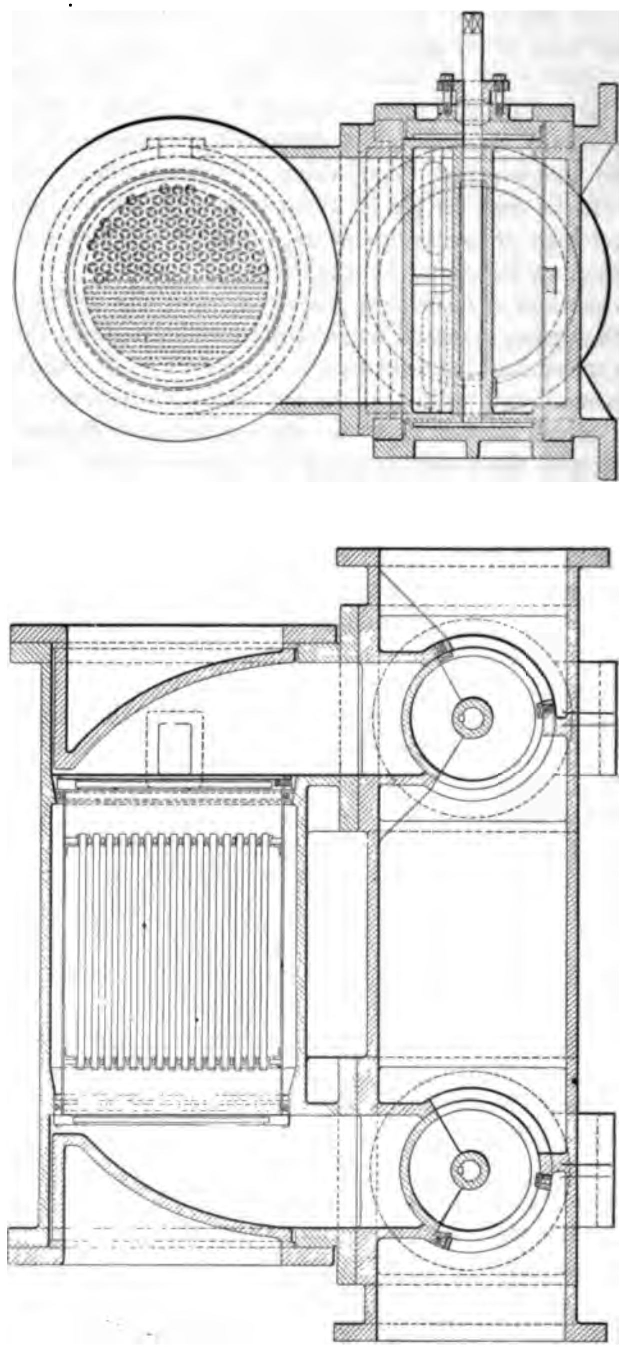


FIG. 9 METER AS ARRANGED FOR MEASURING EITHER QUANTITY OR QUALITY OF STEAM

entire casing, containing heater and thermometers, can be removed and dipped in gasolene for the purpose of removing tar or other deposit. All parts of the meter are of rugged construction, and are of well-developed materials familiar to engineers. The heater units consist of corrugated strips of resistance ribbon about 1½ in. wide, wound spirally into discs of such diameter as to fit the inside of the wooden casing. The number of these discs depends upon the capacity of the meter. The heater shown in Fig. 4 consists of two discs.

25 The same type of meter, modified as shown in Fig. 9, can be used for the measurement of steam, and also for determining the quality or percentage of moisture of steam. When used for measuring the quantity of steam, the steam is first superheated slightly in a superheater of the ordinary type, after it leaves the boilers and before passing through the meter.

26 The heater element in the steam meter consists of tubes, as shown in Fig. 9, made of suitable resistance material and supported on insulating bushings in the tube plates, the construction being similar to that of a surface condenser. The slightly superheated steam is passed through and around these tubes, and is further heated by the electrical energy supplied to the tubes.

27 The difference of temperature produced by a given energy input forms a measure of the weight of steam flowing, just as has been described in the case of the gas meter. In cases where it is desired to make engine or turbine tests with unsuperheated steam, the steam can be reduced in temperature after passing the meter, by the injection of a spray of water. Of course the measurement of superheated steam is simpler than is the case where superheating is not a feature of the regular operation of the plant.

28 The amount of moisture carried by steam can be very accurately determined with this apparatus, by passing all of the steam through the electrical heating material and noting the amount of energy required to "fry out" the water and cause superheating to commence. The pointer over the dial of the instrument connected with the resistance thermometer in the outlet of the calorimeter indicates when the temperature of the steam begins to rise. It is proable that the only way to determine accurately the quality of wet steam is to pass all of the steam, and not a small sample, through a calorimeter. It is of course not always practicable to do this, and in such cases it is necessary to use smaller calorimeters and to sample the steam.

29 When inserted for either regular or intermittent use as a steam

meter or as a calorimeter, the device can be cut off from the steam line in the manner already described for the gas meter, and as shown in Fig. 1.

30 The automatic recording device for the gas meter is so arranged that in case the flow of gas should be interrupted for any reason the current is automatically cut off at the switchboard. Also if the flow of gas becomes so small in amount that the pen reaches within a half inch of the edge of the paper, the current is cut out. When the gas has cooled the heater slightly, the current is automatically cut in again, and if the gas flow is increased the pen goes back toward the middle of the diagram and operation proceeds normally. If the gas flow continues but does not increase beyond that at which the current was cut out, the pen will "hunt" back and forth near the edge of the paper. It can be brought back toward the middle of the paper by the introduction of less energy to the meter. The gas meter is thus fully protected from possible injury due to the complete shutting off of gas supply.

31 At the other edge of the paper, representing the maximum flow of gas, the operation is similar to that already described. In order to bring the recording pen upon the range again the electrical input is increased by manipulation of the hand-wheel on the switchboard. This applies to operation by the second method described in Par. 11, in which the temperature difference between the two thermometers forms the record of gas flow. When the first method is employed, that of maintaining constant temperature difference, the meter is also automatically protected by the motor-controlled rheostat, and the range of the instrument is unlimited and it does not require manipulation by hand. It will be seen by reference to Fig. 7 that the range of the instrument when operated by the second method of varying temperature difference, is very wide, and takes care of extensive fluctuations of gas flow.

THEORY OF THE METER AND METHOD OF OBTAINING STANDARD RESULTS

32 The figures given in paragraphs 14, 15 and 19 can be reduced to standard conditions of temperature and pressure, and the meter readings can be autographically recorded directly in "standard cubic feet" of gas or air. Let

G = cubic feet of gas per hour

E = energy in kilowatts

Then B.t.u. per hr. $= 3412\, E$

T = temperature difference, deg. fahr.

S = specific heat per cu. ft.

Then GST = heat energy equivalent to E, or $GST = 3412\, E$. $\frac{GT}{E}$ $= \frac{3412}{S}$ = a constant K which depends upon the specific heat of the gas.

33 Since the temperature difference T is kept constant, it follows that $\frac{K}{T}$ is constant. Let $\frac{K}{T} = C$. Then $G = \frac{KE}{T} = CE$.

34 It is now proposed to show by reference to the gas and the air curves in Fig. 10, that if the specific heat of gas made under given conditions be calculated from the customary chemical analysis and the specific heat of the constituents, then this specific heat may be used for determining the constant C. From the gas curve (Fig. 10), which was made with illuminating gas at an average temperature of 59 deg. fahr., and under an average absolute pressure of 6 in. water and 29.8 in. mercury,

$$K = 170{,}000 = \frac{3412}{S}$$

35 Therefore for the condition of the gas when the tests were made the specific heat per cubic foot $S = \frac{3412}{170{,}000} = 0.0201$. If this be reduced to standard conditions of 32 deg. fahr. and 29.9 in. mercury, then $S = 0.021$, which is to be compared with the calculated specific heat (Par. 14), giving $S = 0.0211$. If the standard conditions are taken as 62 deg. fahr. and 29.9 in. mercury, the specific heat becomes 0.0198, and the constant becomes

$$K = \frac{3412}{0.0198} = 172{,}500, \text{ nearly}$$

If the temperature difference is kept constant at 5 deg. fahr., then $\frac{K}{T} = \frac{172{,}500}{5} = 3450 = C$, or $G = 3450\, E$.

36 The cross-section paper on the recording wattmeter is ruled so that 3450 E is read directly, instead of the watts E. The record is thus read directly in cubic feet of gas. The regular records of chemical analysis of the gas should be referred to from time to time in order

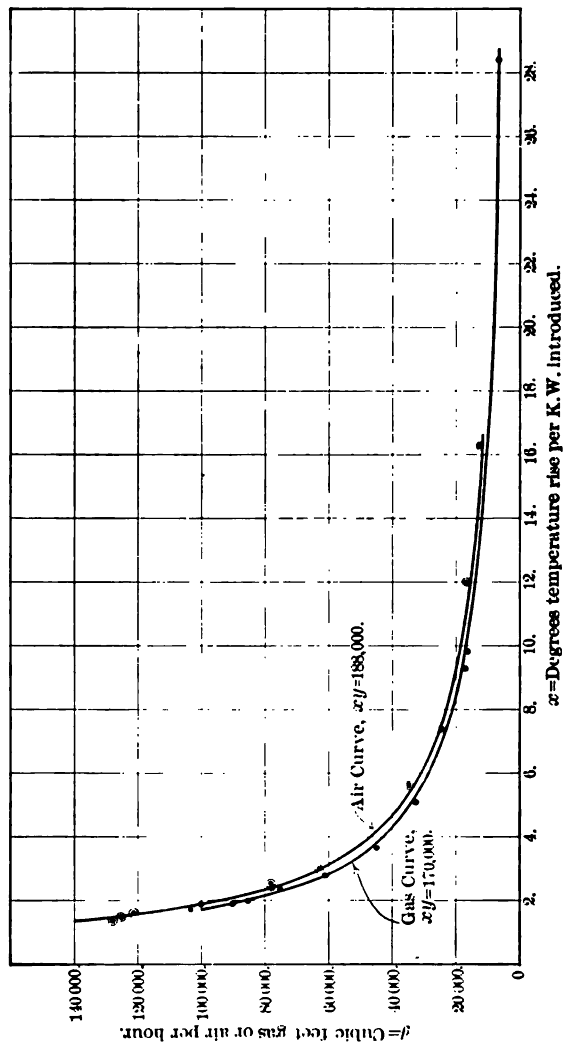


FIG. 10 SHOWING DEGREES TEMPERATURE RISE PER KW. FOR DIFFERENT RATES OF FLOW OF GAS AND AIR

to ascertain what percentage variation takes place in specific heat. It appears, as stated previously, that the elements which vary during the operation of a gas plant are not those whose variation would produce serious variation in specific heat. The variation that does take place is apparently well within the limits of accuracy practicable, or generally considered necessary in the operation of gas plants. By taking frequent chemical analyses the error can be reduced so as to be quite negligible.

37 The conditions during the air tests were as follows: barometer, 29.75; pressure, 6.5 in. water; average temperature of air as measured in the wet meter, 60 deg. fahr. From the air curve obtained under these conditions (Fig. 10)

$$K = 188{,}000, \text{ and } S = \frac{3412}{188{,}000} = 0.0181$$

38 Reducing this to standard conditions of 32 deg. and 29.9 in. mercury, $S = 0.0191$. This is to be compared with the accepted specific heat of air under these conditions, or 0.0192 B.t.u. per cu. ft. This provides perhaps the best evidence that could be obtained, as to the accuracy of these tests, since the specific heat of air is well known at the conditions under which the tests were made. A more commonly familiar figure for specific heat of air is obtained by multiplying 0.0192 by the number of cubic feet of air per pound under the above conditions, or 12.38. The result is 0.2377 B.t.u. per lb. per deg. and this is to be compared with 0.0191×12.38 as given by the meter, or 0.2365.

39 The constant K for air at 32 deg. and 29.9 in. is therefore

$$\frac{3412}{0.0191} = 178{,}630$$

and reducing this to 62 deg. instead of 32 deg.

$$K = \left(1 \times \frac{30}{493}\right) \times 178{,}630 = 189{,}500 \text{ nearly}$$

If $T = 5$ deg., $\frac{K}{T} = 3790$.

40 The error involved in calling this constant 3800 is less than ½ of 1 per cent and well within the limits of accuracy possible under the circumstances. The standard cubic feet of air passing the meter are therefore $G = 3800\ E$, and the autographic records are arranged to read accordingly, in standard cubic feet of air per hour.

41 The development of a new device requires consideration of a large number of questions arising out of the conditions of service proposed. The question of specific heat has been considered in the preceding paragraphs. The degree of success which has been attained with this meter in accurately measuring specific heat is due principally to an extensive experience in this particular class of work, which has served to point out the way to make an electrical heater in which heat losses are negligibly small. The arrangement of the meter is such that the heat given off can go into the gas only, and it necessarily all goes into the gas, with the exception of a negligibly small loss which it is not worth while to minimize further. That the gas receives all the heat, excepting this negligibly small loss, is true whether or not the heating material has collected deposit of some kind. So long as the gas can get through the heater, its temperature is raised proportionately to the heat supplied.

42 The question of the presence of a small amount of water vapor, as part of the gas, has so far not introduced any complications. It is conceivable that if the gas carried a large percentage of water the operation of the meter would be interfered with,—but so would the operation of a gas engine or a burner. The meter can apparently measure accurately any gas that can be used by a gas engine. The absence of moving parts in the meter gives it an advantage over the engine, and dust can be to a considerable extent deposited before entrance of the gas to the meter. The heating element and thermometers can be cleaned by dipping in gasolene, without damaging them.

43 Meters at present under construction are being made with the axis of the cylinder vertical, with a view to greater convenience of access and in making connections.

44 The first large meter of this type to be installed was put in the works of the Milwaukee Gas Light Company, and the writer is indebted to the officials of that company for their coöperation in making extensive tests during the work of development.

45 Referring to Par. 16, for gas or air under the conditions existing during the tests, of approximately 60 deg. fahr., 29.8 in. mercury and 6 in. water pressure, the correction for water vapor introduces a change in the results of less than one-half of one per cent, and was therefore omitted. At other pressures and temperatures the correction for water vapor can be easily made by reference to the charts commonly used in gas works. An interesting confirmation of the statement in Par. 16 appeared during the tests, in that the most

minute addition of electrical energy caused an immediate rise of temperature of the gas or air. This was repeatedly tried with great care, and always with the same result.

APPENDIX

DATA RELATING TO CALIBRATION CURVE, FIG. 8

Time	Wet Meter Reading	Cu. Ft. Gas Per Hr.	Average Temperature Difference Deg. Fahr.	Average Kilowatts Input	Deg. Temp. Rise Per 1000 Watts
A.M.					
10-05	90148.0				
10-10	——				
10-15	90177.5				
10-20	90192.0	17350	10.7	1.153	9.30
10-25	90206.0				
10-30	90220.5				
10-50	90396.0				
10-55	——				
11-00	90470.0				
11-05	90509.0	45200	4.25	1.150	3.69
11-10	90546.0				
11-20	90644.0				
11-25	90695.0	61200	3.25	1.160	2.80
11-30	90746.0				
11-40	90884.0				
11-45	90947.0	75600	2.70	1.160	2.32
11-50	91010.0				
P.M.					
12-05	91098.0				
12-10	91125.5				
12-15	91152.7	32640	6.25	1.22	5.12
12-20	91180.0				
12-25	91206.8				
A.M.					
9-30	91418.2				
9-35	91493.4	90240	3.90	2.05	1.90
9-40	91568.8				
9-45	91643.8				
10-00	91857.1				
10-15	92074.0				
10-30	92291.7				
10-45	92506.2	84960	4.10	2.05	2.00
11-00	92717.5				
11-15	92925.4				
11-30	93131.0				

DISCUSSION

PROF. L. S. MARKS. The meter described by Professor Thomas should prove a valuable addition to the instruments used in gas engine and other testing. The possibilities of error in the indications of such an instrument must be fully examined.

2 This meter is fundamentally an instrument for determining the weight of gas or vapor flowing through it and is made to record volumes. It is obvious that these volumes cannot be those actually flowing but must be the volumes reduced to some standard conditions of temperature and pressure. The author has not mentioned this matter in his paper, but it is of considerable importance. A variation of 5 deg. fahr. in temperature, or of 0.3 lb. in pressure, under ordinary atmospheric conditions, would result in an error of 1 per cent in the indications of the instrument if it were assumed to record actual volumes flowing. The calibration of the instrument by passing through it a known volume of a gas at known pressure and temperature, can easily be reduced to a calibration under standard pressure and temperature conditions.

3 In Par. 16 the author refers to the effect of water vapor carried in with the gas. He states that, in consequence of the small rise of temperature, the water vapor does not experience a change of state, and that, consequently, the latent heat of vaporization does not enter into consideration. It is obvious that he is considering here the case of a gas which not only is saturated with water vapor, but also is bringing with it minute particles of water in suspension. Under these conditions—and they are conditions which may easily obtain with blast-furnace gas which has just passed through the washers—the indications of the instrument will be rendered completely useless.

4 If the gas should enter at a temperature of 70 deg. fahr. it would contain 0.001148 lb. of water vapor. After passing through the meter with a rise of temperature of 5 deg. the same weight of gas could contain 0.001198 lb. of vapor; that is, there would occur a vaporization of 0.00005 lb. of moisture for every cubic foot of gas passing through the meter. The latent heat of vaporization at these temperatures is about 1050 B.t.u., or 0.0525 B.t.u. will be used in converting the water into vapor. As the total heat required for raising one cubic foot of the gas 5 deg. fahr. is only about 0.1 B.t.u., we have here, obviously, the possibility of an error of the magnitude of 20 or 25 per cent in the indications of the instrument in the

case suggested by the author where the gas is supersaturated with vapor.

5 The accuracy of the instrument depends primarily on the accuracy with which the volumetric specific heat of the gas can be determined, and upon the constancy of this quantity while the meter is in operation. For the correct determination of the volumetric specific heat it is necessary to know the volumetric composition of the gas and the volumetric specific heat of each of the constituents. The author has stated that the volumetric specific heat of each kind of gas is very nearly constant and the calibration of the instrument is based upon that assumption; that is, it is proposed to calibrate the instrument with, for example, producer gas, and then to use that calibration when the instrument is used at other times with producer gas. It will be interesting to examine how nearly correct this assumption is. In the four analyses of producer gas, three of them by Mr. Bibbins, and one by Messrs. Garland and Kratz, I have worked out the volumetric specific heats of these gases, using the physical constants given by the author, and I have also taken at random two analyses from tests which I have made on a large anthracite gas producer. The results of the calculations are as follows:

6 For the two lignites in Mr. Bibbins' paper, the values of the specific heats are 0.01920 and 0.01899, which agree very closely with the average stated by the author. For the bituminous coal in Mr. Bibbins' paper, the value is 0.01899, and for the bituminous coal in the paper of Messrs. Garland and Kratz, the value is 0.0186. My own tests with anthracite give values 0.01826 and 0.01848, respectively.

7 It is quite evident from these figures that there is considerable variation, which may be as great as 5 per cent in the volumetric specific heat of producer gas. It may possibly be, as these figures seem to indicate, that the specific heat can be stated with greater accuracy if the type of coal is also specified, since there seems to be a relation between the volatile contents of the coal and the specific heat of the producer gas; but this point has not been sufficiently investigated to permit of any definite conclusions.

8 I have attempted also to see whether the value given for illuminating gas is constant. Only one illuminating gas was considered—that in Cambridge, Mass.—the analysis having been made by the chemist of the gas company. The specific heat calculated from this analysis is 0.02278. The specific heat calculated by the author is

0.02111. The value which he states as being practically constant for illuminating gas is 0.020. There is a variation of over 10 per cent between these values, so it would seem that it is not practicable to calibrate this instrument with illuminating gas at one place and assume it to be accurate when used with illuminating gas at some other place.

9 Moreover it must be recognized that such an analysis as that given by the author for illuminating gas is only approximate; the heavy hydrocarbons are never fully analyzed and some kind of guess must be made as to their composition and specific heats. It cannot even be accepted as true that a calibration made with any particular illuminating gas will hold at some later date for gas from the same source. I have found variation in the composition of the Cambridge gas which would certainly cause a variation of two or three per cent in its specific heat.

10 It appears to me then, that this instrument cannot be accepted for accurate measurement unless analyses are being made of the gas that is going through the meter. In scientific testing, such analyses will naturally be undertaken and consequently the instrument should be extremely valuable in such cases. I would like to know what experience the author has had with this instrument in the measurement of volumes when the flow is variable as, for instance, when gas is flowing through a single-acting, four-cycle gas engine. In this case the flow will occur approximately for only one-fourth of the whole time of the test. The author's contention that the indication of the instrument would be accurate under these circumstances seems reasonable, but it would be valuable to know whether, and to what extent, his statement has been verified by actual investigation.

Prof. W. D. Ennis. I do not quite follow Professor Thomas' explanation that the proper correction has been made for fluctuations in the pressure of the gas. A change of, say, five per cent in the pressure, measured above the zero of pressure, would correspond roughly with a change of five per cent in the absolute temperature, without any addition whatever of heat. A change of five per cent in absolute temperature would mean a very large change in Fahrenheit temperature.

2 A more important point is suggested by the statement in Par. 4: "These thermometers consist of wire wound upon vertical tubes so disposed as to come in contact with all the gas passing through the meter, thereby indicating the average temperature over the cross section of

the gas passage." If that is what the thermometers do, I question whether they indicate the average temperature of the gas, because more gas is passing at a point in the middle of the pipe than at points near the circumference. Do the thermometers indicate the average temperature of the whole weight of gas, which is the temperature that we must have in order to calculate the weight of gas flowing?

Edwin D. Dreyfus. Certain fuel gases—particularly blast-furnace, coke-oven and producer gas—carry with them a considerable quantity of finely divided solid matter, which in turn forms deposits in the piping or in any piece of apparatus through which the gas passes.

2 From their construction, it would seem that the grids in the meter would favor the formation of deposits of this sort, and I would like to ask whether Professor Thomas has made any trials to determine what effect, if any, such deposits have on the accuracy and general reliability of the instrument.

3 In cases where the gas is carried long distances through overhead mains—as in many blast-furnace plants—the temperature of the gas will be influenced largely by the temperature of the atmosphere, as between the summer and winter months the gas temperatures might easily vary as much as 50 deg., and the variation in temperature would have a decided effect on the moisture content. It seems probable that the moisture content of the gas is the most disturbing factor affecting the accuracy of the instrument. If this be so, then it is desirable that the actual significance of this factor should be determined by trials made over as wide a range of conditions as we may reasonably expect to meet in ordinary everyday practice.

A. R. Dodge. I would like to ask Professor Thomas if he has made calculations in regard to the amount of power necessary to operate this meter when used as a steam meter. The specific heat of steam being greater than that of gas and air, the amount of power required is considerable. For instance, Thomas meters on the large turbines of the New York Edison Company would require about 545 kw. at normal load, quite a percentage of the total output of the turbine.

The Author. Bearing upon the questions asked in the discussion, I would say, that in addition to the description of the meter given in the paper, I have given in Fig. 10 completed curves showing the results obtained in calibrating the meter with both illuminating gas and air, reduced to standard conditions of 29.9 in. mercury and 62 deg. fahr. These curves show the method of using the meter for measuring directly standard cubic feet of gas or air at some convenient assumed conditions of pressure and temperature. The results of measurement by the method described in the paper may be considered as given either in standard cubic feet, or in weight of gas passing the meter.

2 These meters are essentially applicable to the measurement of a dry gas or steam, that is, a gas or steam which is either saturated or superheated. Our experience with the gas meters has thus far been with illuminating gas and with air, and these are exceedingly easy of measurement. The gas or air we are measuring is saturated, carrying its full quota of water vapor. The smallest quantity of heat introduced causes an immediate rise in temperature of the gas. If the gas carried a spray or mist of water, the measurement would be in error to a certain extent, because of the difference in specific heat between the water vapor and the gas. The extent of the error would depend upon the percentage of water vapor present.

3 For gas or air under the conditions existing during the tests, of approximately 60 deg. fahr. and 29.8 in. mercury, and 6 in. water pressure, the correction for water vapor introduces a change in the results of less than ½ of 1 per cent, and has therefore been omitted. For other pressures and temperatures the correction for water vapor can be easily made by reference to the charts commonly used in gas works.

4 An interesting confirmation of the statement in Par. 16 is that the most minute addition of electrical energy to the gas or air causes an immediate rise of temperature.

5 Regarding variation of specific heat, the meter prover shown in Fig. 1, herewith, has been developed. It consists of a small electric heater which is placed in the outlet of the meter and discharges into a portable gas-holder such as is used for proving large meters. By this means a small known quantity of the gas is heated, and the specific heat actually determined by direct measurement. This determination can be made as often as desired until the variation of specific heat and satisfactory average values have been determined. So far it appears that the specific heat in a given installation is practically constant from day to day and from one time of day to another. The fact that it is possible thus to determine the specific heat experi-

ıentally affords a most valuable check upon the specific heat deterıined by calculation from chemical analysis, since the methods used ı the latter are at best largely approximations.

6 As to dust and impurities collecting on the heater: A meter ow in operation for some months has been used for measuring in the eighborhood of 100,000 cu. ft. per hr. of illuminating gas. The

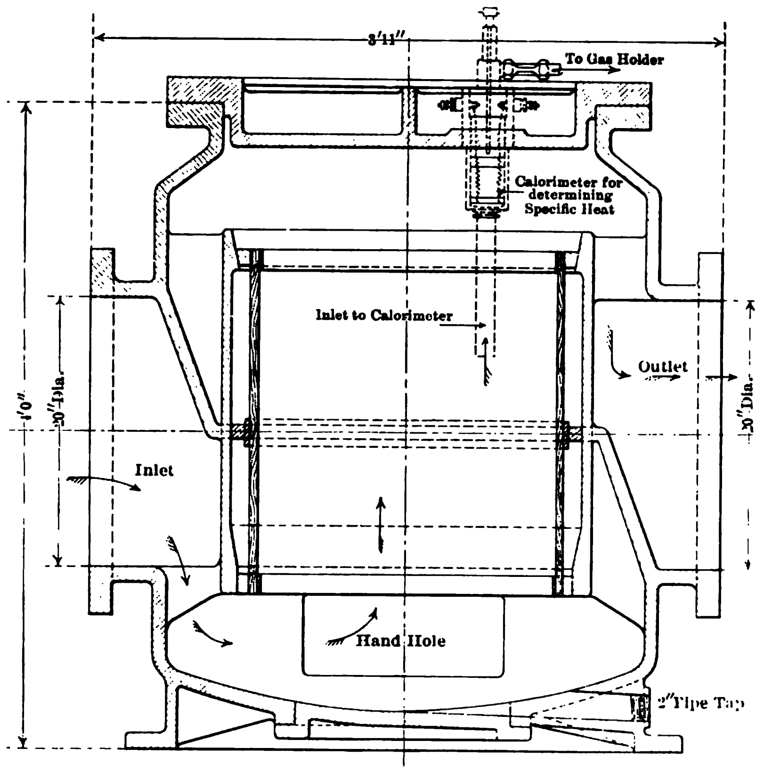


FIG. 1 GAS METER ARRANGED WITH CALORIMETER FOR DETERMINING SPECIFIC HEAT

eater has been taken out once, and in handling it a small amount of rease was found on the heater material. Otherwise the interior f the meter was clean. In handling very impure gas it will of course e necessary to clean out the meter occasionally, simply in order to rovide sufficient area for the passage of the required amount of gas. ll the heat generated in the heater necessarily goes into the gas.

The operation of heating and measuring difference of temperature is all accomplished in a very short length of travel of the gas. This perhaps answers the question regarding the heat-insulating effect of deposits which may be formed on the heater. The rise of temperature of the material of the heater is only 15 or 20 deg. fahr. This temperature rise might be effected by a considerable deposit on the heater, but the heat generated must necessarily be liberated from the heater and given up to the gas, resulting in no error in gas measurement.

7 As to variable flow, the best evidence is presented by the curves and calculations on the chart. The entire regularity of operation, during experiments conducted under circumstances very favorable to accuracy of observation seem to show that no error is introduced by non-uniformity of flow. If such a cause of error existed, it seems probable that it would have been found during experiments such as have been made with this meter, covering the wide range of from 6000 cu. ft. to about 127,500 cu. ft. per hr. The meter is now being built so that the gas passes in a vertical direction through the heater and thermometers, and this would seem to favor regularity of distribution over the cross section of the passage. The change from horizontal to vertical position was however dictated by convenience of attachment and in order to obtain accessibility, although it seems favorable to the above-mentioned consideration.

8 During the air tests extensive fluctuations of pressure took place, due to the pulsations of the blower supplying the air. These were so great at times as to cause the water to be thrown completely out of the pressure gages, but the results obtained remained entirely regular, as shown on the chart. A small meter has been used on a single-acting three-cylinder four-cycle gas engine delivering from 30 h.p. to 60 h.p. The meter was constructed of sheet iron, and although the pressure fluctuations were such that the sides of the heater "panted" continuously, the measurement of gas was accurately accomplished.

9 Answering Mr. Dodge's question regarding the amount of energy required to measure steam with these meters, we are using 5 deg. fahr. temperature difference, which can be measured to an accuracy of 1 per cent and the energy required is 1 kw. per 1000 lb. of steam per hr. Taking a water rate of 12 lb. per h.p-hr., 1 kw. would measure the steam used for about 80 h.p.

10 Stated generally this meter seems to be particularly suitable for the measurement of dry saturated or superheated gas, air or steam.

The substance to be measured should be dry, but it may be of any pressure and temperature which the materials of construction will stand, and the measurement is independent of fluctuations of pressure and temperature. The recording mechanism can be placed in any convenient position, as, for instance, in an office, instead of near the meter, and the graphical record is thus continually observable. It is not necessary that a graphical record should be taken. An ordinary integrating wattmeter showing the amount of energy it has required, to maintain the constant temperature difference of 5 deg. between inlet and outlet of the meter, suffices as a record of rate of flow, though the variation is best shown by an autographic record.

No. 1258

TAN BARK AS A BOILER FUEL

By David Moffat Myers, New York
Associate Member

It is the object of this paper to give the chief characteristics of wet spent tan as a boiler fuel. It is believed that tan bark contains a larger percentage of moisture than any of the other moist fuels. It is perhaps safe, therefore, to assume that its correct treatment as a fuel may indicate to a certain extent the most efficient means of dealing with other moist fuels.

2 The writer has been engaged in testing, remodeling, operating and constructing steam plants depending partly or principally upon tan bark for their fuel; and in connection with this work the following tests and observations were made, not only to determine the value of the fuel, but principally to indicate the most favorable conditions for its complete combustion and to improve the economy of the plants visited.

PHYSICAL CONDITIONS

3 The nature of tan bark is too well known to require minute description, but it may be stated, as relating directly to its use as a fuel, that its condition in the fireroom varies with the method of its preparation for leaching, and its treatment in the leaches. Bark which has been finely disintegrated, and has been blown a long distance through a pipe or flue, will reach the furnaces almost in the state of a wet powder, difficult to burn. On the other hand, with less thorough disintegration and a shorter fan drive, the tan will be in larger pieces which allow a freer passage to the draft. The tan in the fireroom varies in temperature according to its final treatment in the leach house and according to the distance it is conveyed from the leach to the furnaces. Under good conditions it often reaches the fireroom at a temperature of 110 deg. fahr. The amount of moisture

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in the tan varies with the leaching process and usually runs from 62 to 70 per cent. Oak tan is easier to burn than hemlock because it is harder and does not become soggy and pack on the grates, but admits the draft more freely than the hemlock.

CALORIMETER TESTS

4 The calorific tests given in Table 1 were made by Dr. Henry C. Sherman of Columbia University, the samples of tan being dried before burning.

TABLE 1 CALORIFIC VALUES OF DRIED SPENT HEMLOCK TAN

Sample No.	B. t.u. per Pound	Sample No.	B. t.u. per Pound
1	9406	7	9519
2	9378	8	9450
3	9463	9	9504
4	9500	10	9850
5	9516	11	9472
6	9482	Average.......	9504

Note—Samples 7, 8 and 9 each contained about 6 per cent of oak tan.

EFFECTS OF LEACHING

5 Calorific tests were made to determine the effect of leaching on the fuel value of the tan, and it was found that the percentage of tannin left in the bark does not affect its calorific value per dry pound. As a check a further test was made, comparing a sample of leached bark and one of unleached bark. The result showed only a slight difference of heat units between the two and in fact a trifle more heat in the leached bark per dry pound. In other words it can be stated that the degree of leaching to which tan bark is subjected does not affect its fuel value, except inasmuch as actual weight is subtracted by the leaching process, so that a smaller quantity of fuel reaches the fireroom.

6 As regards the loss of weight due to leaching, 100 lb. of air-dry bark fed to the mill will produce 74.7 lb. of tan in the fireroom under good leaching conditions. This tan contains 65 per cent moisture, e. g., 100 lb. of air-dry bark ground at the mill will result in about 213 lb. of spent tan, containing 65 per cent moisture, in the fire-room; i. e., the weight of the spent tan is 2.13 times the weight of the bark ground.

MOISTURE

7 Ten of these samples contained moisture varying from 63.6 per cent to 68.27 per cent, the average moisture being 65.5 per cent. Many more moisture tests were made, but these ten samples are typical and represent about the average conditions in this respect. Taking the average calorific power of dried hemlock tan at 9500 B.t.u., and the average moisture at 65 per cent, and calculating the loss due to moisture, we have: Loss by moisture = $0.65[(212° - t) + 966 + 0.48\ (T - 212°)]$. For t we have the temperature of the moist fuel = 100 deg., and for T the temperature of the flue gases, which in tan burning averages about 500 deg. Substituting these values gives:

Moisture loss = $0.65\,[(212 - 100) + 966 + 0.48(500 - 212)] =$ 660 B.t.u.

8 Since a pound of fuel is 65 per cent moisture, 0.35 lb., or the dry amount, must evaporate 0.65 lb. moisture. The calorific value of 0.35 lb. of dry tan is 0.35 of 9500 B.t.u., or 3325 B.t.u., which is the total B.t.u. in a pound of moist fuel. Subtracting the moisture loss we have 3325 B.t.u. − ~~660~~ 792 B.t.u. = ~~2665~~ 2533 B.t.u., as the available heat in a pound of fuel as fired.

TAN BARK COMPARED TO COAL

9 As previously deduced, the weight of the bark as ground × 2.13 = the weight of the spent tan. Therefore the available heat for boiler purposes obtained from a pound of average air-dry hemlock bark is 2.13 × ~~2665~~ 2533 B.t.u. = ~~5676~~ 5395 B.t.u. As compared to coal of 13,500 B.t.u., 1 ton of hemlock bark ground at mill = ~~0.42~~ 0.40 tons coal.

CHEMICAL ANALYSIS

10 The chemical composition of dry tan is as follows:

	Per cent
Mineral ash	1.42
Hydrogen	6.04
Carbon	51.80
Oxygen	40.74
	100.00

This sample was hemlock tan containing 6 per cent oak tan. When dried at 110 deg. cent., it lost 66.77 moisture.

EVAPORATIVE TESTS

11 The actual evaporative power of tan was taken from the results of 22 complete boiler tests burning tan alone. Eight of these were thermal-efficiency tests.

12 Four complete thermal-efficiency tests were also made using both tan and coal for fuel. The above tests were conducted in accordance with the code of The American Society of Mechanical Engineers, and in different parts of the country, including the South, the East and the Middle West. The type of boiler in every case was the horizontal tubular. The furnaces varied considerably in design and economic results; but they were all of the Dutch oven type, set in front of the boiler; and all were fired from the top through feed-holes, with the exception of the furnace designed by the writer, the results and design of which will be described later. All the above tests were made under natural draft. Tables 2 and 3 show one of the 24-hr. boiler and furnace efficiency tests, selected to show what may be considered ordinary conditions and results in tan burning.

13 By designing and constructing a special furnace for the purpose the writer obtained a thermal efficiency of furnace and boiler of 71.1 per cent, burning hemlock tan, efficiency based on available heat of fuel. The principal features of this furnace, which was sub-

TABLE 2 BOILER TEST "X" TANNERY

DATA AND RESULTS OF EVAPORATIVE TEST ON BOILER NO. 6 TO DETERMINE HORSEPOWER DEVELOPED AND EFFICIENCY OF EXISTING METHODS AND CONDITIONS FOR BURNING WET SPENT HEMLOCK TAN AS FUEL FOR STEAM GENERATION

Kind of fuel	Wet spent hemlock tan
Kind of furnace	Wide high oven, 2 rows firing eyes
Method of starting and stopping test	Alternate
Grate surface, square feet	92
Water heating surface, square feet	2089
Heating surface + grate surface	22.7

TOTAL QUANTITIES

Date of trial	November 5 and 6, 1902
Duration of trial, hours	24

Weight of tan as fired, pounds....53,000
Percentage of moisture in tan....65.6
Total weight of dry tan consumed, pounds....18,232
Total weight of water fed to boiler, pounds....69,825
Factor of evaporation....1.1216
Equivalent water evaporated into steam from and at 212 deg., pounds....78,316

HOURLY QUANTITIES

Wet tan consumed per hr. pounds....2208
Dry tan consumed per hr., pounds....1823
Water evaporated per hr., pounds....2909
Equivalent evaporation per hr., from and at 212 deg., pounds....3263

AVERAGE TEMPERATURES, PRESSURES, ETC.

Steam pressure by gage, pounds....70.25
Temperature feed-water entering boiler....126.5
Temperature escaping gases from boiler....471
Force of draft between damper and boiler, inches water....0.39
Temperature outside air....64
Temperature of tan as fired....83

HORSEPOWER

Horsepower developed....94.6
Rated horsepower, at 15 sq. ft....139
Percentage rated horsepower developed....68

ECONOMIC RESULTS

Water evaporated under actual conditions per lb. tan as fired, pounds....1.32
Equivalent evaporation from and at 212 deg. per lb. tan as fired, pounds....1.48
Equivalent evaporation from and at 212 deg. per lb. dry tan, pounds....4.30

EFFICIENCY

Calorific power of dry tan per lb., British thermal units....9463
Percentage moisture in tan as fired....65.6
Available heat in 1 lb. of wet tan as fired, after subtracting loss due to evaporating moisture, British thermal units....2443
Efficiency of boiler including furnace, based on available heat in fuel, per cent....58.2
Efficiency of boiler including furnace, based on total heat in fuel, per cent....43.9

FIRING

2 rows of eyes, 3 eyes in each row. Eyes were fired according to regular method employed at this tannery, i. e., they were refilled as they burned low, a large amount of tan being fired each time. 51 eyes were fired during the 24-hr. test.

TABLE 3 FLUE-GAS ANALYSIS

DAMPER WIDE OPEN

No.	TIME	CO_2 %	O %	CO %	FIRING
1	9:45	10.8	7.5	....	30 minutes after
2	10:21	4.2	12.3	....	30 minutes after
3	11:48	11.5	7.0	....	14 minutes after
4	12:29	5.0	15.0	....	22 minutes after
5	1:37	4.6	15.1	0.5	While firing
6	2:06	5.0	13.0	0.3	14 minutes after
7	3:05	9.2	....	....	40 minutes after
8	4:08	9.8	8.4	....	While firing
9	5:30	12.6	6.6	0.1	5 minutes after
10	6:20	12.6	7.0	0.5	26 minutes after
11	6:55	11.6	7.4	0.2	25 minutes after
12	7:50	13.0	6.0	0.6	20 minutes after
13	8:55	13.0	7.2	0.1	While firing
14	10:09	11.0	9.1	...	6 minutes after
15	11:07	12.5	6.9	0.1	29 minutes after
16	12:23	11.5	7.5	0.8	23 minutes after
17	1:07	7.6	11.8	1.1	67 minutes after
18	2:10	12.5	7.7	0.0	24 minutes after
19	2:50	6.0	13.5	...	20 minutes after
20	4:00	4.3	15.7	...	58 minutes after
21	5:30	3.0	16.4	0.3	While firing
22	6:02	11.0	8.8	0.0	27 minutes after
23	6:37	9.0	9.8	0.0	6 minutes after
24	7:35	8.7	11.0	0.2	9 minutes after
Average	....	9.2	8.2	0.32	

sequently installed in a number of plants, were: *a* Large combustion space over the burning fuel. *b* Automatic stoking with rotating comb shafts. *c* Oppositely inclined grate surfaces converging downward to a set of shaking and dumping grates. *d* Drying on dead plates over which fuel passed before receiving air supply. *e* Concentrated draft from opposing grate surfaces to a focus of combustion, caused by parallel spacing of longitudinal flat grate bars with beveled edges. *f* Reverberating draft action resulting from concentrated draft-currents and the curvature of the arch, thus directing flames back upon dead plates. The full results of a test on this furnace are given in Table 4.

EFFECT OF PRESSING AND BURNING MIXED WITH COAL

14 It was desirable to learn the exact economic effect of burning coal with pressed tan and also the result of pressing the tan to eliminate some of the moisture. For these purposes three thermal-

TABLE 4 BOILER TEST: AUTOMATIC STOKER FURNACE DESIGNED BY THE WRITER

DATA AND RESULTS OF EVAPORATIVE TEST ON BOILER NO. 1 TO DETERMINE HORSEPOWER DEVELOPED, AND EFFICIENCY OF BOILER WITH AUTOMATIC TAN STOKER

Kind of fuel	Wet spent hemlock tan
Kind of furnace	Automatic stoker
Method of starting and stopping test	Alternate
Grate surface, square feet	61.5
Water heating surface, square feet	1795
TOTAL QUANTITIES	
Date of trial	August 14, 1903
Duration of trial, hours	12
Weight of tan as fired, pounds	18,600
Percentage moisture in tan	65.3
Total weight of dry tan consumed, pounds	6449
Total weight of water fed to boiler, pounds	33,004
Factor of evaporation	1.085
Equivalent water evaporated into steam from and at 212 deg., pounds	35,809
HOURLY QUANTITIES	
Wet tan consumed per hr., pounds	1550
Dry tan consumed per hr., pounds	537
Water evaporated per hr., pounds	2750
Equivalent evaporation per hr. from and at 212 deg., pounds	2984
AVERAGE TEMPERATURES, PRESSURES, ETC.	
Steam pressure by gage, pounds	43
Temperature feed-water entering boiler	155
Temperature inside furnace	1100
Temperature in combustion chamber	1475
Temperature gases escaping from boiler	493
Temperature outside air	71
Temperature of air entering ash pit	78
Temperature of tan as fired	101
Force of draft between damper and boiler, inches water	0.4
HORSE POWER	
Horsepower developed	86.5
Rated horsepower of boiler at 15 sq. ft., per h.p.	120
Percentage rated horsepower developed	72.1
ECONOMIC RESULTS	
Water evaporated under actual conditions per lb. of tan as fired, pounds	1.77
Equivalent evaporation from and at 212 deg. per lb. of tan as fired, pounds	1.93
Equivalent evaporation from and at 212 deg. per lb. of dry tan, pounds	5.55

EFFICIENCY

Calorific value of dry tan per lb., B.t.u. 9850
Percentage of moisture in tan as fired 65.33
Available heat per lb. of tan as fired, B.t.u. 2623
Efficiency of boiler, including furnace, based on available heat in fuel, per cent 71.1
Efficiency of boiler, including furnace, based on total heat in fuel, per cent... 54.4

FIRING

Firing was even and continuous with automatic stoking. Bed of fuel maintained on grates was about 9 in. thick.

TABLE 5 FLUE GAS ANALYSES

DAMPER OPEN

No.	TIME	CO_2 %	O %	CO %
1	2:25	11	7.5	0
2	4:30	12.8	5.6	0
3	7:15	12	7.8	0
4	9:05	13	6.2	0
5	11:10	13	6.4	0
Average		12.4	6.7	0

TABLE 6 ANALYSIS OF SPENT BARK

CHEMIST'S NO. S.-736, ANALYZED AUGUST 17, 1903

	%		%
Moisture	65.33	Non-tannins	2.35
Total solids	5.11	Available tannin	1.89
Total soluble solids	4.24	Reds	0.87

efficiency tests were conducted, all on the same boiler and furnace, the conditions as far as possible being the same in all tests, and the furnace operated by the same firemen. The furnaces and boiler were of the general type already referred to. The combustion chamber under the boiler differed by having a system of air admission through the bridge wall. The opening in the wall for admitting air to this system was set one-fourth open in the first test and remained so in all three tests, each of ten hours duration. An effort was made in each of the two tests without coal to equal the boiler output of the test in which coal was used; but as shown below this could not be done.

15 The furnace arch was equipped with two rows of firing-eyes of three eyes each, and two eyes on the same side of the furnace were usually fired at a time. The firing was heavy, the cone of tan nearly reaching the arch each time after firing. The furnace had a grate surface of 11 ft. 4 in. by 7 ft. 0 in. = 79.3 sq.ft., and the perpendicular distance from the grate surface to the highest point of the arch inside was 48 in. The boiler was 6 ft. by 18 ft. and contained 2089

TABLE 7 COMPARATIVE RESULTS OF THREE TESTS

	Test No. 1	Test No. 2	Test No. 3
Thermal efficiency of boiler including furnace, based on available heat in fuel, per cent	63.4	59.4	59.
Evaporation from and at 212 deg. per lb. tan as fired, pounds	1.98	1.90	1.65
Evaporation from and at 212 deg. per lb., dry tan, pounds	5.43	4.71	4.54
Percentage rated h.p. developed	135.5	92.	72.4
Type of furnace		wide and high	
Heating surface ÷ grate surface	26.3	26.3	26.3
Number of firing-eyes	6	6	6
Depth of combustion chamber, inches	48	48	48
Kind of firing	deep	deep	deep
Firing intervals, minutes	18.8	18.8	19.4
Kind of fuel	pressed tan and coal	pressed tan	unpressed tan
Moisture in tan, per cent	60.4	59.6	63.6
Steam pressure, pounds	76	74	63.4
Temperature feed-water	163	160	161
Flue temperature	561	486	445
Force of draft in uptake, inches water	0.51	0.49	0.43
Duration of trial, hours	10	10	10
Flue-gas analyses, averages per cent CO_2	13.8	10.9	10.5
Per cent O	4.9	7.8	8.6
Per cent CO	0.15	0.93	0.68

sq.ft. heating surface, the ratio of heating surface to grate surface being 26.3 to 1.

16 Nine flue-gas analyses were made during test No. 1, seven in each of the following tests, and the averages of these results are shown in Table 7 which gives the most essential results of the three tests.

17 The results of this series of tests show:

a That the burning of coal with pressed tan increased the thermal efficiency from 59.4 per cent to 63.4 per cent.

b That the burning of coal with pressed tan increased the boiler output from 92 per cent of rated capacity to 135.5

per cent of rated capacity, the force of draft differing by only 0.02 in. water-gage.

c That the degree of combustion as indicated by the amount of CO_2 was raised from 10.9 per cent CO_2 with pressed tan to 13.8 per cent CO_2 with pressed tan and coal.

d That the flue temperature was increased from 445 deg. with unpressed tan and 486 with pressed tan to 561 deg. with pressed tan and coal.

e That the thermal efficiency based on available heat obtained with pressed tan was 59.4 per cent as compared to 59 per cent with unpressed tan.

f That the burning of pressed tan gave a boiler output of 92 per cent rated capacity as compared to 72.4 per cent with unpressed tan, the draft being only 0.04 in. water-gage stronger in the test of greater capacity.

g That since the thermal efficiency was practically the same with both pressed and unpressed tan the principal advantage of pressed over unpressed tan in this particular case lies in the intrinsically greater calorific value of the pressed tan owing to its reduced moisture. In this particular case, in which the unpressed tan contained 63.6 per cent and the pressed tan 59.6 per cent moisture, the actual gain in available heat-units was 4.5 per cent over the unpressed tan or 1⅛ per cent gain in heat value for each per cent decrease in moisture.

TAN PRESSES

18 Tan presses for reducing the moisture in actual practice result in a total economic gain of only about 7 per cent; and against this must be charged the power to run them, maintenance, repairs and often the disadvantage of noise and vibration in the fireroom.

19 The question of installing a press must be determined principally by local conditions, such as amount of tan available, amount and proportion of coal used and its cost, intelligence of labor, average moisture in the tan as delivered to the fireroom, design of furnace and grate, cost of changing such design, etc. Especially when the firing is poor a press is of much value, for certain firemen who can get excellent steaming with pressed tan cannot raise steam with unpressed tan. When a press is introduced for tan burning and the moisture is reduced, the rate of combustion with given grate and draft

is increased. Therefore to burn the pressed tan in the time required to burn the wet tan, it is necessary to reduce the grate surface proportionately.

20 For example, two complete 24-hr. evaporative tests were made on the same boiler and furnace, with the same firemen and with the same conditions as nearly as possible, the first with unpressed and the second with pressed tan, with the following results:

Test No. 1, without press, 64.4 per cent moisture, consumption 14,565 lb. dry tan with 0.42 draft.

Test No. 2, with press 60.4 per cent moisture, consumption 19,879 lb. dry tan with 0.38 draft.

Thus the rate of combustion on the same grate was increased 36.5 per cent. (It may here be noted that the boiler-horsepower developed in the first test was 89.3 h.p., and in the test with pressed tan 116.8 h.p.)

21 Therefore in this case to burn the tan in the same time as formerly it would be necessary to reduce the grate surface about 27 per cent. This is based on the above drop in moisture from 64.4 per cent to 60.4 per cent, or 4 per cent. The drop in moisture with fair pressing will run between 6 and 7 per cent, representing a total gain of available heat of 7 to 8 per cent. With good pressing the grate surface may be properly reduced one-third.

22 There have been cases of failure with pressing the tan simply because the precaution of reducing grate surface was not taken. Tannery superintendents have said, "We have tried presses and had to throw them out, because the tan would not last throughout the day, and we had to use more coal to make up the deficiency." The point here is that a tannery will almost invariably consume or waste every pound of steam its boilers will supply. With the press in operation the steaming was increased proportionately as long as the tan "lasted," no provision being made to correct the grate surface, and consequently the result would be exactly as stated by the superintendent. This has led to, or confirmed, the erroneous opinion held by some that the moisture in tan aids its combustion.

EFFECT OF SMALL COMBUSTION SPACE OVER THE FIRE

23 A special test on a low-arched furnace was made to obtain comparison with high arches. A furnace containing nine feed-holes,

and having a height from arch to grate of 26 in., was tested, with the following results[1]:

Efficiency of boiler and grate, per cent	26.4
Evaporation from and at 212 deg. per lb. tan as fired	0.63
Percentage rated h.p. developed	39
Heating surface ÷ grate surface	28
Depth combustion chamber under boiler	3′6″
Intervals between firing, minutes	6.3
Kind of tan	Hemlock
Percentage moisture in tan	67.2
Temperature flue-gases	422
Draft at damper, inches	0.37

24 These miserable results are to be attributed to the low arch and the consequently inadequate combustion-space between the fuel and the arch of the furnace. Owing to this small space the velocity of draft and gases was very high, thus removing the volatile fuel-gases from the furnace before their combustion could be fairly begun, and allowing no heating or mixing action in the furnace, and when they struck the boiler the combustion was checked or stopped.

25 The worst evaporative results ever obtained on various designs of higher furnaces, on tan alone and with poor firing, are at least 100 per cent better than this, and these other furnaces ran from 3 ft. 0 in. to 5 ft. and 6 ft. high; it is perhaps significant that the highest thermal efficiency was obtained in the furnace with the greatest amount of free combustion space between the fuel bed and the top of the arch.

26 During recent years tanneries have been using more extracts and less bark for their liquors. At the same time a greater amount of machinery has been introduced and it has been found beneficial to use more heat in leaching. All these changes have resulted in greater steam requirements and less spent tan bark for its generation. Therefore more coal is required in the boiler room and stricter economy of fuel is necessary.

BURNING A MIXTURE OF TAN AND COAL

27 Among other methods of burning the two fuels, the ordinary coal-burning setting has been tried. The grates are usually of the shaking variety and are set directly under the boiler, and generally only 24 in. to 30 in. beneath the shell of the boiler. The coal and tan

[1]Average flue-gas analyses: CO_2, 4.3 per cent; O, 11 per cent; CO, 20 per cent

are fired alternately by hand. Owing to the large volume of tan as compared to its heating value (spent tan weighs about 33 lb. per cu. ft.), the fire doors have to be opened very often, thus admitting a great excess of cold air to the fire. The fire bed has a strong tendency to burn through in spots and form blowholes. The temperature-retaining effect of the Dutch oven arch is lost, and instead, the fuel gases come into direct contact with the shell of the boiler, which cools them before combustion is fairly under way. The result of every such case the writer has observed has been a dull, smoky fire of greatly varying temperature and never good combustion. In a certain plant where this method was thoroughly tried out, it proved so unsuccessful that it was discontinued and the grates were replaced by Dutch ovens in which the mixture of tan and coal was burned, the fuels being mixed before firing.

28 The above illustration affords a good comparison of combustion with and without a brick arch over the fire bed with hand firing, and goes to show that a brick or refractory arch is a necessity for good combustion, when the two fuels are so related in quantity that their heat values are about equal. As a further demonstration, experiments with automatic stoking may be cited. The mixed fuel was fed into a chain-grate stoker operating on induced draft. This was done under the most favorable conditions possible and under the supervision of an engineer representing the chain-grate company, with the assistance of the writer. The steam pressure and the furnace temperature dropped rapidly, and it was soon necessary to eliminate the tan and feed coal alone, to prevent the fire from going out. Experiments with different methods of mixing and feeding the two fuels all resulted in killing the fire.

29 Like experiments were conducted on a Detroit stoker set in front of boiler and having a brick arch completely over the fire. No scientific tests were conducted, but the mixed fuel easily carried the load formerly carried by the coal, it being necessary only to increase the speed of the stoker. The steam pressure was increased and as far as the eye could detect there was not the slightest decrease of furnace temperature, the combustion being clear and to all appearances excellent.

30 From these experiments it seems entirely safe to state that for efficient combustion of a mixture of tan and coal in ratio by weight of 5.2 to 1 a refractory arch over the greater part of the fire, and preferably over the entire fire-bed, is not only advisable but necessary. It follows that tan bark of the usual moisture and heat content can-

not be burned without the application of a refractory arch, or some device of similar function, such as a large combustion chamber lined with refractory material.

31 Wood is sometimes burned in combination with tan in the same furnace. Good results are obtained when the wood is ground to about the same fineness as the tan, when it is known in the sawmill districts as "hog feed." When wood in slab or log form is burned in the same furnace with tan the results naturally are usually very poor, owing to blowholes formed in the fire bed and consequent large excess of air. A case of this description came under the writer's observation. "Edgings" from a nearby sawmill cost less than coal per 1000 B.t.u., and it was the custom to fire the edgings by hand into the front fire doors of the tan furnaces, the tan being fired from above in the usual manner. A large amount of labor was required and the furnace fronts required constant repairing. By substituting coal for the wood in this case, and mixing with the tan before firing, a very good saving was effected in fuel, labor and repairs.

32 In regard to draft for tan burning, in 26 evaporative tests on tan-burning furnaces the average force of draft between boiler and damper was 0.45 in. water gage.

33 The average draft, in 13 tests on tan where less than the rated capacity of the boiler was developed, was 0.42 in. water gage between boiler and damper.

DRAFT AND GRATE SURFACE

34 The average draft, in 12 tests on tan where rated capacity up to 60 per cent over-capacity was developed, was 0.47 in. water gage. This comparison is significant only in a very general way, owing to the great variety of ratios of heating to grate surfaces in the different tests, and the different methods and time intervals of firing. The highest force of draft found in any case between damper and boiler with chimney draft, furnaces burning tan only, was 0.63 in., and the lowest 0.32 in.

35 The lightest draft with which rated capacity was obtained was 0.32 in., and in this case the ratio of heating surface to grate surface was 21.8 to 1. In the test where 160 per cent rated capacity was developed the force of draft was 0.51 in. and the ratio of heating surface to grate surface was 19.3 to 1. The greatest ratio of heating surface to grate surface which developed full rated capacity was 29.9 to 1, giving 110 per cent rated capacity with force of draft in uptake

0.42 in. The firing in this case was excellent with intervals of 7.4 min.

36 Owing to space required, facility in cleaning fires, minimum radiation from furnace and expense of building and repairing, it is desirable to make the furnace grate area as small as possible and still conform to capacity requirements. It is therefore necessary to know the boiler-horsepower developed per square foot of grate surface. The force of draft, method of firing and ratio of heating surface to grate surface, are all factors in this result, also the moisture in the tan bark. Table 8 gives the most important factors with results of nine actual tests.

TABLE 8 RESULTS OF NINE TAN-BURNING TESTS

Test No.	Tan	Moisture %	Draft Inches	H.S. / G.S.	Firing Intervals Minutes	B.H.P. per Sq. Ft. G. S.
1	Oak	0.53	0.63	29	15	2.35
2	Hemlock	....	0.51	19.3	8	2.07
3	Oak	0.64	0.58	29	5	2.37
4	Oak	0.64	0.61	34.8	....	1.53
5	Hemlock	0.65	0.61	17.4	50.8	0.92
6	Hemlock	....	0.32	21.8	5.2	1.51
7	Hemlock	0.68	0.39	29.9	7.4	2.34
8	Hemlock	0.64	0.42	22.7	30	0.97
9	Hemlock	0.65	0.40	29.2	Continuous	1.31

37 Inspection of this table shows that one factor above all the others influences the b.h.p. developed per square foot of grate, and this factor is the method of firing. Thus the tests in which the time intervals between firing are small show a marked increase in b.h.p. per square foot of grate. This test is upheld by many other tests of the writer. From the table we may take the following average results:

For oak tan with 60 per cent moisture, 0.61 in. draft, in uptake of horizontal tubular boiler, and good firing of 15 min. or less between firing, 2.08 b.h.p. per sq.ft. of grate surface.

For hemlock tan with 65½ per cent moisture, 0.44 in. draft in uptake of horizontal tubular boiler, and good firing of say 17 min. or less between firing, 1.52 b.h.p. per sq. ft. of grate surface.

GRATE SURFACE FOR COAL AND TAN

38 The above results are for burning tan alone. At the present time, however, it is very general practice, as previously set forth, to burn coal with the tan. The best results in most cases are obtained when the coal is mixed evenly with the tan, and the mixture burned in a Dutch oven with shaking and dumping grates. This practice modifies the boiler horsepower per square foot of grate. The grate surface often allowed is at the rate of about 3½ b.h.p. per sq. ft. of grate surface, the ratio of heating surface to grate surface ranging from 50 to 1 to 33 to 1.

TABLE 9 RESULTS OF TESTS OF BURNING COAL WITH TAN IN DUTCH OVENS

Test	Tan	Grates	Tan Moisture %	Uptake Draft	H. S. / G. S.	Firing Intervals Minutes	Weight of Coal to Weight of Tan as Fired	B.H.P per sq. ft. of Grate
a	Hemlock	Shaking	68	0.40	33.5	20	1 to 8.9	2.52
b	Hemlock	Stationary	60.4	0.51	28.1	19	1 to 6.1	2.53
c	Hemlock	Stationary	66.7	0.67	37.3	6.3	1 to 46.4	2.70
**d*	Oak	Shaking	67.4	0.7	29.4	47	1 to 2.9	2.66
e	Hemlock	Stationary	66.5	1.05	32.8		1 to 22.8	1.33

*Tubular Boiler, coal fired independently in front doors of double arched tan furnace. The 47-min. interval refers to the tan only.

39 Of course, as with tan alone, the development per square foot of grate depends upon the same factors, with the additional factors of the proportion of coal mixed with the tan, and the B.t.u. and character of the coal. The figures in Table 9, selected at random, show the results in terms of b.h.p. per square foot of grate surface obtained in actual tests of burning coal with tan in Dutch ovens. It is of interest here to note that when the ratio of weight of coal to weight of tan as fired is 1 to 5.2, then the heat developed by each fuel is practically the same. This is figuring on 13,000 B.t.u. coal, and "tan as fired" containing 2500 B.t.u. available heat, which is good average value.

40 In further reference to draft, a test which showed excellent results was made on a plant operating on induced fan draft. A small proportion of coal was used in the tan furnaces and a thermal efficiency of 66.1 per cent was found under regular working conditions, the CO_2 in the flue gases averaging nearly 12 per cent. The writer has also observed an oak tan-burning furnace operated temporarily on fan

draft with closed ashpit. The combustion was apparently very good; a hot fire was maintained, but additional care was required to keep blow-holes from forming on the grates.

41 A reliable report has also been obtained on the installation of a steam-induced forced-draft system, applied under the grate of a tan-burning furnace. According to the report, the steam used for the draft condensed on the grate bars, with low temperature due to contact with the large mass of wet tan, and the concern that made the installation after considerable experimenting removed the appliance and gave up tan burning.

GRATES

42 Grates for tan burning are made in different patterns, but usually contain from 20 per cent to 30 per cent air space, the actual opening between the bars being $\frac{3}{16}$ in. to $\frac{1}{4}$ in. Larger spaces than these allow the tan to fall through into the ashpit. A shaking grate is hardly necessary for burning tan alone, owing to the small percentage of ash. No clinker is formed, only a very fragile crust on the grate obtains in three or four hours. In some cases fires are cleaned only once in 12 hr. with good results. Unlike a coal fire, a tan fire should be shaken or disturbed very little. If a slice bar is used on a brightly burning tan fire, dense smoke results and the flame is killed for some time.

43 The temperature in the throat of a properly designed tan furnace, burning tan alone, will reach 1500 deg. fahr.

44 The depth of fuel on the grate varies with the design of the furnace and the method of firing. In ordinary practice the tops of the cones of tan directly beneath the firing eyes vary from 2 ft. to 5 ft. above the grate surface, while the depth of tan where the cones meet will be from 6 in. to 18 in., depending on the design and firing of the furnace.

45 When spent hemlock tan forms a cone from the conveyor-discharge on the fireroom floor it forms an angle of slide of about 55 deg. to the horizontal. Inside a hot furnace, however, this angle is about 45 deg., and in laying out a tan furnace the latter angle is used in figuring the distribution of fuel on the grate. The angle of slide of tan on a sloping grate is 39 deg. to 41 deg., and these angles were used in designing the automatic tan furnace referred to previously.

46 Among other experiments with this fuel the writer has made

tests to compare results of double settings with single settings. By double setting is meant a setting so arranged that two furnaces discharge their gases into a common combustion chamber beneath two or more boilers. Tests showed no tangible results in favor of either method as regards evaporation or fuel economy. It is possible however to imagine a case where the fire in a furnace might be in poor condition, when if the setting were a double one, the second furnace might maintain in the common combustion chamber a temperature sufficiently high to ignite and burn fuel gases discharged from the dull fire which otherwise would escape unburned. It is the opinion of the writer, however, that owing to the impossibility of laying off a single boiler for cleaning or repairs, a double setting is less desirable than a single setting. Furthermore, good firing and handling would in any case offset any possible advantage of the double setting.

47 The combustion of tan, as indicated by flue-gas analyses made with an Orsat apparatus, compares most favorably with the combustion of coal in factory plants where the same amount of attention is paid to the boiler room. In fact, the percentage of CO_2 runs higher than in the ordinary coal-burning plant. For instance, the average CO_2 of seventeen evaporative tests on different tan-burning furnaces was 11.3 per cent; the lowest average CO_2 in any of the tests was 4.3 per cent, which represented unusually and abnormally poor conditions; and the highest average CO_2 in any one of them was 16.9 per cent. The CO averages about 0.5 to 0.6 per cent and ranges from practically zero up to 2 per cent.

SUMMARY

48 The following statements concerning tan bark as a boiler fuel have been demonstrated by the foregoing data and confirmed by constant checking in actual practice:

49 *Moisture.* In condition for firing, wet spent hemlock tan usually contains close to 65 per cent of moisture.

50 *Available B.t.u.* Bomb calorimeter tests on many samples of spent hemlock tan give an average value of about 9500 B.t.u. per lb., sample being dried before burning. The available heat per pound as fired, after subtracting moisture loss, is about ~~2665~~ 2700 B.t.u.

51 *Ratio weights.* Since the weight of the spent tan in the fire room is 2.13 times the weight of bark ground at the mill, 1 lb. of ground bark produces an available heat value of ~~5676~~ 5755 B.t.u. Hence,

as compared to coal of 13,500 B.t.u., one ton of hemlock bark is equivalent to ~~0.42~~ 0.10 ton coal.

52 *Effect of leaching on B.t.u.* The heat value of spent hemlock tan is not affected by the degree of leaching, except inasmuch as the actual weight is affected.

53 *Chemical composition.* The chemical composition of hemlock tan (sample containing 6 per cent of oak) was:

$$C = 51.8, H = 6.04, O = 40.74, \text{Mineral ash} = 1.42 \text{ per cent}$$

As a fuel analysis this is of interest as regards the small amount of ash and the large amount of oxygen.

54 *Boiler tests.* The actual operation of tan bark as a fuel is shown by complete boiler tests made under working conditions, records of which are given above.

55 *Improved efficiency.* A considerable improvement in efficiency was produced by a specially designed furnace providing automatic feeding, large combustion space over fuel and special draft admission.

56 *Tan Presses.* The use of presses for reducing the moisture in the tan before firing may be good economy if the amount of tan compared to the amount of coal used is considerable, and providing the grate surface is properly reduced to meet the demands of the more rapid combustion. In practice the grate surface is sometimes reduced one-third on the introduction of tan presses.

57 Under usual tan-burning conditions the total gain in available heat from the use of a press is about 1½ per cent for each per cent drop of moisture content, and this drop rarely exceeds 7 per cent, i. e., a reduction of moisture from say 65 to 58 per cent.

58 The writer is of the opinion that the combined boiler and furnace efficiency would naturally rise considerably when tan presses are used, owing to increased furnace temperature and better combustion, although as shown by Table 7 the thermal efficiency was raised only from 59 per cent to 59.4 per cent, the moisture in the tan being 63.6 per cent and 59.6 per cent in the two tests respectively. This may be explained, at least partially, by the fact that the grate surface was not reduced for burning the pressed tan.

59 *Addition of coal.* As a result of special comparative tests, made under similar conditions on the same furnace and boiler, the addition of about one pound of coal to six of pressed tan increased the combined furnace and boiler thermal efficiency from 59.4 per

cent to 63.4 per cent. The addition of coal also resulted in raising the average CO_2 in the flue gases from 10.9 to 13.8 per cent. The percentage rated capacity of the boiler was increased from 92 per cent (without coal) to 135.5 per cent with coal.

60 Although the above precise statements were obtained from a single pair of accurate tests, the writer's subsequent experiences in many plants confirm the results as generally typical; and it may be safely stated that under usual working conditions the addition of coal to tan produces a decided increase in degree of combustion and a consequent rise of thermal efficiency. The increase of boiler output attending the use of coal with tan is also a factor of economy not to be disregarded.

61 *Ample combustion space.* One of the most important factors in designing furnaces for tan-burning, is that of ample combustion space. In usual tan-burning practice this means a high arch over the furnace. Low-arched furnaces are conducive to bad combustion, whereas furnaces providing an arch at a considerable height above the fuel on the grates give decidedly better combustion and higher efficiency. (See tests and discussion, par. 23 *et seq.*)

62 *Refractory arch.* A refractory arch or similar combustion arrangement is necessary for the economic combustion of tan. Tan in its usual condition cannot be burned in a common coal-burning setting without an arch separating the fire from the cooling surface of the boiler shell or tubes. Even when tan is enriched with coal to a point where the two fuels form equal heating values (about 1 lb. coal to 5.2 lb. wet spent tan) good combustion cannot be obtained in an ordinary coal setting. This holds true both for hand-firing and automatic stoking. (See tests and discussion, par. 27 *et seq.*)

63 *Draft.* The force of draft required in general practice for tan burning is somewhat higher than that required for producing the same boiler output with bituminous coal. No definite figures can be given for force of draft as related to tan burned per square foot of grate surface, owing to the widely varying designs of furnaces and methods of firing. It may be stated, however, that for full capacity not less than ½-in. draft should be provided for in the uptake of a horizontal tubular boiler, when the boiler is set so that the gases flow under the shell and through the tubes directly to the stack or breechen.

64 *Horsepower per square foot of grate.* The boiler-horse power obtainable per square foot of grate surface, depending upon the rate of combustion of the tan, is an extremely variable quantity; but aver-

aging the results of nine typical boiler tests it may be said that under usual conditions for oak tan the boiler-horsepower per square foot of grate is about 2.08; and for hemlock tan, 1.5 boiler-horsepower per square foot of grate is an average figure. As demonstrated by results of tests given in Table 8, the one factor which above all others influences the boiler output per square foot of grate is the method and comparative excellence of the firing, very short firing-intervals producing the higher output results.

65 *Horsepower with coal and tan.* When coal is burned in the same furnace with the tan the development per square foot of grate varies of course with the richness of the mixture and with the other usual conditions, and it is best in designing furnaces for any particular plant to use the results of tests made under similar conditions. In four cases out of five selected at random the boiler-horsepower per square foot of grate was 2.5 to 2.7.

66 *Draft area and air spacing.* Grates for tan burning provide 20 to 30 per cent draft area and the spacing is from $\frac{3}{16}$ in. to $\frac{1}{4}$ in.

67 *Clinker and ash.* Tan forms no hard clinker, only a light fragile crust, and a very small amount of ash (less than $1\frac{1}{2}$ per cent based on dry matter), and a shaking grate is unnecessary except when coal is fired with the tan.

68 *Furnace temperature.* Excellent combustion of tan has given a temperature of 1500 deg. fahr., in the throat of the furnace.

69 *Single vs. double setting.* As a result of comparative tests made on single vs. double settings for tan burning, the writer believes that the single setting should have the decided preference. A possible though not a decided improvement in combustion is sometimes obtainable with the double setting, but this is more than offset, especially in the small plant, by the advantages of convenience for cleaning, repairs and adaptation to load given by the single setting.

70 *Use of wood with tan.* When wood is used as a supplementary fuel to tan and fired in the same furnace, the best results are obtained by first grinding the wood to the size known as "hog feed" The firing of wood in log or slab form is disastrous both to the furnaces and to combustion.

71 *Flue gas analyses.* Basing comparison on flue gas analyses, tan burns with a higher combustion than coal, under equally favorable conditions. The large amount of moisture in the tan produces a comparatively low furnace temperature, even with good chemical combustion, and acts against an equally high combined efficiency of urnace and boiler.

72 The highest thermal-efficiency test obtained gave 71.1 per cent efficiency (on boiler and furnace, based on available heat in tan), and flue-gas analyses during this test averaged 12.4 per cent CO_2. The approximate relation of CO_2 to efficiency as obtained under actual running conditions is shown by the curve, Fig. 1, which was plotted

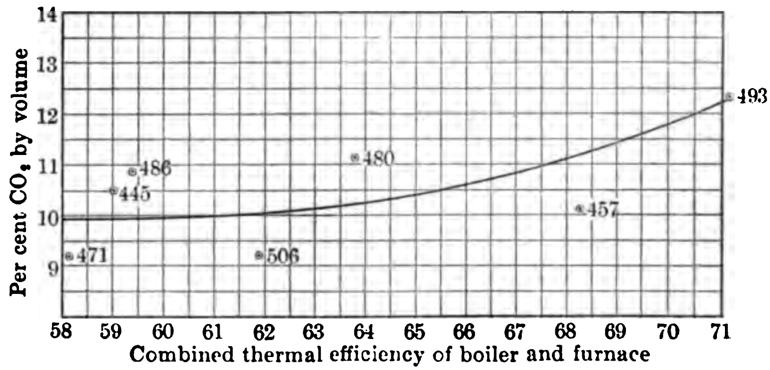


FIG. 1 CURVE SHOWING RELATION OF CO_2 TO EFFICIENCY IN TESTS ON TAN ALONE

from seven thermal efficiency tests made on various designs of furnaces burning tan alone. The figure given at each point plotted is the temperature of the escaping gases during the test.

73 The following illustrations, reproduced from working drawings and sketches, will give some idea of the construction of furnaces for burning spent tan bark, sawdust, and bagasse.

74 Fig. 2 is a working drawing of the self-feeding tan-burning furnace designed by the writer and referred to in Par. 12 and Table 4.

75 Fig. 3 shows the application to bagasse burning of special grates shown in Fig. 4. The stokers are done away with in this case, the fuel being fed by gravity to the feed chutes with weighted flaps which are used all over the islands of Cuba and Porto Rico. This burner has not yet been applied to bagasse burning.

76 Fig. 4 shows the construction of the grate bars, which provide the horizontal draft opening tending to produce the draft action referred to in Par. 13.

77 Fig. 5 and Fig. 6 show the types of tan furnaces found by the writer in common use throughout the country. Fig. 5 shows what was known as the old Hoyt furnace. It was originally designed when tan bark was so plentiful that it was necessary to burn it. The writer has found these furnaces with inside lengths as great as 24 ft.

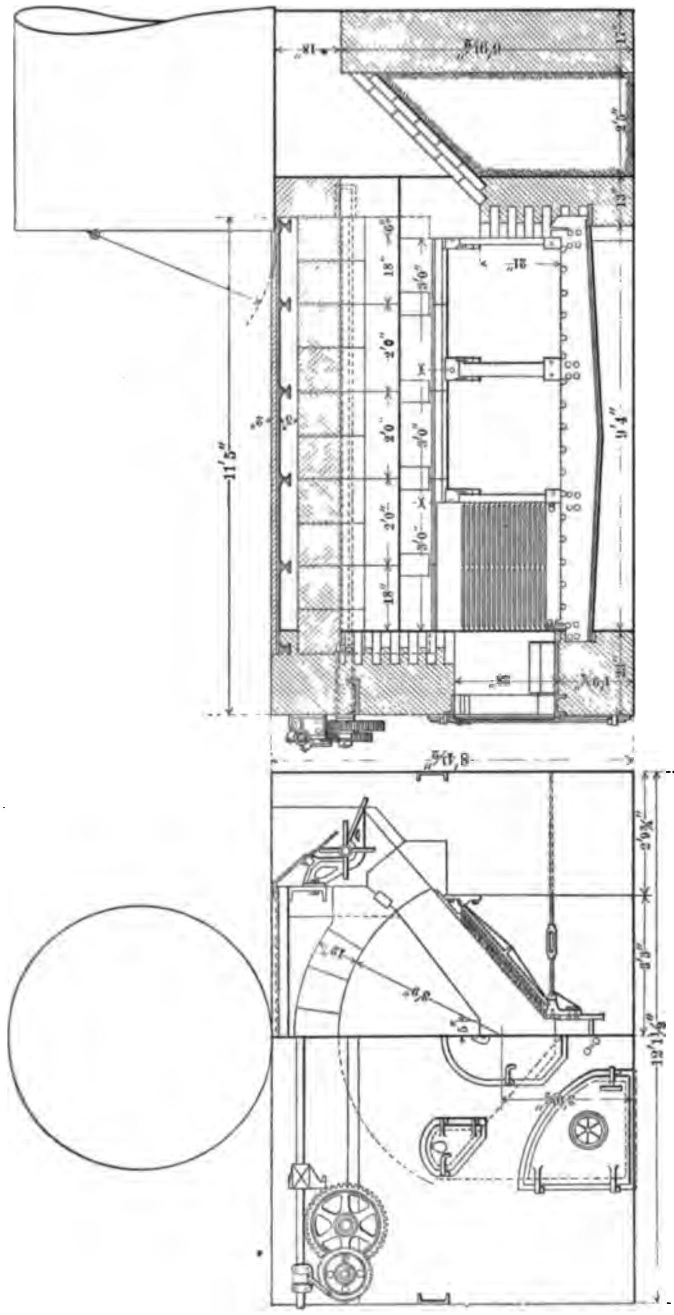


FIG. 2 END AND SIDE SECTIONAL ELEVATION OF THE MYERS TAN-BARK FURNACE

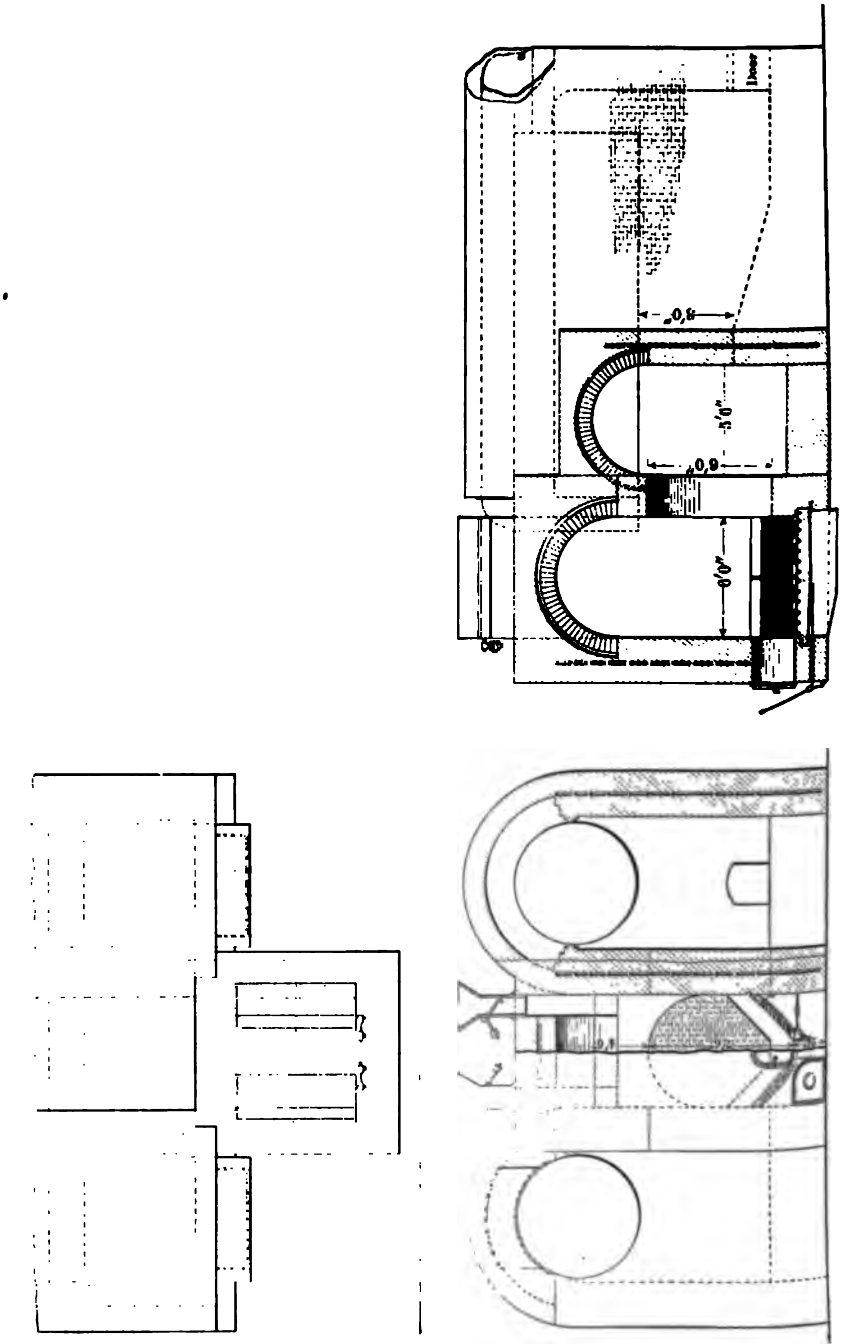


Fig. 3 Partial Plan, Side and End Elevations of the Myers Furnace for Burning Bagasse

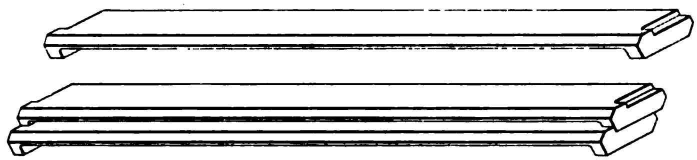


FIG. 4 DETAIL OF THE GRATE BARS OF THE MYERS FURNACE

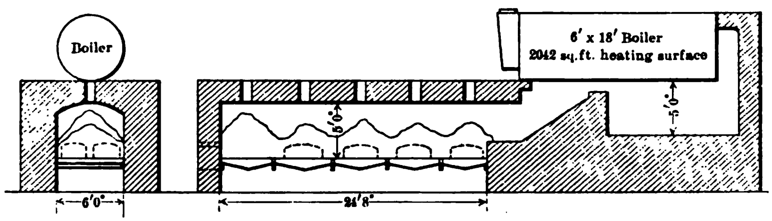


FIG. 5 THE EARLY HOYT FURNACE FOR BURNING TAN BARK

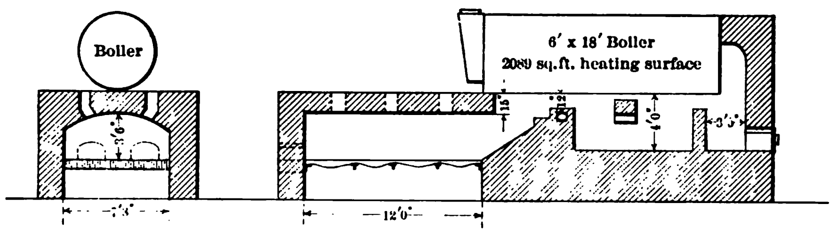


FIG. 6 A TAN FURNACE WITH SIX FEED HOLES

THE SETTING HAD AIR ADMISSION IN THE BRIDGE WALL AND A BAFFLE ARCH IN THE COMBUSTION CHAMBER. VERY GOOD RESULTS WERE OBTAINED.

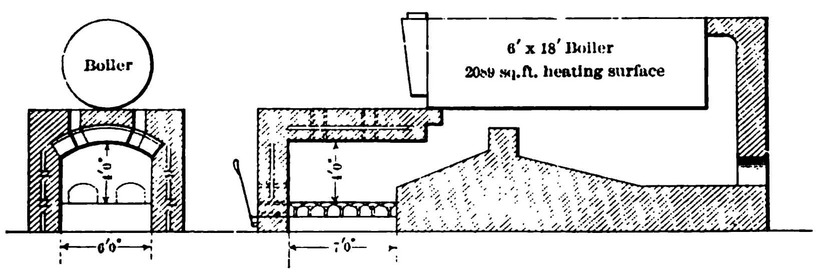


FIG. 7 A FURNACE WITH SHAKING GRATES FOR BURNING A COAL AND TAN MIXTURE

AIR SPACES OVER FIRE ARCH AND IN WALLS OF FURNACE AND BOILER WALLS. DISTANCE FROM GRATE TO TOP OF ARCH INSIDE SHOULD NOT BE LESS THAN 4 FT.

on the grate surface. Fig. 6 shows a more modern type of burner designed to give a more even distribution of the fuel on the grates.

78 Fig. 7 shows a more up-to-date furnace designed for the hand

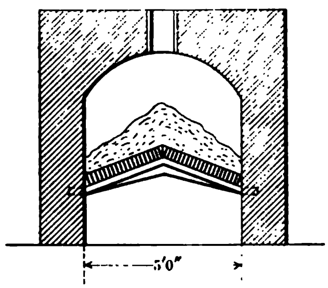


Fig. 8 Cross Section of Furnace with Hump-Back Grates and Bearing Bar

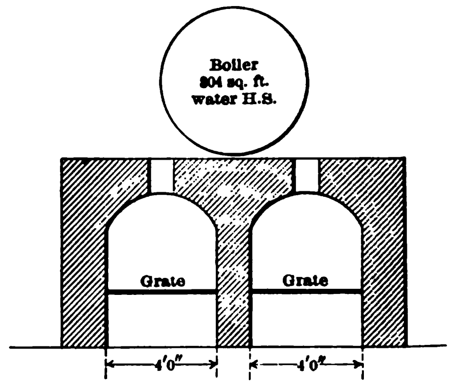


Fig. 9 Cross Section of a Double-Arch Tan Furnace of the Thompson Type on which a Test was Run

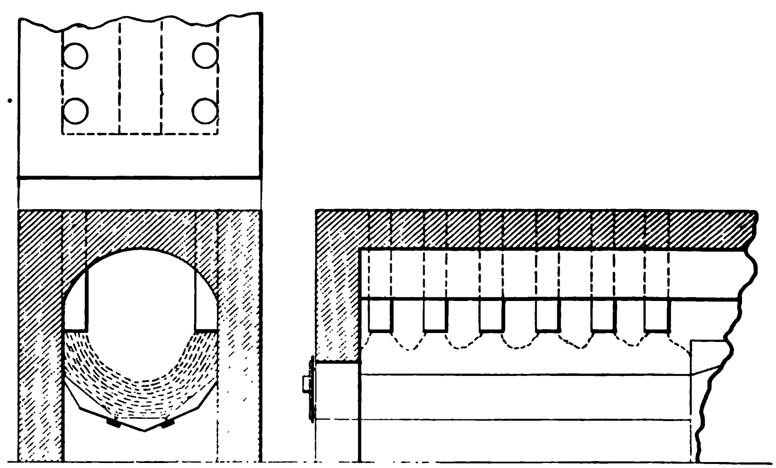


Fig. 10 Bush Tan Furnace with Multi-Tube Feed

firing of a mixture of coal and tan, the coal being mixed with the tan before entering the furnace, which is supplied with shaking or shaking and dumping grates. When coal is mixed with tan in any consider-

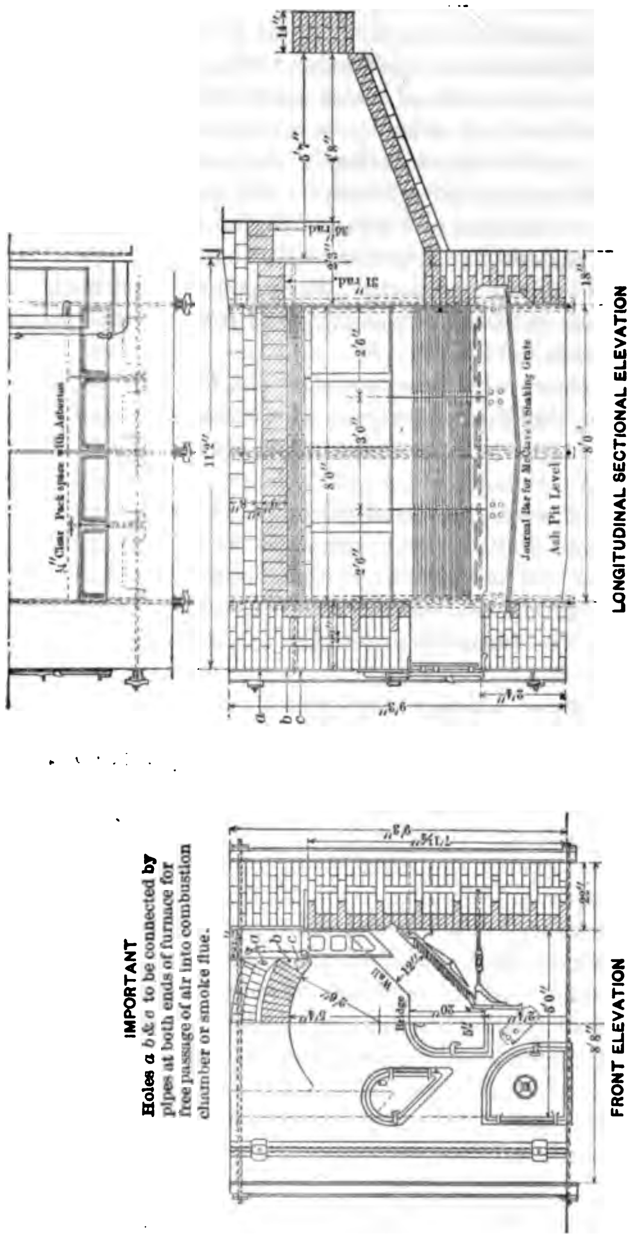


Fig. 11 A Myers Furnace with Adjustable Gravity Feed. Adapted for Burning Tan Bark or Sawdust

able proportion, more air is required for combustion, the best air spacing in the grate bar being found to be $\frac{3}{8}$ in. The percentage of draft area for this purpose should be about 40 to 50 per cent, depending upon how large a percentage of coal is used with the tan.

79 Fig. 8 shows what is known as a hump-back grate, which has been installed in different tanneries for the purpose of increasing the consumption of fuel in a given furnace. For instance, in a plant that had trouble in consuming all its tan bark, the writer merely took out the grate bars and put in a ridge bar as shown and converted the grate surface into the hump-back form. The result was that the consumption of tan bark per furnace was increased from 12 tons per day on the dry bark basis to 15 tons.

80 Fig. 12 shows what was known as the Thompson type of tan furnace. The MacMurray furnace, with a convex grate surface and feed pipes, is a type quite a number of which the writer has seen in operation.

81 Fig. 10 is another form of tan furnace which gave good results in a plant in the South. The hump-back form of grate is reversed something like that used in the writer's stoker furnace, except that the tan is fed through a number of feed holes along the upper edges of these grates. This furnace was designed by the foreman in a Southern tannery.

82 Fig. 11 shows a design of the writer's for an adjustable gravity-feed furnace for burning tan or sawdust. The feed chutes are rectangular in section and contain adjustable chutes to regulate the depth of tan on the grates for any condition of draft, etc.

DISCUSSION

Albert A. Cary. The furnace described by Mr. Myers consiste of an extension in front of the regular boiler setting, with a numbes of circular stoke holes, or openings through the top arch, over thr grate. No little trouble has been experienced with this construction, due to the destruction of the lower end of these circular fire-brick tubes through which the fuel is charged to the furnace.

2 If these stoke holes were always completely filled with fuel, so as to prevent inrushes of air, this destructive effect could be materially checked. However, as the method of charging fuel by hand is an intermittent one, the upper end of the cone of spent tan bark, soon after charging, drops below the level of the top of the arch, the inrushing air meets the hot furnace gases at these points and intense

combustion results. For this reason, and due to the fact that when the excessive moisture in the fuel rises as a vapor against the arch rapidly abstracting its heat (to become superheated steam), the fire brick cracks and disintegrates, finally resulting in a chipping off of the brick-work of the reverberatory arch around the lower end of the stoke holes. Repairs are therefore frequently necessary.

3 A continuous automatic feeding device, which would keep these stoke-holes constantly filled with the moist fuel, would undoubtedly do much to relieve this trouble by preventing an excessive infiltration of air at frequent intervals of time. Mr. K. McMurray of New York, has devised a very ingenous method for overcoming this trouble in hand-stoked furnaces. Fig. 1 herewith shows both front and side sectional elevations of this furnace.

4 In the stoke hole is fitted a circular lining of cast iron which does not extend to the level of the inside of the arch. The lining is finished with a shoulder which diminishes the diameter of the opening by about two inches. A tube or open thimble drops into this frame, being held by a rim cast around its upper end. The lower end of the thimble extends about a foot into the furnace.

5 The fuel charged into the stoke hole falls through the thimble, and forms a cone-shaped pile below it on the grates. When the stoke hole becomes uncovered, the in-rushing air causes the intense combustion to take place, not on a level with the brick-work, but at a level below the thimble, and the life of the fire-brick arch at the stoke-hole openings is thus greatly prolonged. The ends of the cast-iron thimbles burn off gradually, but they cost very little, and may easily be pulled out and new ones inserted in their place.

6 Another trouble met with in this type of furnace is the rapid burning away of the fuel next to the side walls and the consequent large infiltration of air from the ash pit. This trouble has been largely overcome by reducing the width of the furnace about a foot at the grate level, as shown in the front sectional elevation. The ledges formed on either side of the lower part of the furnace support the cone of charged fuel on each side, thus keeping the grate effectually covered with fuel.

7 In this construction it will also be seen that instead of using a flat grate, the grate bars are curved so that the grate surface is higher at the center of the furnace than at the sides. This design decreases the thickness of the fuel bed under the stoke holes and causes a thickening of the fuel bed at the sides of the furnace.

8 Since water can be evaporated in the furnace itself only at

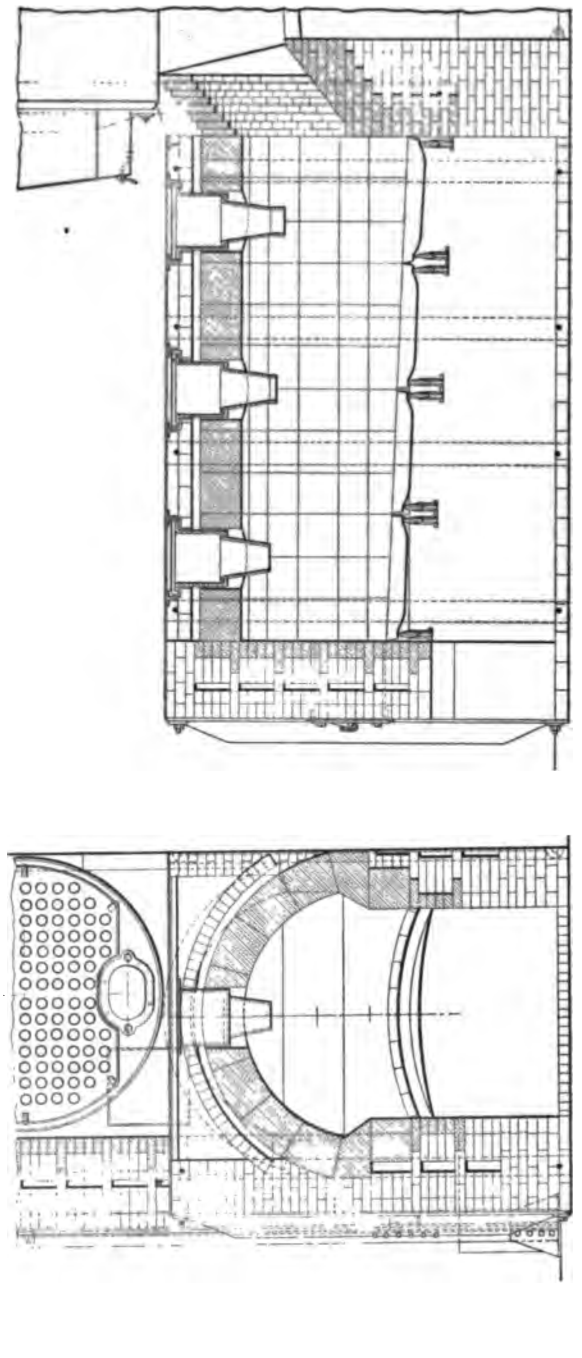


Fig. 1 The McMurray Furnace for Burning Spent Tan Bark

a great loss, every practicable facility should be utilized for depriving the wet fuel of its moisture. Mr. Myers has mentioned the comparatively small gain from pressing the moisture out of the spent tan. I have used special rolls for extracting the water from moist fuels, with a desirable gain resulting. These rolls are of cast iron and run in pairs, one roll being about 12 in. in diameter, the other about 14 in. and both held together by heavy springs. As both rolls are revolved at the same number of revolutions per minute their surface speeds are necessarily different. The faces of the rolls are roughened by having a shallow checker work pattern cast upon them The fuel is fed to the rolls continuously, and due to the tearing or macerating action between the faces of the rolls more than double the amount of water is thus worked out, as compared with the press results given in Mr. Myers' paper.

9 In one case, where the chimney was located some distance from the boilers, a wrought-iron rectangular flue was used to connect them, a shallow iron trough being formed on the surface of the flue by having the edges on the two vertical sides continued above the level of the top. The other three sides of the flue were covered in the usual way. The moist fuel was fed upon the chimney end of the flue and was drawn by a conveyor towards the boiler and over its top, whence it was delivered on top of the extension furnace. A small evaporation of moisture took place, sufficient to make this device desirable. The heat from the top of the boiler and the extension furnace may also be used in this way. The waste heat from the boiler may also be used to pre-heat the air delivered to the ash pit. I know of no condition where pre-heated air can be used to better advantage than with moist fuels.

10 Mr. Myers has spoken of the advantage of the high furnace over the low furnace. My experience thoroughly endorses this. When the moist fuel is charged into the hot furnace, a cloud of steam is evolved, which when crowded down upon the burning fuel in a low furnace hinders combustion. A sufficient amount of steam would eventually extinguish the fire.

11 In addition to the effect of moisture described in Par. 7 and Par. 8, the large space occupied by the steam in the combustion chamber interferes with the combination of oxygen and the combustible gases evolved from the fuel.

12 In the flue-gas analysis obtained with moist fuels, of course the water in the gases condenses and is not accounted for in the analysis given.

William Kent. I consider this the most important paper on the subject of tan bark as a boiler fuel which has appeared in over thirty years. The only other paper that I know of is one by Professor Thurston published in 1874 in the Journal of the Franklin Institute. He made some boiler tests on tan bark for fuel, using two different styles of furnace, some of his results being better than those given by Mr. Myers. I think that still better results are yet to be obtained from the use of tan bark as a fuel, by compressing out as much as possible of the moisture and using the waste heat of gases to dry the bark before it is put in the furnace. For burning the bark we must have a large fire-brick combustion chamber and give plenty of time to the burning of the gases, and then we will get as near the theoretically possible economy as can be expected.

2 The principal cause of poor economy in the burning of tan bark, besides the difficulty of securing good combustion in the furnace, is the amount of heat that is carried away in the shape of superheated steam in the chimney gases. If the bark, after being partly dried by compression, were further dried in a rotary drier by the waste heat from the chimney gases, there would be a very important gain in economy.

3 I have made a calculation showing the theoretical results that may be obtained in burning tan bark of different degrees of moisture under certain assumed conditions, the results of which are given herewith. The dry bark is assumed to have the following composition: $C = 0.50$; $H = 0.06$; $O = 0.40$; N and ash $= 0.04$. Substituting in Dulong's formula $14{,}600\ C + 62{,}000\left(H - \frac{O}{8}\right)$, its heating value is 7920 B.t.u per lb. Bark containing 20 per cent moisture would have a heating value of $0.80 \times 7920 = 6336$ B.t.u.

4 Assuming the chimney gases to escape at 600 deg., the heat required to evaporate the water from 62 deg. and to superheat the steam to 600 would be $(212-62) + 970 + 0.48\ (600-212) = 1306$, or for 20 per cent moisture, 261 B.t.u. per pound of tan.

5 The 0.06 lb. of H in a pound of dry tan will unite with $0.06 \times 8 = 0.48$ O, making 0.54 lb. H_2O, which escapes as superheated steam carrying away $0.54 \times 1306 = 705$ B.t.u. for each pound of dry tan or $0.80 \times 705 = 564$ B.tu. for tan with 20 per cent moisture.

6 Assuming 25 lb. of air to be required per pound of C + H in the fuel or $25 \times 0.56 = 14$ lb. of dry tan, the heat carried away by this air heated to 600 deg. is $0.24 \times 14 \times (600-62) = 1808$ B.t.u. per pound of dry tan or 1446 B.t.u. for tan with 20 per cent moisture. Using the figures thus found the following Table 1 is constructed.

TABLE 1 THEORETICAL EFFICIENCY OF TAN BARK FUEL

Moisture	B.t.u. per lb. wet tan	Losses of heat due to			Sum of losses	Net heat value B.t.u.	Efficiency per cent	Lb. Evap. per lb. wet tan
		Moisture	H in fuel	Heating air				
0.20	6336	261	564	1446	2271	4065	64.2	4 19
0.30	5544	392	493	1266	2151	3393	61.2	3.50
0.40	4752	522	423	1085	2030	2772	57.3	2.81
0.50	3960	653	352	904	1909	2051	51.8	2.11
0.60	3168	784	282	723	1789	1379	43.5	1.42
0.70	2376	914	211	542	1667	709	29 8	0.73
0.80	1584	1045	141	362	1548	36	2.5	0.03

7 Suppose that tan with 60 per cent moisture were dried to 20 per cent before being put into the furnace, using for this purpose the waste heat of the chimney gases, we would then have 0.40 dry tan + 0.60 moisture dried to 0.40 dry tan + 0.10 moisture, 0.50 water being removed. Suppose the moisture and the waste gases left the drying chamber at 300 degrees. Each pound of water dried out would take $(212 - 62) + 970 + 0.48\ (300-212) = 1162$ B.t.u. and 0.5 lb. would take 581 B.t.u. The H in the 0.40 lb. of dry tan would make 0.216 H_2O, which would take away $0.216 \times 1162 = 251$ B.t.u. Heating the air would take $0.40 \times 14 \times 0.24 \times (300 - 62) = 320$ B.t.u. The sum of these is 1152, which subtracted from 3168, the total heating value of tan with 60 per cent moisture, leaves a net value of 2016 instead of 1379, the figure given in the table. The efficiency would be $2016 \div 3168 = 63.6$ per cent, instead of 43.5 per cent, and the evaporation from and at 212 deg. $2016 \div 970 = 2.08$ lb. instead o 1.42 lb.

PROF. F. R. HUTTON. In 1874, the late Robert H. Thurston presented a paper on The Efficiency of Furnaces Burning Wet Fuel, before the American Society of Civil Engineers.[1] At that date few engineers were paying attention to fuel economy, and there was little widespread knowledge as to the details by which it would be obtained. There was of course no formulated code for boiler testing. This paper introduced the writer at that time to the problems of boiler testing, and recorded for the first time for him the formulæ for the barrel type of steam calorimeter.

2 The two furnaces examined were designed to meet the same requirements as are assumed in Mr. Myers' paper; but the press

[1]Trans. Am. Soc. C. E., No. 102, Vol. III, 1874, p. 290.

which may be expected to expel a proportion of the water absorbed from the leaching process was not in use, and no data were given as to the proportion between the dry bark ground at the mill and the weight of wet leached fuel delivered at the fire room. The Dutch oven type of furnace was in use, consisting of a fire-brick chamber covered with a reverberatory arched roof. The fuel was fed in at the top of the oven through two holes in the length of the grate. The grate was of fire brick moulded to obtain a semi-cylindrical surface to the upper and lower surface of each bar unit, the concave side being downward towards the ash pit. A large proportion of the finer tan lumps was expected to fall through the holes in the arched bars of the grate and complete their combustion there on the ash-pit floor.

3 But it is very plain from the results of the tests that the furnaces were on very much the same plane of efficiency as those reported by Mr. Myers, since the respective results of evaporation from and at 212 deg. per lb. of combustible were for the Crockett furnace 4.41 lb., for the Thompson 5.68 lb. and for the Myers' furnace 5.43, 4.71 and 4.54 lb., if equal accuracy be assumed in the old tests as compared with the new. This is open to doubt, however, as certain figures were assumed or deducted from other experiments and were criticized in the discussion of the results.

4 The present paper is especially interesting to the writer, because it represents the work of a furnace designed by Mr. Myers which seems to incorporate some eminently sound principles. I think all will agree that the three cardinal principles for the complete and smokeless combustion of a reluctant fuel involve the following:

a Time enough for access of oxygen in the air to the carbon gas from the fuel.
b Temperature enough for the rapid and complete chemical union of this oxygen with carbon and hydrogen.
c Room enough for each atom of fuel gas to meet the oxygen atoms with which it is to unite.

The practical attainment of these results is made more difficult when the fuel is wet and in small particles of light weight.

5 We have the conflicting conditions of a hot fire and a slow rate of combustion to combine with an intensity of draft which shall not be high. Mr. Myers does this by using the step-grate idea, so as to admit the necessary air horizontally between the overlapping bars, whereby the dropping of fine fuel into the ash-pit is prevented: but in addition and as a special excellence of the design, the grate

is made to consist of two sections facing each other with their planes parallel to the long axis of the Dutch oven and the shell of the boiler. They are, as it were, upon the inclines of a truncated capital letter V.

6 The bark is fed by a measuring stoker cylinder, which drops a determined volume upon the whole length of the upper bar at each partial revolution, and this fall of new material displaces downward some of what has been drying and growing ready to ignite from the previous charges. At the bottom of the truncated V is a dumping grate from which the residue of ash may be released at intervals.

7 The consequence of the inclination of the two grate sections, with a horizontal inflow of the air, seems to be the same as is produced in a successful form of burner for acetylene gas. The two currents of gas and draft appear to meet in the center or in the axis of the V and an intense combustion takes place there, the heat of which reverberates downward from the arch of the oven, raises the temperature of the upper layers of fuel, and stimulates the rate of the union of combustile with oxygen. Such a furnace of course is not subject to the alternations of the "famine and feast" conditions when excess of wet fuel deadens the fire and causes a smoky and slow combustion, alternating with high heat and good flame and followed in turn burned-out spots in the fire until fresh charge comes in through the holes.

8 The system is also most effective for bagasse, the wet juicy fibre of the sugar cane after passing through the pressing rolls. This is more difficult to stoke mechanically than the comminuted bark, but the requirements for its successful combustion are very satisfactorily met. Sawdust and scrap from wood-working shops are also burned in furnaces of this design with less danger from sparks at the stack.

9 Referring to the summaries by the author, it should be plain that the grate surface must be reduced if the tan is press-treated (Par. 56) to remove moisture. Bulk for bulk, there are more heat units per unit of volume or of weight after a volume or weight of water has been expelled than there were when the tan was saturated and not pressed. If the fire is hot enough to dissociate the oxygen and hydrogen which compose the water, the heat for such dissociation is drawn from somewhere; doubtless from the flaming gases, where the process takes place, and of course they are cooled, and perhaps killed.

9 If such oxygen and hydrogen recombine, nothing is lost, and perhaps a mechanic-thermal advantage is reaped because the hydrogen flame is longer than the carbon flame. If for any reason such

dissociated hydrogen does not get a chance to recombine from lack of temperature or time or room, there is a loss. Mr. Myers' results should serve to check the claims still advanced at intervals, that the combustion of steam-gas is a source of great possible economy.

The Author. Mr. Cary in his discussion has described the McMurray tan furnace, one of many different types and designs now in use. All the ordinary forms of tan furnaces feed the fuel through holes in the top or arch over the grate. The number and arrangement of these feed holes vary in the different designs, but they all form a bed of fuel composed of cones of tan. For this reason they are all subject to the objection made by Mr. Cary, i. e., that the fuel burns away most rapidly around the bottom of these cones where the depth of fuel is least. The central parts of the cones offer great resistance to the draft so that active combustion takes place on only a small percentage of the entire grate surface. This necessitates large grate surfaces and large furnaces with attendant radiation losses.

2 Another objection to the cone method of feeding the fuel, especially when only a single row of feed holes is employed, is that the fire is actually divided into a number of small fires around the bottom of the cones. This multiplicity of small fires, separated by heaps of wet tan of low temperature, results in lowering the furnace temperature and in retarding combustion.

3 In furnaces of this type with careless firing the writer has seen fully one-half of the grate surface doing no work at all in the way of any active combustion. These ill effects are best eliminated by very frequent feeding of the tan in small amounts, so that the percentage of wet tan in the furnace at any time is very small compared to the actively burning mass. High furnace temperature is thus maintained, more grate surface is active and the rate of combustion per square foot is greatly increased. The result is less grate surface required, smaller radiation loss due to smaller furnaces and greater ease in handling and cleaning the fires.

4 In general, the greater the number of feed holes the higher will be the rate of combustion and the smaller the furnace required. Rapid firing in small amounts to equal the rate of combustion in the furnace is productive of best efficiency with any of the usual types of tan furnaces.

5 Tan presses of different makes, but all of the same type described by Mr. Cary, have been experimented with by the writer. It was found that with careful adjustment and attendance the presses

would equal the performance quoted by Mr. Cary but that under tannery conditions of indifferent attendance and unskilled labor the presses do not maintain their efficiency.

6 The interference of the steam—gas evolved from the fuel with the union of the combustible gases with the oxygen must be overcome by providing large combustion space, preferably over the fuel bed, by special baffles or by special draft action as in the writer's design of automatic furnace shown in Figs. 2 and 3 and referred to by Professor Hutton.

7 The chemical composition of tan is assumed by Professor Kent to be practically the same as that given from an actual analysis in the author's paper. The heating value according to Dulong's formula is 7920 B.t.u. per lb., whereas the results of a large number of tests in a bomb calorimeter by Dr. Sherman, shows the heating value of a pound of dry hemlock tan to be close to an average of 9500 B.t.u.

8 I have carefully read the record of tests on tan burning furnaces made by Prof. R. H. Thurston, and presented in a paper before the Franklin Institute in 1874. Professor Kent states that some of the results there given are higher than those determined in recent practice by the writer. The two evaporative results by Thurston are given as 4.24 lb. equivalent evaporation from and at 212 deg. in the boiler per pound of combustible for the Thompson furnace, and 3.19 lb. for the Crockett furnace. The corresponding figure obtained by the writer in his automatic furnace was 5.55 lb.; that is, over 31 per cent better than Thurston's best result.

9 The writer finds that the evaporations of 5.68 and 4.41 for the Thompson and Crockett furnaces respectively were obtained by Thurston by *adding to the evaporation in the boiler the amount of moisture in the fuel evaporated from and at* 212 *deg.* A similar addition to the writer's evaporation in the boiler of 5.55 lb. would make an evaporation of 7.75 lb. including the moisture in the fuel. The latter figure is therefore the one to be compared to Thurston's result of 5.68 lb. On the same basis of caloulation the economic result of present best practice is over 36 per cent higher than the best result recorded by Thurston.

10 Moreover the highest result in the Thurston test was obtained by a rough volumetric approximation of the weight of the fuel used. It was not weighed to the fireman as in all the author's tests. Furthermore, both the weight and temperature of the feed water were merely approximated and assumed to be correct in the Thompson

furnace test; whereas these values in the author's tests were all observed and recorded in a most accurate and systematic manner.

11 The accuracy and reliability of these old tests is very much to be doubted, as Professor Hutton suggests. But even if taken at their full values it is seen that the results of present practice have exceeded the old results by over 30 per cent.

12 Actually the present results are probably even higher than this, from a comparative standpoint, for the reason that in the old days of tanning, the moisture in the tan was less than in present practice. This consideration would have given the Thurston tests a decided advantage in the shape of a greater available heat value of the fuel. Thurston gives the moisture contents of the fuel as fired as 55 and 59 per cent, whereas the moisture in the writer's automatic furnace test was 65.3 per cent.

13 This increase in moisture is due to radical changes in the process of leaching the bark. Where formerly the bark was treated with cold, or nearly cold, water it now is leached at temperatures as near the boiling point as possible, and is subjected to the leaching process two or three times as long as in the former methods. This is on account of the high price of bark nowadays, which makes it pay to leach out as much of the tannin as is practically possible. Some tanneries to-day leach their bark so thoroughly that only ½ to 1 per cent of tannin remains in the spent tan.

14 The author desires to add that all results and data given in his paper are results of actual tests made under working conditions. No assumptions or theoretical calculations are involved in the conclusions. The feed water was in every case measured by means of two tanks or barrels set above a reservoir from which a separate feed pump supplied the boiler. Feed connections were so separated that it was physically impossible to pump the water elsewhere than in the boiler being tested. All connections involving a chance for leakage were blanked off. Valves were never assumed to be tight but were proved so during the entire test by means of an open-T arrangement which would show any leakage.

15 The temperature of water entering as well as leaving the measuring barrels was taken at frequent regular intervals. The barrels were calibrated by weighing when filled to their overflow pipes with water at the temperature which the feed water had averaged during the test.

16 The fuel was in every case weighed in equal amounts to the fireman. A sample corresponding to each 200 lb. was taken, kept

in closed receptacles and at the end of the test was mixed, and quartered down to a quart or two quart sample which was sent in sealed jars to Dr. Sherman for determination of B.t.u. and moisture. All readings and observations were obtained with like regard for accuracy of results.

17 In Par. 3 Professor Hutton also compares the best results obtained by the writer with those of Professor Thurston; but as before pointed out, the results are on a very different basis and are not comparable, unless the moisture in the fuel is also added to the equivalent evaporation obtained in the boiler. If this is done the following table gives a correct comparison:

POUNDS EQUIVALENT EVAPORATION FROM AND AT 212 DEG.

Including Water in Fuel		Excluding Water in Fuel	
Thurston Tests	*Myers Test*	*Thurston Tests*	*Myers Tests*
5.68 for Thompson furnace	7.75 for Myers furnace	4.24 Thompson furnace	5.55 for Myers furnace
4.41 for Crockett furnace	6.63 for present ordinary furnace	3.19 for Crockett furnace	4.30 for present ordinary furnace

The table shows that when compared on the same basis of efficiency the art of tan burning has been greatly improved over the old methods, both with improved and ordinary furnaces.

18 Thermal efficiency is of course the safest and most accurate basis of comparing results of various boiler and furnace settings, and the highest result yet obtained in a reliable witnessed test in tan burning was 71.1 per cent. This is based on available heat in the fuel as fired after allowance is made for evaporating the moisture in the fuel. This test, which was made on the automatically stoked furnace before referred to, showed an efficiency of boiler and furnace of 54.4 per cent, based on the total heat of the fuel.

No. 1259

COOLING TOWERS FOR STEAM AND GAS-POWER PLANTS

WITH PARTICULAR REFERENCE TO THE POSSIBILITIES OF THE NATURAL-DRAFT AND AUXILIARY-DRAFT TYPE

By J. R. Bibbins, New York

Member of the Society

The object of this paper is to bring to the attention of the members of the Society a subject which has received relatively little attention in the past, but which the author believes merits the careful study of all engineers interested in future power-plant development. The cooling tower has been looked upon as a makeshift, and its use has been correspondingly restricted. This, however, is largely due to the extremely limited information of an exact or technical nature, available to the general public, relative to depreciation and performance under unfavorable weather conditions. And, further, it is the author's belief that the present high prices[1] constitute the greatest obstacle to the more widespread adoption of the cooling tower in both turbine and gas-power plants.

2 Believing that interest may be aroused in this subject by a more widespread dissemination of engineering data, the author will present for discussion a type of tower with which some personal experience has been acquired, and suggest a type of combined fan and natural draft suited to most efficient running on peak as well as light loads. It is not the intention to discredit the cooling tower in its present forms but rather to bring about a more general recognition of its inherent advantages.

[1] Some recent quotations from a number of builders of forced-draft towers suitable for a load of several thousand kilowatts, (*not including* the motor or engine for driving the fan), ranged between $4.80 and $6.93 per kw., as much as the entire condensing equipment.

Presented at the Annual Meeting, New York, (December 1909), of The American Society of Mechanical Engineers.

PRESENT FIELD

3 There is a continual demand for cooling towers from inland power stations where the condensing water supply is costly or restricted. Turbine-driven plants, as a rule, operate with higher vacuum than engine-driven, with the result that the perform-

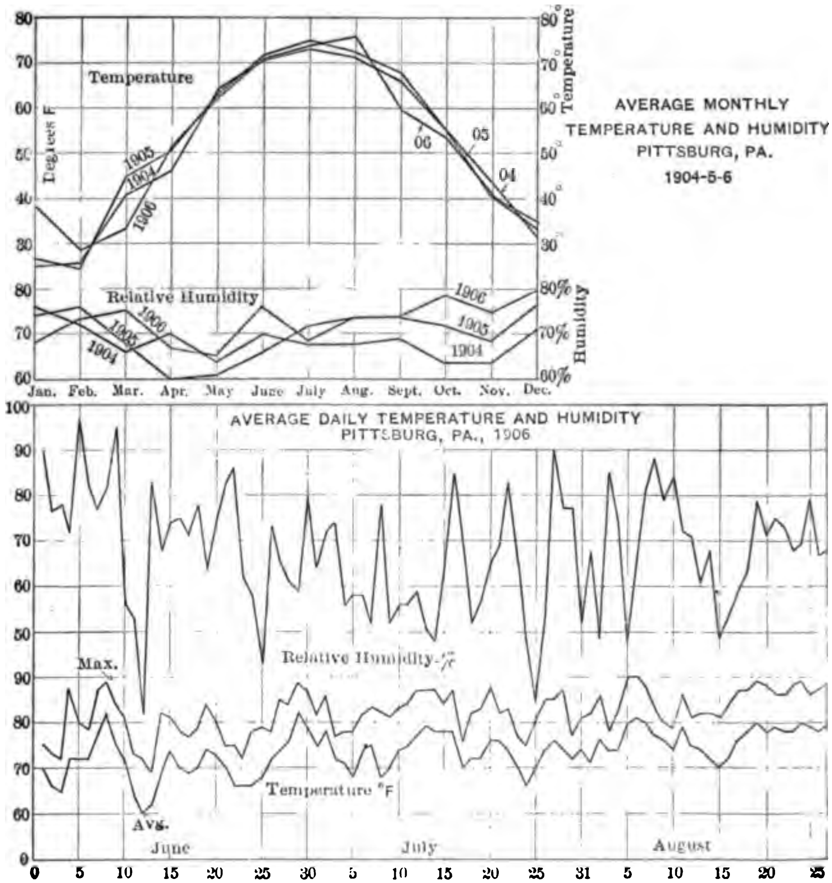


FIG. 1 CYCLES OF AVERAGE TEMPERATURE AND HUMIDITY AT PITTSBURG, PA.

ance demanded of the tower must be proportionately better. In the past, cooling towers have generally been associated with badly run plants and low vacuum. This, however, is clearly a question of design and adaptation and not an inherent fault. To be sure, service requirements are not easily met (see Fig. 1). Not infrequently atmospheric temperatures of 90 deg. to 100 deg. fahr. are encountered, with cooling water at 80 deg. to 90 deg. fahr. and humidity above 85 per cent saturation. Yet the auxiliary plant must be moderate in bulk and the power consumption low. Furthermore, it must be capable of overload capacity to tide over daily peak loads and periods of

TABLE 1 WEATHER CONDITIONS, PITTSBURG, PA.

DATE	AVERAGE TEMPERATURE DEG. FAHR.	AVERAGE HUMIDITY PER CENT
1904	52.8	73
1905	52.1	68
1906	52.0	70
Average..............	52.3	70.3

MAXIMUM TEMPERATURE AND HUMIDITY RANGES, SUMMER OF 1906

TEMPERATURE ABOVE	DAYS IN MONTH			AVERAGE FOR MONTH
	JUNE	JULY	AUGUST	
90 deg.	..	..	2	0.6
85	6	9	14	10.5
80	8	14	10	20.5
75	10	6	4	27.7
70	5	2	1	23.7
Below	..	..	..	..

HUMIDITY ABOVE	DAYS IN MONTH			AVERAGE FOR MONTH
	JUNE	JULY	AUGUST	
90 %	3	1	3	2.3
80	14	10	15	13.0
70	8	11	11	10.0
60	3	8	2	4.3
50	2	1	..	1.0
Below	..	..	..	..

Data from Pittsburg Weather Bureau.

excessively hot and humid weather, all with small investment cost.

4 Curiously enough, there is an active demand for cooling towers in the South, e. g., Florida, where the atmospheric conditions are the most unsuitable; also in the Western mining and coast regions. For-

tunately, low humidity prevails here as a general rule (e. g., Colorado Springs ranges around 50 per cent).

5 In gas-power work, the demand for cooling towers is especially pressing. The large quantity of water required for engine jackets and for gas cooling and washing entails a heavy expense if water is scarce and costly. Gas-engine discharge water, being quite pure, should not be wasted, but cooled and returned to the plant. Even with deep well pumps supplying sufficient water for cooling, a station

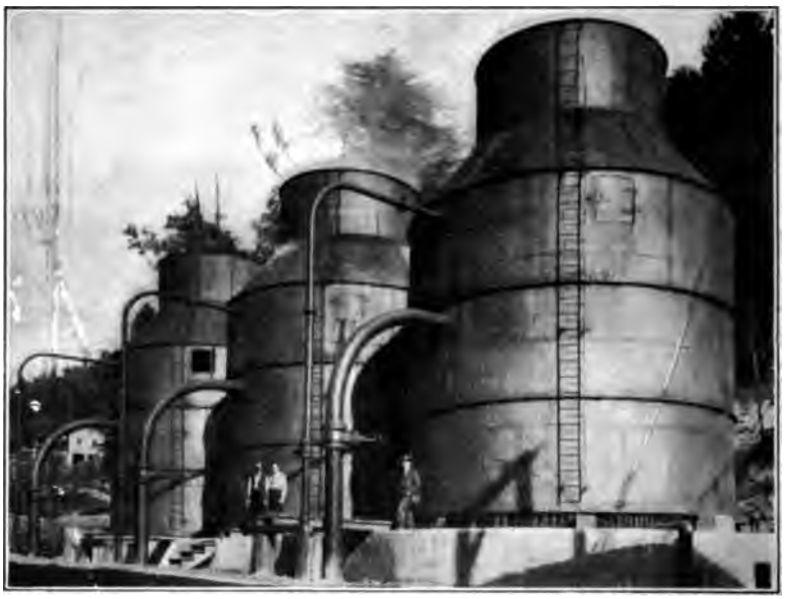


FIG. 2 4500-H.P NATURAL-FORCED-DRAFT COOLING TOWER AT GARY' W. VA., FOR A LOW PRESSURE AND HIGH-PRESSURE TURBINE INSTALLATION

THESE TOWERS HAVE AUXILIARY FANS IN THE STACK DRIVEN BY PELTON WHEELS AND SMALL TURBINE-DRIVEN ROTARY PUMPS LOCATED IN THE POWER STATION, AND OPERATE WITH NATURAL DRAFT WHEN NOT HEAVILY LOADED

is handicapped by a large expense for auxiliary power consumption. In one instance in an Arizona mining plant, the only water available for cooling was so impure as to make it necessary to install a completely *closed cooling system* for the engine jackets, in which no evaporation took place, simply cooling by conduction. In city light and power plants not fortunate enough to be located on water frontage, cooling towers built upon the roof have been utilized for engine cool-

3 1500-H.P. OPEN-TYPE STEEL TOWER AT SAN LUIS POTOSI, MEXICO, FOR A GAS-POWER CENTRAL STATION

TOWER SERVES THE ENGINE JACKETS, THE OTHER THE PRODUCER SCRUBBER. WIRE-CREEN SIDE CASING IS USED TO REDUCE THE WATER LOST BY WINDAGE. COOLING, 10 TO 0 DEG. HUMIDITY GENERALLY LOW—ABOUT 50 PER CENT.

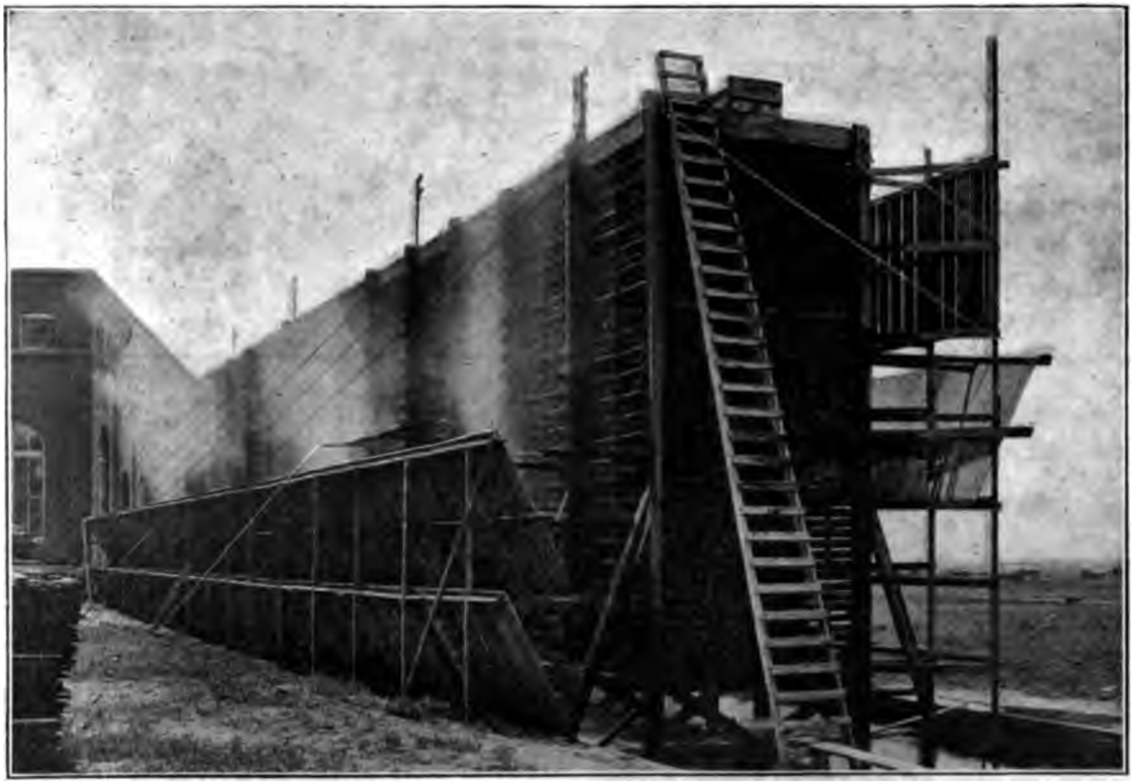


FIG. 4 2900-H.P. OPEN-TYPE WOODEN TOWER AT COLORADO SPRINGS FOR A LOW-PRESSURE TURBINE INSTALLATION WITH A LEBLANC CONDENSER

. . . VAPOR DURING HIGH WINDS. COOLING, 20 TO 30 DEG. HUMIDITY GENERALLY LOW . . . ABOUT 50 PER CENT

Fig. 5 2500-h.p. Open-Type Screen Tower for a Steam Turbine Plant at Visalia, Cal.

ing. The expense of buying city water for this purpose would otherwise be prohibitive except in large plants.

REPRESENTATIVE INSTALLATIONS

6 As examples of present cooling-tower practice in connection with high-grade power properties, the following may be mentioned:

a At Gary, W. Va. (Fig. 2), three towers 25 ft. in diameter by 18 ft. high serve a plant of both high and low-pressure turbines

FIG. 6 4000-KW. NATURAL-DRAFT TOWER AT BUTTE, MONT., STEAM TURBINE STATION. WOOD CONSTRUCTION.

recently installed in connection with a non-condensing engine plant furnishing the mines with light and power. A unique feature is the induced-draft fan located in the stack and driven by a Pelton water motor, which is served in turn by a small turbine-driven centrifugal pump in the power house. This equipment will be referred to later in connection with the combined natural-forced-draft type.

b The 13th & Mt. Vernon Street station of the Philadelphia Rapid Transit Company is another low-pressure turbine plant employing forced-draft cooling towers.

c At the central station at San Luis Potosi, in the Mexican highlands (Fig. 3), separate cooling towers serve engine

TABLE 2 WEATHER CONDITIONS, BUTTE, MONT.

AVERAGE TEMPERATURE AND HUMIDITY, 1894–1904 (WEATHER BUREAU, HELENA, MONT.)

MONTH	TEMPERATURE DEGREES	HUMIDITY PER CENT	MONTH	TEMPERATURE DEGREES	HUMIDITY PER CENT
January	23.7	66.8	July	62.8	45.3
February	23.9	68.6	August....	62.9	43.4
March.......	28.3	60.8	September.	52.0	53.6
April........	40.0	54.3	October...	45.1	54.4
May.........	49.0	54.9	November.	33.7	62.7
June.........	52.0	50.0	December .	26.5	70.5
Av. for year	41.6			58.9	

TEMPERATURE RANGES

BETWEEN DEGREES	DAYS	ABOVE DEGREES	DAYS	PER CENT YEAR	HOURS	PER CENT YEAR
70– 75	25	70	99	27	276	3.15
75– 80	24	75	74	20	196	2.22
80– 85	28	80	50	13.7	158	1.8
85– 90	19	85	22	0.6	51	.58
90– 95	2	90	3	0.82	9	.097
95–100	1	95	1	0.28	1	.01
		Total	249	62.40	691	7.9

Data from M. H. Gerry, Jr.

and producer systems, the make-up water being furnished from deep well pumps. This plant has been in operation since 1904.

d Western plants of inexpensive construction are the Mt. Whitney Power Company (Fig. 5), and the Colorado Springs Light & Power Company (Fig. 4), both turbine plants.

e Perhaps the best example of the adaptability of cooling towers is an equipment designed and built by the Helena

Power Transmission Company for its auxiliary turbine station at Butte, Mont. (Fig. 6).

7 This latter tower was made the subject of an exhaustive preliminary study and a subsequent test by the company's engineering organization, and through the courtesy of M. H. Gerry, Jr., chief engineer and general manager, the writer has been able to place the complete report at the disposal of the Society for future consideration. The tower (shown in Fig. 6) serves a turbine plant of 4000-kw. capacity at 28-in. vacuum. The designers state that after two years' experience the results coincide closely with the theoretical deductions made before its construction.

SPECIAL PHASES OF COOLING TOWER OPERATION

8 Two important factors fortunately contribute to the effective operation of a cooling tower:

a One factor is the well-known characteristic of a natural-draft tower considered as a "chimney"—increase in capacity with increase in temperature head (see Fig. 12).

9 In steam work, especially with high vacuum, the general range of discharge temperatures is relatively low; in gas-engine work, on the other hand, it is high. Pistons are today operated at temperatures of 140 deg. to 160 deg., cylinders from 120 deg. to 150 deg., and occasionally higher. Owing to the small volume of water in the minor circuits, such as valves, packings, etc., these temperatures have little effect upon the average outlet temperature of the engine, which ranges from 115 deg. to 130 deg. in the large engines, and 140 deg. in the smaller sizes and verticals. This would correspond to a very poor vacuum in a steam plant, not more than 24 in. to 26 in.; practically out of the question in turbine work. However, this high temperature results in a high rate of heat dissipation in the tower per unit of cooling surface, with a corresponding reduction in bulk of tower.

b The second factor relates to developments in the efficiency of the steam-condensing plant.

10 The function of a condenser is, primarily, that of a water heater and the measure of its efficiency as a condensing vessel is the difference between the temperature of the exhaust steam and that of the discharge water. A theoretically perfect condenser would heat the outgoing cooling water exactly to the temperature of the incoming steam. But in practice from 10 deg. to 50 deg. difference exists,

depending upon the type of condenser and the volumetric ratio of water to steam. A good surface-condensing plant with dry-air pump should operate at 28-in. vacuum with a temperature difference of 15 deg.; often it is more, and the author has seen 25 deg. to 40 deg. difference in some of the largest stations in the country. A good barometric or centrifugal jet condenser, with dry-air pump, should operate with a temperature-difference of 10 deg. to 15 deg. Although it is possible for this type to operate on less—perhaps 5 deg. to 10 deg.—commercial practice rarely concedes such results.

11 A very recent development in air pumps has made it possible to operate on a still smaller difference (from 2 deg. to 5 deg.) with a

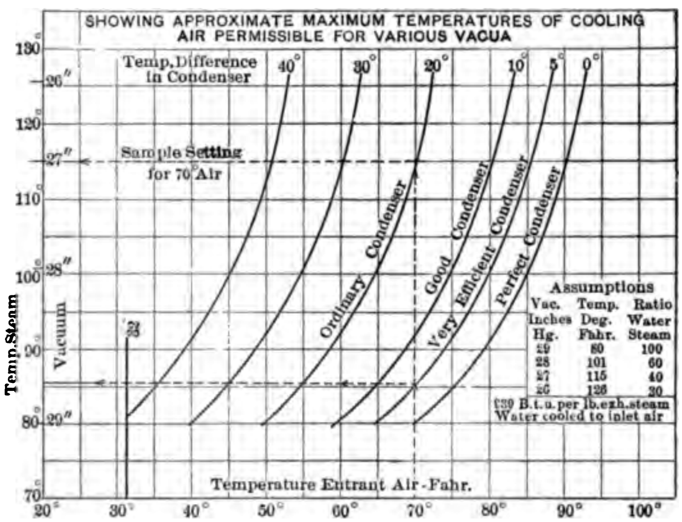


FIG. 7 SHOWING MAXIMUM AIR-INLET TEMPERATURES FOR VARIOUS VACUA

reasonable water ratio, and even to approximate theoretical conditions. All this in the right direction. The smaller this temperature differential, the higher the maximum inlet temperatures permissible for a given set of conditions—both water to condenser and air to tower. The curves in Fig. 7 show this relation in approximate form—vacuum possible with varying condenser and fixed cooling-tower performance. For example, with 28-in. vacuum, 20 deg. differential in the condenser, water ratio 60, and 15 deg. cooling in the tower, the highest possible temperature of outside air would be 65 deg. fahr. With warmer air, the vacuum would necessarily fall.

Under the same conditions, with 10 deg. differential, a maximum air temperature of 76 deg. would be permissible; and with 5 deg., 81 deg. inlet air. It is therefore apparent that the tendency of modern condenser development toward higher efficiencies will materially assist in the successful operation of cooling towers under extremely adverse conditions.

ELEMENTS OF DESIGN

12 The most important elements entering into the design may be considered under the following heads, having special reference to the enclosed type of cooling tower, which for a given floor space has by far the greatest cooling capacity:

a Type of cooling surface.

b Water distribution system.

c Draft and air distribution.

13 The following are a number of essential points that seem to the writer to have a most important bearing upon any type of tower designed for maximum duty and efficiency.

a All tortuous or unduly obstructed passages should be avoided. It is of no advantage to give ample spacing in one part of the tower and contract it in another, unless sufficient stack height is provided to overcome the additional resistance.

b Avoid free falling water. It should be distributed so as to descend clinging to some form of wetted surface.

c Avoid open spaces in the mat work, usually occurring at points where it is difficult to fill in between the frame of the tower. This will "short-circuit" and invariably diminish the effectiveness of the working sections.

d Reduce working section to minimum possible height, adding extra stack if necessary. The power required to elevate the water is important, and the working height of the tower is lost, even in a closed-condenser circulating system.

e Baffles or variable spacing are often necessary to obtain uniform air distribution.

f A settling basin of liberal depth is always advisable in order that entrained air may separate. In all jet-condenser installations, this is extremely important owing to the amount of air returned to the condenser; and even in surface installations, this air will find its way back to the condenser via the feed water: result, impaired vacuum.

g All wooden mat surface is subject to swelling. Means should be taken to insure permanent alignment; otherwise serious reduction in draft area and capacity may be encountered.

h For maximum effectiveness, a cooling surface is required which provides uninterrupted descent of water, in a thin film at all times in intimate contact with ascending air. If any interruption is necessary, the descending sheet should be guided into place to avoid free fall.

PRESENT TYPES

14 The various types of cooling systems now in use naturally group themselves into a few general classes:

a The simple spiral-spray nozzle discharging into an open pond.

15 A prominent example is the 10,000-kw. Wyoming Avenue turbine station of the Philadelphia Rapid Transit Company, where this cooling pond is employed during a portion of the summer months. It has been suggested that the sprays be mounted upon the power-station roof, thereby taking advantage of the inclined surface of the roof for extra cooling effect, suitable gutters returning the water to the cold well. There might be some hesitancy about installing a reservoir on the roof; but in one notable instance, the recently designed gas-power station of the Duquesne Lighting Company, Pittsburg, Pa., the roof reservoir forms a very effective part of the cooling system. Here a small cascade type of tower assists in cooling. Without other agency this simple system requires only 10 to 20 per cent make-up water.

b The simple tray type, Fig. 3, with water dripping through perforations, and cooling entirely by means of transverse air currents from the side.

16 Here no direct draft is possible, and the tower has no direct cooling surface. The trays operate simply to arrest the fall of the water. In this respect, the type is a simple mechanical refinement of a rough frame tower filled with brush, such as has often been employed in temporary power work. It is, however, comparatively inexpensive, and under some conditions, may be utilized to advantage. The tower of the Potosina Electric Company, San Luis Potosi, Mexico, is built entirely of structural materials and cools from 10 deg.

to 30 deg. with very low humidities. Although encased in netting to prevent loss by spraying, as much as 10 per cent of the volume passing through the tower is carried away during a brisk wind. The inexpensive construction is shown in Fig. 5, using horizontal wire screens instead of perforated trays and without wind screens.

c The simple cascade type, constructed either of wood or of corrugated sheet, in which a considerable part of the cooling is by actual conduction.

17 In the case of the original gas engine service plant of the Union Switch & Signal Company, Swissdale, Pa., this cascade system materially assisted in the work of cooling the gas engine jacket water, but the abstraction of heat through the concrete walls of a large reservoir was largely responsible for the cooling. This cascade system seems to have been overrated. In one prominent plant, the author understands it to have been a decided failure; in any form, it is extremely primitive and not in accordance with effective design.

d Another representative of the simple types of construction is the multiple cascade. (See Fig. 8 *a*)

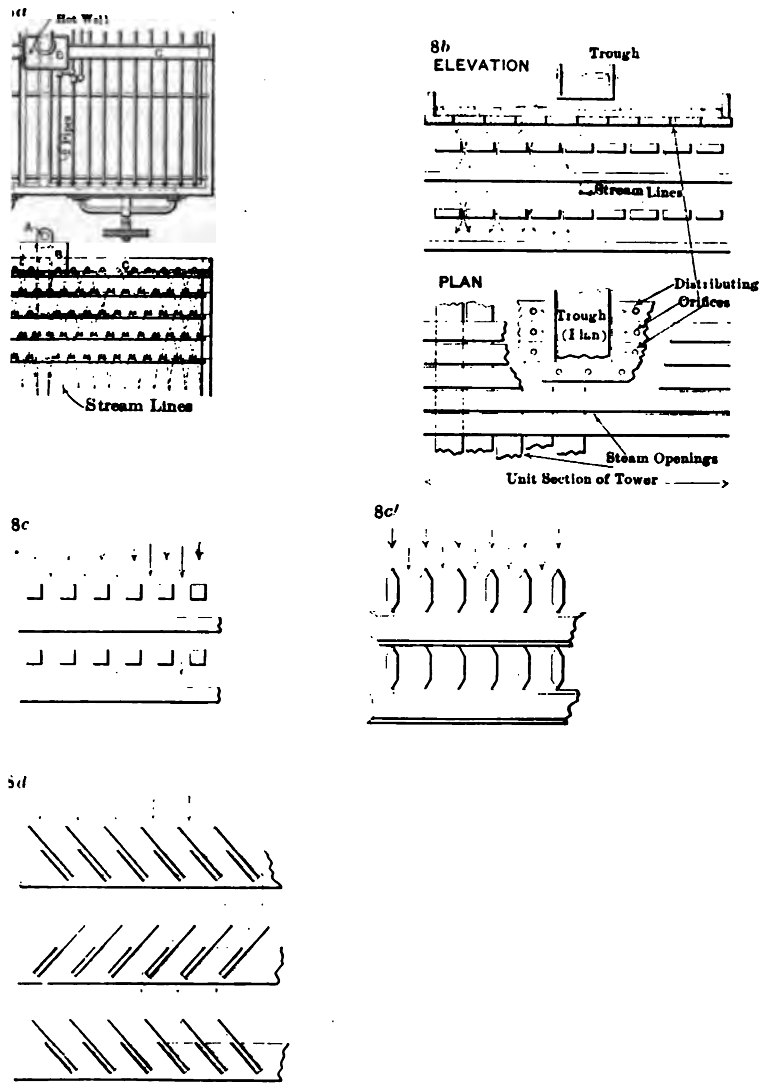
18 Here the fall of water is simply interrupted at short intervals, and no cooling surface is installed. It is evident that successful operation is dependent entirely upon the accuracy with which the trajectory of the falling particles can be predetermined in the spacing of trays and maintained in the subsequent operation of the tower. This would require an absolutely constant head.

19 The tower at Colorado Springs (Fig. 4) utilizes the construction, as in Fig. 8 *b*, a horizontal slotted surface with wind shields to prevent spray loss. This tower gives 40 deg. cooling in fair weather. The humidity however is very low, around 50 per cent (relative).

e Several American towers are constructed simply of horizontal lattice work, usually of cypress, the numerous tiers being staggered in order to break more effectively the fall of water (See Fig. 8 *c*).

20 In some, the upper and lower faces of the lattice work are beveled (Fig. 8 *c*) to lessen the resistance of descending water and ascending air. Cooling water is distributed by atomizing nozzles, by numerous spray pipes, or by Barker's mill.[1] This type evidently does not lend itself readily to natural-draft work, owing to the serious resistance offered to the draft by the lattice work.

[1] Radial arm distributor propelled by lateral reaction of its own jets.



3. 8 *a*, *b*, *c*, *d* TYPES OF COOLING SURFACES EMPLOYED IN VARIOUS TOWERS

f A modification of the multiple cascade system, used in a German tower (Fig. 8 *d*), endeavors to utilize partly the inclined deflecting surface as a cooling medium, although it is a question whether this is of much effect owing to

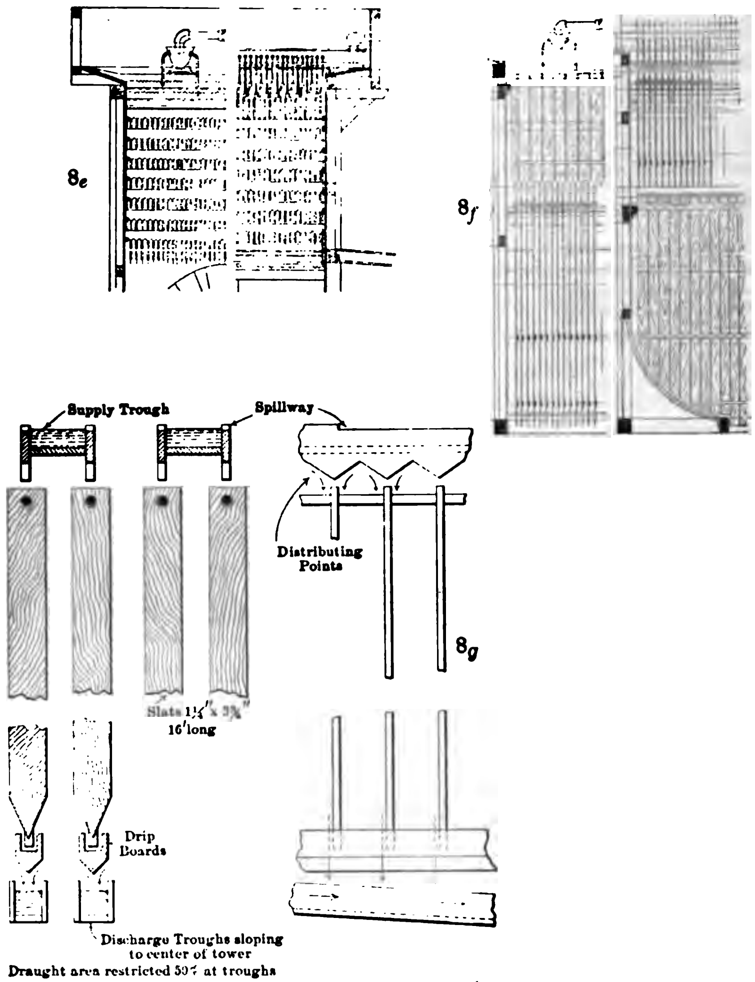


Fig. 8 *e, f, g* Types of Cooling Surfaces

the fact that the ascending air in all cases impinges on the lower surface of the deflectors, and not on the upper wetted surface.

g Another German design (Fig. 8 *e*), which has been introduced into this country, advances one step in introducing vertical cooling surface in transverse tiers. But most important is the attempt to guide the water downward in the form of a film, by forming each slat with a saw-tooth edge, meeting the lower transverse slats and guiding the water streams thereon.

21 The designer has evidently appreciated the necessity of avoiding free fall of water and deliberately piped the water from the flume to small troughs serving the upper row of slats.

22 An American builder has modified this system (Fig. 8 *f*) by practically discarding the numerous tiers of slats for vertical ones extending halfway down the tower, turning them 90 deg. at the middle of the tower, ostensibly for the purpose of equalizing air distribution. However effective this may be, we have here the desirable elements of continuously wetted surface and no free fall of water.

h A well-known American type, resembling the multiple-surface German tower (Fig. 8 *e*), employs numerous tiers of galvanized iron or tile cylinders, with a distributor at the top, of the Barker's mill type, propelled simply by reaction of the issuing jets.

23 Although highly effective in fan-type towers, there is much free falling water owing to the non-continuity of cooling surface; and in the author's experience, there is some objection to the Barker's mill distributor in the difficulty of maintaining ball bearings in proper condition.[1]

i Several builders employ continuous galvanized-iron surface from top to bottom of tower, either in the form of corrugated sheathing or of wire mesh, the water being carefully guided to the sheets so as to avoid free fall. The principle is right. With the close spacing permissible, a most intimate contact of air and descending film may be maintained.

j Coming now to exclusively wooden-mat construction, an example of the attempt to combine in a single-slat construction all the above-mentioned desirable features, is that shown in Fig. 8 *g*.

24 This resembles the form employed by Mr. Gerry, at Butte, Mont., although it is sketched from a design by Mr. Moser, of the Newhouse Mines & Smelters Company, Newhouse, Utah. This tower is

[1] In a Detroit station, the entire condensing plant lost its vacuum on several occasions at peak load owing to the stoppage of this distributor; and, finally, a three-deck phosphor-bronze ball bearing had to be designed to withstand the corrosion from the ascending vapor.

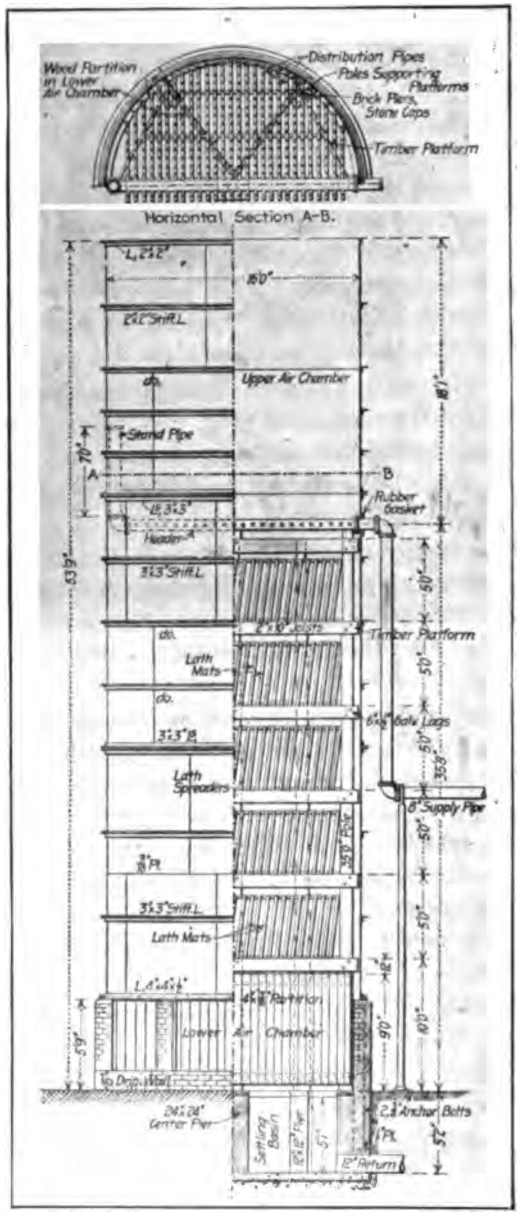


FIG. 9 EXPERIMENTAL NATURAL-DRAFT TOWER AT DETROIT, MICH.

of the natural-draft type with a side flume communicating with numerous transverse ducts which discharge upon continuous vertical slats, the saw-tooth construction being employed to guide the water on to the wetted surface. At the bottom, instead of allowing the water to fall freely into the receiving basin, each descending sheet is caught in a small trough and conveyed to the center of the tower, where it descends without retarding the ascending current of air. These distributing troughs reduce the effective draft area of the tower by about 40 per cent; but, on the other hand, the reduction in area is fairly uniform throughout the tower, and the area correspondingly diminished. That this type is extremely effective is proved by the results of the tests at Butte.

LATH MAT CONSTRUCTION

25 In a design originated in Detroit, Mich., shown in detail in Fig. 9,[1] an attempt was made to subdivide the cooling surface into sections or tiers, while maintaining the advantages of continuous vertical surface. This it was thought would facilitate the construction and repair of the tower; it was also hoped to avoid the distortion of the mat surface occasioned by the swelling[2] of the timber, which it is hard to avoid when long slats are employed.

26 This tower was designed under the direction of Alex Dow, Mem.Am.Soc.M.E., general manager of the Detroit Edison Company, largely in order to try out the natural-draft type under conditions of central station operation. There were many features which could have been improved upon, but that the type of mat surface employed was extremely effective is shown by the results of tests made in 1902.[3] The important point in design was reduced cost of construction. With the exception of the sheet-steel shell furnished by a local boiler

[1] Described in *Engineering News*, March 20, 1902.

[2] For example, this difficulty was experienced in a large gas-engine cooling tower in Texas. Plain horizontal platforms were used, with boards spaced far enough apart for the water to drip through. After some time in service, the timber had swelled to such an extent as practically to close off two-thirds of the tower, deflecting the greater portion of the water to the sides, where it descended without being cooled to any extent. This trouble, of course, is not so serious in the vertical slat tower; yet in a Boston plant employing rough boards set vertically on edge, the boards so swelled and warped that they practically closed the intervening air passages at certain points. This distortion may be noted somewhat in the Colorado Springs tower (Fig. 4).

[3] Conducted by the author and by Messrs. Armstrong and Richardson.

maker, the tower was built by unskilled labor employed about the station, and its total cost, including shell, concrete and brick work, material and labor, was in the neighborhood of $1350, serving a 1000-h.p. engine-driven plant. The shell was designed self-supporting with an independent internal frame work for bearing the weight of the mass. Wooden sheathing could have been used to advantage, however, and the entire tower constructed by unskilled labor.

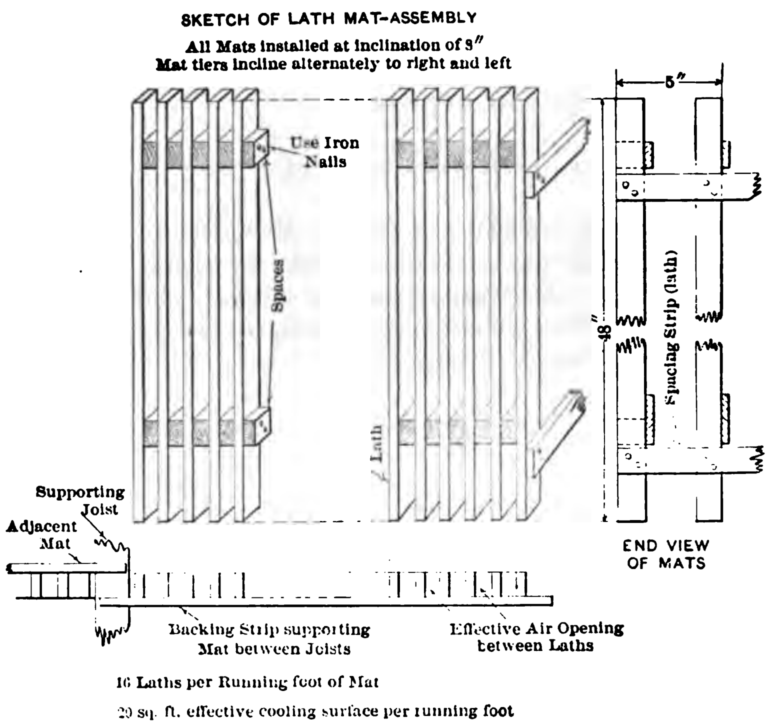


Fig. 10 *a* Details of Sectional Mat Surface

27 The mat surface was constructed of common wood lath, assembled on a form, with iron nails protected from corrosion by being imbedded in the wood. Mat details are shown in Fig. 10 *a*. These lath mats produced a very desirable form of cooling surface. The rough surface kept the descending stream in constant agitation, and there was sufficient slope to prevent free falling water for any great distance, and also to constrain the ascending air to slice upward through the interstices, thereby bringing into use both sides as well

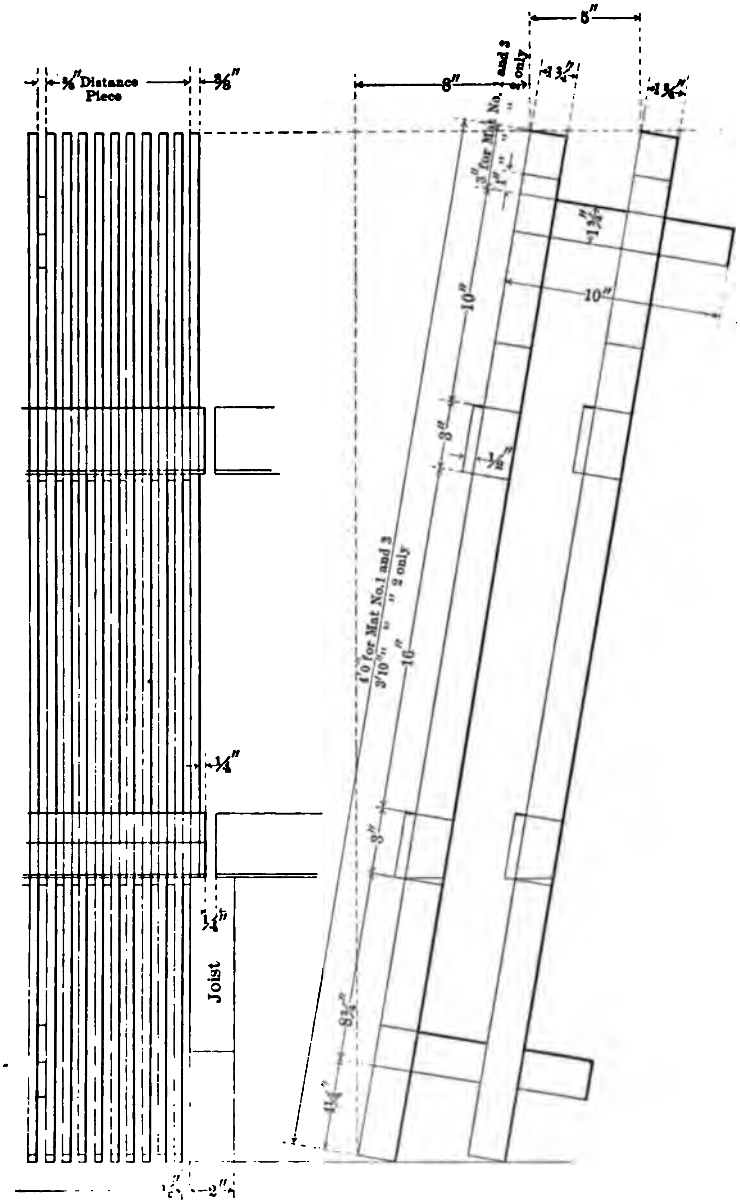


FIG. 10 b DETAILS OF SECTIONAL MAT SURFACE as both edges of the lath. Thus a cooling surface of approximately 20 sq. ft. per running foot of lath mat was obtained. The various tiers were readily assembled in succession, working from the shell inward until full. Uniform water distribution was effected by means of the pipe-spray system, with laterals spaced like the mats below.

28 Two series of tests[1] were made at Detroit at different times, first with only the two upper tiers, and finally with all the mats in position. Tables 4 and 5 and Figs. 11 to 13 show the relation between

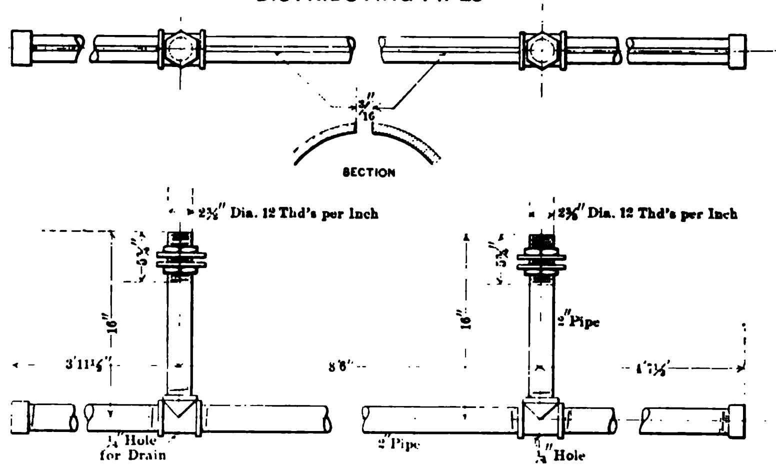


FIG. 10 *c* DETAILS OF DISTRIBUTING PIPES

the various quantities observed. It was very noticeable that the rate of heat dissipation in B.t.u. per sq. ft. per hr. was considerably higher for the uncompleted tower with only about three-fifths of the mats in operation. However, by the addition of the remaining mat surfaces the tower was enabled to work on lower water temperature, and we should therefore expect a lower rate of heat dissipation. This would indicate that the upper tiers of towers were more effective

[1] In this plant the condensing system was not well adapted to economic working. Air and circulating pumps were direct-coupled, making it impossible to control the tower water separately from the condensation. There was considerable air in the system from a long run of exhaust piping; and with no dry-air pumps, a vacuum of 24 in. was normal practice. But the condenser was operated with a temperature differential of 47 deg., so that *with an efficient condenser*, a vacuum of 28 in. might have been obtained with the same tower performance, 16 deg. cooling, 71 deg. cold well.

TABLE 3 COMPARATIVE DATA, FORCED AND NATURAL-DRAFT TOWERS

TYPE	FORCED-DRAFT STATION *A*	NATURAL-DRAFT STATION *C*
Rated Engine, i.h.p.	1500	900
Cooling surface, sq. ft.	34,780	24,500
Surface per h.p.	23.2	27.2
Space occupied, cu. ft.	7064	10,850
Space occupied, sq. ft.	175	200
Cost complete	$3900	$1350
Cost per h.p.	$ 2.60	$ 1.50
Auxiliaries, e.h.p.	13	..
DIMENSIONS		
Delivery pipe above ground	29 ft. 0 in.	35 ft. 8 in.
Height over all	40 ft. 6 in.	53 ft. 9 in.
Height mat section	17 ft. 0 in.	25 ft. 0 in.
Height stack	12 ft. 6 in.	18 ft. 1 in.
Height outlet	10 ft. 10 in.	9 ft. 0 in.
Diameter tower	14 ft. 10 in.	16 ft. 0 in.
Diameter fan	9 ft. 3 in.	..

TABLE 4 TESTS OF NATURAL-DRAFT COOLING TOWER, DETROIT

INCOMPLETE, THREE-FIFTHS SURFACE INSTALLED

TIME	Temperature, Deg. Fahr.					Quantities				
	AIR	HOT WELL*	COLD WELL	WATER COOLING	TOTAL HEAT HEAD†	TOWER WATER LB. PER HR.	HEAT DISSIPATED B.T.U. LB. PER HR.	HEAT PER SQ. FT. COOLING SURFACE B.T.U. PER HR.‡	CIRCULATING WATER PER SQ. FT. LB. PER HR.	LOAD KW.
1	2	3	4	5	6	7	8	9	10	11
12 noon	34	102	89	13	68	375,000	4,880,000	332	25	270
1.30	35	106.5	90	16.5	71.5	§ {375,000 370,200	6,108,000	415	24.8	{315 290
2.30	35	106.5	87.5	19	71.5	375,000	7,120,000	484	25	315
3.30	35	113	88.5	24.5	78	375,000	9,000,000	613	25	350
4.30	32.5	100	84	16	67.5	399,000	6,384,000	434	26.6	365
5.00	28.5	103.5	88	15.5	75	445,500	6,900,000	470	29.7	485
6.00	26	125	94	31	99	417,000	12,930,000	880	27.8	655
7.00	24	121	94	27	97	427,000	11,532,000	785	27.4	570
8.00	24	123	94.5	28.5	99	427,000	12,174,000	827	27.4	600

*Assuming a more efficient condenser, say 10 deg. difference, the probable vacuum would be 26 deg. to 27.5 deg. This condenser actually operated at 40 deg. to 50 deg. difference.

†Total heat head = air heating + lost head.

‡Only three-fifths cooling surface installed.

§Difference due to rapid change in load.

TABLE 5 TESTS OF NATURAL-DRAFT TOWER, DETROIT

Complete, Five-Fifths Surface Installed

Engines:	**Two 400. i.h.p. 300. kw. McIntosh & Seymour tandem compound engines, overhung generators.**			
Condensers:	**Worthington surface (admiralty type) 1600-sq.ft. reciprocating wet-air pump and circulating pump.**			
Tower:	**Wood mat construction, 24,500-sq.ft. evaporating surface, exclusive of shell.**			
Test:	**March 15 to 16, 1901, 4 p.m. to 4 p.m., 24 hr.**			
		A.M.	**P.M.**	**AVERAGE**
Weather:	**Barometer (abs.), min.**	**30.22**	**30.07; 30.14**	**30.27**
	Temperature air, deg.	**18.5**	**25; 30**	**25**
	Relative humidity, per cent	**76**	**82; 38**	**72**
Load:	**600 kw. max. to 50 kw. min. Average**			**244.9 kw**
	Engine efficiency = 92.5 =875 i.h.p. max. Average			**354.8 i.h.p**
Steam:	**Weight of condensed steam per hr., lbs**			**5910.6**
	Temperature exhaust steam, deg. fahr			**134.38**
	Temperature condensed steam, deg. fahr			**108.78**
	Weight of steam per hr., max. load, lbs			**13,500**
	Vacuum (abs.) 25 to 19, average about			**22**
	Vacuum corresponding to temperature exhaust steam			**25**
	Vacuum possible with good condenser (10 deg. difference)			**28**
Water:	Circulated per hr., lbs			293,536
	Temperature hot well, average, deg. fahr			87.50
	Temperature cold well, average, deg. fahr			71.27
	Vaporization loss per hr., lbs			5,970
Results:	Condenser surface per kw., sq. ft			2.66
	Steam per kw. hr., lbs			24.3
	Steam per i.h.p. hr., lbs			16.66
	Circulating water per lb. of steam, lbs			49.6
	Steam per sq. ft. condenser surface per hr., lbs			3.7
	Circulating water per sq. ft. tower surface, lbs			12.0
	Difference in temperature between exhaust steam and discharge, deg. fahr			47
Cooling:	Max. 20 deg., min. 3 deg.-5 deg. Average			16.23
	Heat dissipated per hr., B.t.u			4,769,000
	Heat per sq. ft. tower surface, B.t.u			195
	Heat per sq. ft. per 1000 lb. water, B.t.u			0.665
Evaporation:	Circulating water, per cent			2.03
	Engine steam, per cent			101
Tower:	Surface per kw. (average load 245 kw.), sq. ft			100
	Surface per kw. (max. load 600 kw.), sq. ft			408
	Surface per 1000 lb. steam max. load, sq. ft			1.82
	Surface per 1000 lb. steam average load, sq. ft			4.14
	Surface per 1000 lb. circulating water per deg. max. cooling, sq. ft			5.22

than the lower. The heat dissipation during the tests on the complete tower ranged from 200 B.t.u. to 300 B.t.u. per sq. ft. of surface per hour under normal conditions, and this could undoubtedly have been increased in a carefully constructed tower with suitable condenser apparatus.

29 In general, the tower showed very little difference in efficiency summer and winter, rather against expectations. Apparently the increased evaporation possible in the higher air temperatures of summer offset the greater conduction of heat in the colder air of winter. In very hot weather, a negligible effect from conduction was apparent, from the fact that at certain maximum loads the vacuum fell rapidly, indicating that the capacity of the tower had been reached, due to complete saturation of air, while in cold weather the vacuum would hold up better at the same load. This shows that with air fully saturated and evaporation checked, the dissipation of heat by conduction in hot weather was quite insufficient to give an appreciable margin of overload.

30 This tower was in constant use for a period of about four and a half years, cooling all of the condensing water for the central station. Depreciation was at first thought to be a serious factor, but later, when the tower was finally dismantled, the frame work and mats were found to be in excellent condition. The only parts showing deterioration were the upper sheets of the tower shell lying above the distributor, where corrosion had taken place owing to the alternate wetting and drying of the surface during the last six months of service in 1905 (10-hr. operation) The mats themselves were as sound as when put in. After a few months' service, the mat surface became coated with scale due to the incrustating properties of the water. This scale would accumulate, crack off and fall to the settling basin.

31 Although fairly successful, this experimental design might have been considerably improved. By straddling the supporting joists in the manner shown in detail in Fig. 10 *b*, the various tiers of mats may be brought together into a practically continuous surface from top to bottom, thus entirely preventing the fall of water. At the bottom , the obstruction to draft may be prevented by employing deflecting troughs under each mat, to convey the water to the center of the tower, as in the Moser tower, Fig. 8 *g*. A better distribution system in the form of horizontal slotted laterals discharg ing upward and over-flowing directly on to the respective mat sections is shown in Fig. 10 *c*.

32 In any system of stationary jets, it is extremely difficult to

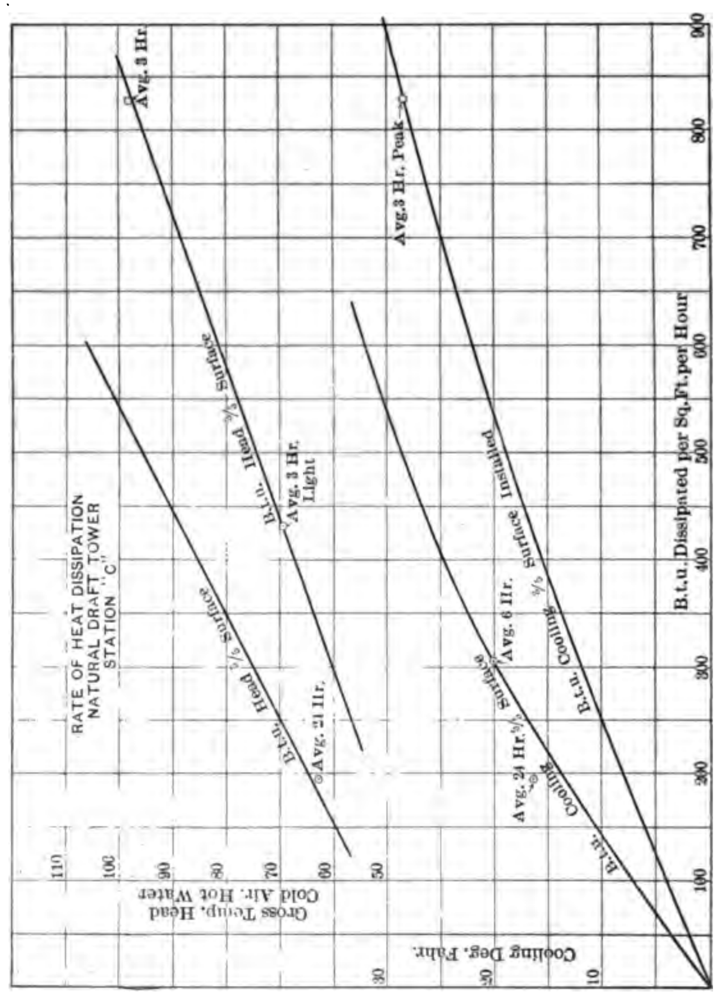


Fig. 11 Rate of Heat Dissipation of the Detroit Tower

tain uniform distribution of water over the entire tower at different ;es of flow. With the slotted pipes, it is an easy matter to open close the slots so as to distribute uniformly, and as they are laid rizontally, this adjustment is permanent. It is also easy to free ; laterals from foreign matter, as is not the case with jets. This because of the ample section of the laterals; whereas any system ng a large number of small distributing pipes or apertures involves .imate trouble from clogging.

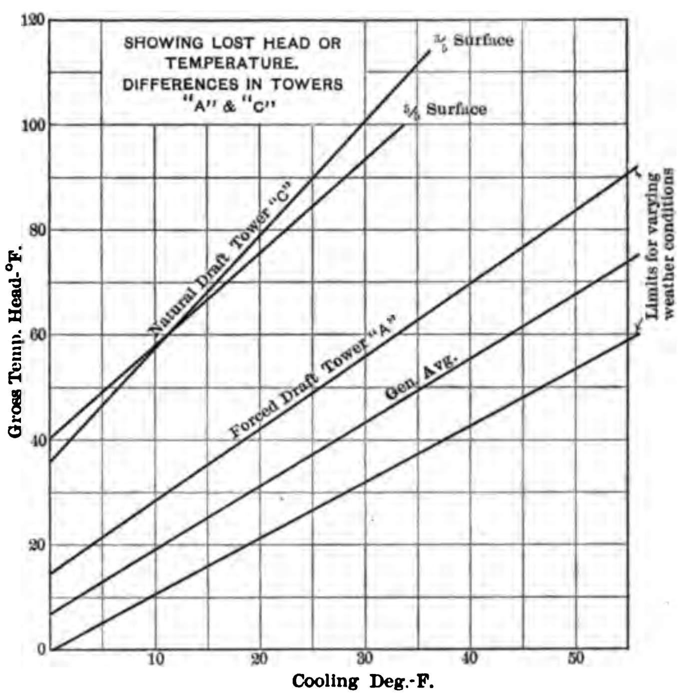


FIG. 12 LOST HEAD OF NATURAL AND FORCED-DRAFT TOWERS

THE EVAPORATIVE COOLER

;3 In gas-engine work it is often necessary to economize water the greatest possible extent. In an Arizona mining plant employ- gas engines, where the mine water was so foul and acid as to hibit entirely its use for cooling jackets, an evaporative cooler was ently constructed of ordinary hot-water radiators arranged in ies-parallel, with air forced over the surface by a motor-driven fan. The well-known counter-current system was employed, and the outfit was fairly efficient, the jacket water being cooled 15 deg. with a power consumption of 5 per cent of the output of the engine.

34 It occurs to the author that by keeping the radiator surface continually wet the effect of evaporation as well as convection might be utilized in cooling. The foul mine water may sometimes be used for this purpose without contaminating the jacket circulation. With an expenditure of 2¾ per cent in evaporation, an increase of 24 per cent in cooling would be obtained, assuming the air entering and

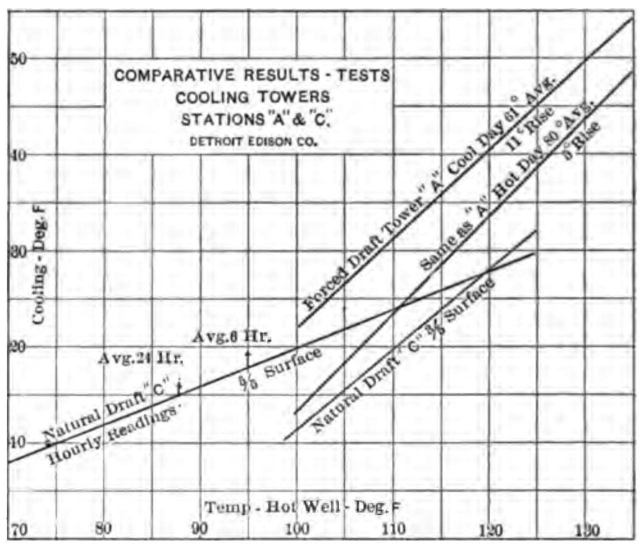
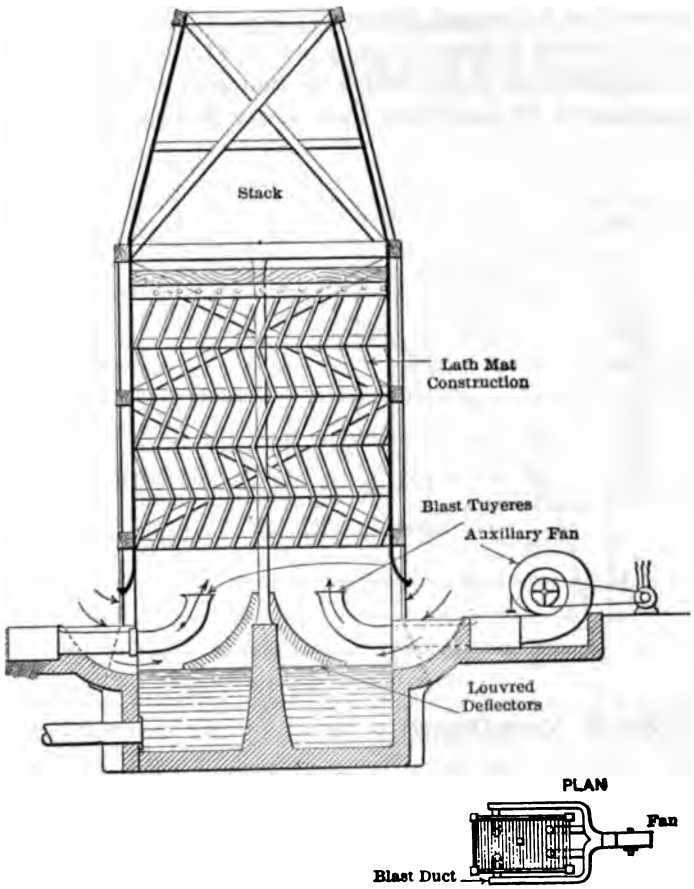


FIG. 13 COMPARATIVE RESULTS, NATURAL AND FORCED-DRAFT TOWERS

leaving to be fully saturated (Fig. 15). This system has been attempted in connection with steam condensers, but apparently without much success. The principle seems entirely logical, but the difficulty of maintaining tight joints with thin-walled tubes of sufficient diameter to permit of the passage of the proper amount of air, would seriously detract from the effectiveness of this apparatus by reason of air leakage. The low vacuum shown during tests of such apparatus largely confirms this supposition. For gas-power plants, however, the type seems admirably suited.

STANDARDS OF DESIGN

The cooling tower should be designed with the same flexibility
er good power-plant apparatus, as regards capacity under
conditions of operation; it is subject to the same peak-loads as
ne mover. As a matter of fact, relatively more heat must be



PROPOSED COMBINATION TOWER WITH NATURAL AND AUXILIARY-FAN DRAFT

ted by the tower during peaks owing to the higher steam con-
on of a steam engine per horsepower-hour on overloads. Con-
or example, a normal central station load. The evening peak
extends over three hours, and usually the most severe demands

on the generating apparatus occur within a period of one hour. Here, then, a definite overload capacity in the cooling tower is as desirable as in the engine or boiler; and some means should be employed to relieve during these peaks the tower which would have ample capacity to operate unaided during the remainder of the day.

36 Again, consider the comparatively short periods of unfavorable weather in normal climates. Reports[1] from the Butte plant (Table 2) reveal a mean temperature of 41.6 deg. fahr. Yet there were 99 days of the year in which the temperature was above 70 deg.; 50 days above 80, and three days above 90 deg. fahr. Taking 70

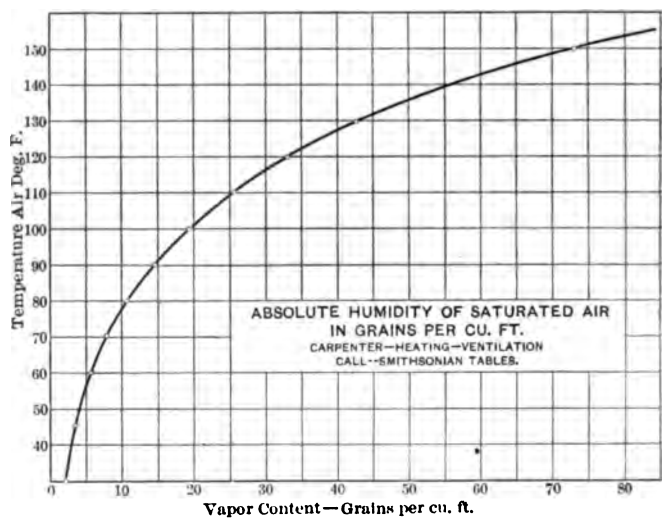


FIG. 15 VAPOR-CONTENT IN AIR AT VARIOUS TEMPERATURES

deg. as an empirical standard, it is apparent that for two-thirds of the year this temperature would not be exceeded. But careful *hourly* observations at Butte show that a temperature of 70 deg. was exceeded only for 691 hr. throughout the year; i. e., ***7.9 per cent of the actual time.***

37 A study of Pittsburg conditions shows similar results (Table 1). Average throughout the year, 52.3 deg. The temperature during June, July and August averaged 10.5 days per month above 85 deg., and the humidity, 15.3 days per month above 80 per cent.

[1] Provided by Mr. Gerry.

Although the actual *hours* of maximum are not available from the Weather Bureau reports, it is safe to say that these unfavorable atmospheric conditions existed not more than one-tenth of the daily period, or 2.5 hr.

TABLE 6 OPERATING DATA, OPEN-SCREEN TOWERS

MT. WHITNEY POWER CO., VISALIA, CAL.

Tower designed for 1500 kw.—2000 kw. at 27-in. vacuum

Horizontal screen surface, sq. ft.	9550
Circulating water handled, gal. per hr.	1,720,000
Rate of circulation, lb. per sq. ft. per hr.	1500
Dimensions, ft.	30 by 47 by 15 high
10 tiers galvanized iron screens.	5 mesh per in.
Cost of tower including concrete form.	$2,000

OBSERVATIONS, OCTOBER 23, 1906

Maximum load carried, 5.20 p.m., kw.	1130
Temperature atmosphere, deg. fahr	55
Depression, wet bulb thermometer, deg.	8.5
Relative humidity, 50 per cent absolute, gr. per cu. ft.	2.35
Temperature incoming hot water, deg. fahr	110
Temperature outgoing cold water, deg. fahr	100
Cooling, deg. fahr. (minimum for day)	10
Vacuum carried (ref. 30-in. barometer), in. Hg.	26.6
Difference between temperature steam and condenser discharge	
Possible vacuum (10 deg. difference in condenser)	
Maximum cooling for day (730 kw.), deg. fahr	16

Data from Hunt, Mirk & Co., Engineers, San Francisco, Cal.

38 It is apparent from the above that the problem of maximum capacity in cooling towers involves a condition of peak load existing only 5 per cent of the time, and high temperature only 8 per cent of the time. Moreover, these maximum demands will not generally occur at the *same hours* of the day. In the example cited in Par. 11, 5 deg. difference in the condenser, the maximum permissible air temperature would be 81 deg.; the more efficient the condenser, the higher the allowable air temperature. Yet at Butte, the temperature of 85 deg. was exceeded during only 22 days of the year, or 51 hr. This is equivalent to 2.5 hr. during mid-day and less than 0.6 of 1 per cent of the total time.

39 Is it, therefore, good engineering to design a cooling tower installation with a vacuum-producing capacity *large enough for any*

and all emergencies; or, on the other hand, to provide auxiliary means for assisting during these brief periods of maximum demand, while keeping the proportions of the tower within reasonable limits for normal operation? Might not even a considerable impairment of vacuum under the most unfavorable operation be better tolerated than the increased expense of equipment suited to maximum demand?

40 This, of course, applies particularly to natural-draft towers. Flexibility already exists in the forced-draft tower through the speeding of the fans; but even here there are some drawbacks owing to the high velocities already employed for normal working. Any large increase in the velocity of the fan may seriously disturb the uniformity of air distribution over the tower surface and give rise to eddies destructive to efficiency. That this condition exists, is very plainly shown by a survey of the discharge velocity by means of an anemometer. Examination of one defective installation by this method revealed the fact that fully one-third of the area was practically ineffective and that reverse currents actually took place in some parts. The air-distribution problem is exceedingly important, and more so in the forced-draft than in the natural-draft tower, where low velocities favor uniformity.

"BOOSTER" TYPE OF TOWER

41 The natural-draft tower is of itself ill-adapted for operating with a fixed temperature head. It thrives on the weakness of the condensing system. The lower the vacuum, the better the tower works, because of the increase in temperature head. And as this is clearly a problem of chimney design, the only way out of the difficulty is apparently by some method of auxiliary draft. As the specific heat of air is about 0.23, it is evident that an increase of 25 per cent in heat dissipation would require roughly double this increase in quantity of air, in order to maintain the same temperature conditions. This, however, is well within the capacity of a comparatively small fan auxiliary.

42 There are two methods of accomplishing this result:

- *a* By locating in the stack an induced-draft fan which normally remains idle.
- *b* By installing at the base of the tower a forced-draft system so designed as to supplement the natural draft without causing a back-flow.

43 The method of *a* is used in the cooling towers at Gary, W. Va. (Fig. 2). Although designed for constant service, the arrangement of the stack fan is precisely as suggested. Whether the Pelton type of motor, direct-connected to the fan, is superior to belt or chain drive from a motor mounted on the outside of the tower, is a question of mechanical convenience: the essential elements are present.

44 The second suggested arrangement is crudely shown in Fig. 14. The auxiliary air is delivered to the tower through four "L"-nozzles supplied from a concrete duct surrounding the base of the tower. With this arrangement, the natural draft under the base of

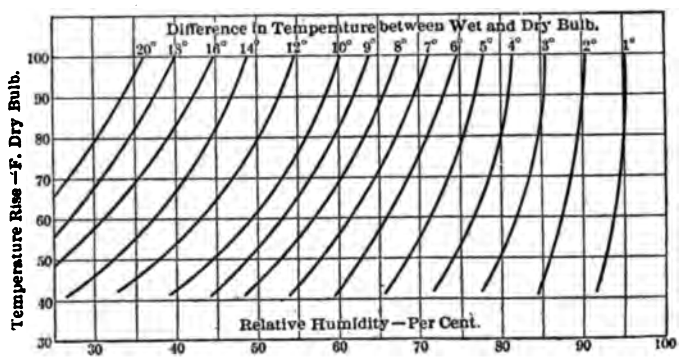


FIG. 16 HUMIDITY CHART FOR WET-BULB THERMOMETER

the tower might tend to be reversed owing to the back-pressure resulting from the blast. It is believed, however, that with a fairly open mat structure such as has been described above, the introduction of four auxiliary blast ducts would serve only to entrain more air and further assist the tower in the absorption of heat.

45 Of the two systems, the former has the advantage of being already put into practice. However, there is to be said in favor of the latter that no working parts, such as fan bearings, belt transmission, etc., are in the current of vapor; and with a tower operated intermittently, corrosion is an important matter, as was proved by the deterioration of the upper-sheets at Detroit. Furthermore this system lends itself more readily to a square or rectangular-shaped tower, which may be desirable in large sizes.

CONCLUSIONS

46 Improved types of cooling towers are in active demand for high-grade power plants both steam and gas, especially for low-pressure turbine installations.

47 Effectiveness of working is largely dependent on efficiency of condenser, i. e., minimum temperature difference between steam and discharge water is desirable to increase the temperature head on the tower.

48 Cooling towers are particularly adaptable to gas power plants. The bulk of the tower is reduced by the high temperature head available with hot jacket water.

49 Elements of most effective design: avoid free falling water; maximum retardation of descent with minimum obstruction of draft; insure uniform distribution of water and air; provide the maximum exposed wetted surface for a given bulk and an interrupted descent of fluid film.

50 In locations of low humidity simple forms of construction usually serve the purpose, except where ground space is valuable.

51 Sectional lath mat type of tower well adapted to natural-draft work. The construction suggested is simple, durable and inexpensive.

52 Normal rate of heat dissipation obtained by lath mat construction, 200 B.t.u. per sq. ft.

53 Auxiliary fan "booster" suggested as the best means of obtaining the desired overload capacity—a combination of natural and forced-draft. Overload conditions (high temperature, humidity or load) usually last but a small percentage of the time—1 to 5 per cent. Natural draft suffices for the major portion.

DISCUSSION

Geo. J. Foran. Evidently Mr. Bibbins has intentionally restricted his discussion to the subject of the paper, the cooling tower. He has, however, presented certain tables which, without discussion, are liable to be misleading with reference to the condensers and general cooling-tower condensing situation.

2 The paper discusses the tower quite fully, but classifies the condenser as good, bad or worse without discussion. This is made possible by assuming that the various condenser results obtained are simply a question of condenser design. This permits the inference to be

drawn that the various results can be obtained at the same, or practically the same, cost, which is incorrect. Some of the results stated are possible of attainment, but would not show profitable investment.

3 It is impossible to differentiate the tower and condenser quite so completely as in the paper. Each is strongly influenced by the possible range in operation of the other, and I would like to show just how the relative sizes and consequent costs of the plants will be modified by the results desired.

4 Observers agree that the heat transferred through condensing surface varies directly with the mean temperature difference between the two sides of the tubes. Whether this mean should be arithmetical or geometrical is immaterial for the present discussion, and for simplicity I have selected the arithmetical mean.

5 It is unnecessary to assume condensers of varying grades of design and efficiency; in fact, it hopelessly complicates the question, and for my discussion I have assumed a condenser of uniform design and maximum efficiency with a varying amount of surface, which will permit us to obtain the various results tabulated by Mr. Bibbins.

6 The fairly universal practice for high-vacuum work for the past few years has been that for a 15-deg. rise in temperature of the incoming circulating water, during its passage through the condenser, it will be brought to within 15 deg. of the temperature corresponding to the vacuum. The proposition is frequently made to add only 10 deg. to the water and bring it to within 10 deg. of the vacuum. This is perfectly feasible, but we must see what this involves.

7 It means, first, that if we must carry away the heat from the steam by increasing the temperature of the circulating water 10 deg. instead of 15 deg., we must have 50 per cent more water with consequently larger and more expensive circulating plant and piping. With a 15-deg. rise to within 15 deg. of the vacuum temperature, the mean temperature difference between the steam and water side of the tubes will be 22½ deg. With a 10-deg. rise to within 10 deg. of the vacuum temperature, the difference will be only 15 deg. or, in the latter case, 50 per cent more surface will be required.

8 Following the 28-in. vacuum line in Fig. 7, it will be noted that Mr. Bibbins has added practically 15 deg. to the condensing water and has given three curves—one for a good condenser with a temperature difference of 10 deg.; a very efficient condenser, 5 deg.; a perfect condenser, 0 deg.

9 Let us consider only the perfect or maximum-effect condenser with varying surface to produce the results named. For the 0-

deg. curve the mean difference between the steam and water side of the tubes will be 7½ deg; for the 5-deg. curve this becomes 12½ deg. and for the 10-deg. curve, 17½ deg. Or, if we should take the case where we add but 10 deg. to the water, these three mean differences would become 5 deg., 10 deg., and 15 deg. respectively, so that the condenser for the 0-deg. curve would have twice the surface required by the condenser on the 5-deg. curve and three times the surface required for the 10-deg. curve.

10 While there are several plants which report a circulating delivery temperature at approximately the temperature of the vacuum, it is evident that no plant should depend upon such a performance to obtain the economical results upon which the plant investment is based, as this would require absolutely perfect test conditions in every day operation; it would give no leeway at all and would result in too wide a variation in performance for a slight falling off in operating efficiency. Even a slight air leak would result in lowering the temperature in the vacuum space 5 deg., with a consequent loss in heat head and reduction in heat transference, owing to the presence of the air itself. These matters must be considered in addition to the question of cost.

11 Again, following the 28-in. vacuum line in Fig. 7 until it intercepts the 10-deg. curve, it will be found that it calls for water at 75 deg., the 5-deg. curve calls for 80 deg. and the 0-deg. curve for 85 deg. All these conditions assume that these results depend only upon the condenser, and if I understand the table correctly, call for the same quantity of steam and water, the temperature of the circulating water, it will be noted, being raised 15 deg. in each case. The author also assumes that the water is cooled to the temperature of the outside air.

12 Although I am sure that the author does not intend to convey the apparent meaning, the further statement is made that this calls for a fixed cooling-tower performance; in other words, as I understand it, that the size of tower and the performance will be the same, to cool a given quantity of water through the same range in temperature, irrespective of the temperature of the air.

13 Let us follow this a little further, and in line with the general assumptions, assume for this purpose that the hot air leaves the tower at the temperature of the hot water and 100 per cent saturation. By reference to psychrometric tables it will be seen that each cubic foot of air at 70 deg. temperature and 70 per cent humidity, when increased to 85 deg. and 100 per cent, will take on 7.15 gr. of moisture, whereas

a cubic foot increased from 47 deg. and 70 per cent to 62 deg. and 100 per cent, will take on only 3.575 gr. of moisture; that is, although the temperature is increased 15 deg. just the same, the air carries away but one-half the moisture at the lower temperature, showing that twice the air capacity of tower efficiency will be required at the lower temperature. This is better understood when we consider that within the usual air temperature ranges, the moisture-carrying capacity of the air is doubled for each 22-deg. rise in temperature. To be brief and to avoid confusion, I have used the ordinary nomenclature, which is scientifically incorrect. We all understand that it is the space and not the air which is saturated, but this splitting of hairs would not affect the point under discussion.

14 I have purposely neglected the several minor considerations as they affect the question to a very small extent. For example, the volume of the air entering the tower at 70 deg. and 70 per cent humidity, and leaving at 85 deg. and 100 per cent humidity, is increased nearly 5½ per cent, due partly to the increased temperature and partly to the reduced pressure of the air itself, owing to the increased saturation and vapor present. It is well known that the cooling tower performs its work *principally* by the withdrawal of heat from the main body of water which provides the latent heat for the evaporation of a small portion of the water carried away in the form of vapor as increased humidity of the cooling air.

15 Temporarily omitting the *perfect* plant, let us consider an *average* operating plant in a location having air at 70 deg. and 70 per cent humidity. The usual cooling-tower turbine plant would carry a vacuum of 27 in. with water cooled from 100 deg. to 85 deg. If it is desired to cool this water from 90 deg. to 75 deg., this would permit of carrying a vacuum of 27¾ in. with the same amount of surface and water, but would require an increase in the quantity of air and of tower capacity of approximately 50 per cent. If it is desired to cool the water through only 10 deg., that is, from 85 deg. to 75 deg. and to bring the water within 10 deg. of the vacuum (28¼ in.) this would call for 50 per cent more water, 50 per cent more surface and over 100 per cent more air and cooling tower capacity than for the usual 27-in. vacuum plant.

16 There are hardly two plants which have quite the same determining factors. The determination as to the advisable vacuum and plant must be decided in each case, but there are few plants where the conditions would warrant the installation of a plant to produce the maximum vacuum under the most severe conditions.

17 With reference to the type of tower with fans in the stack, as shown in Fig. 2, the Worthington Company installed their first tower of this type with rope fan drive, in 1900, and recent reports indicate as good results as when the tower was installed. As a general proposition, however, there are several questions to be considered in comparing this type. There is a saving in the number of fans over the arrangement with the fans below the tower filling, but the fan operates in the hot, highly saturated air, is more or less inaccessible and out of sight, and therefore will not receive the best of attention. It requires good installation and is more difficult to maintain in good condition owing to the fact that it is an exhauster. Any of us would prefer to install a pressure fan rather than an exhauster; the capacity of the fan in the stack must be somewhat larger for the reason that as neither the circulation nor the surface efficiency is improved, the total volume of free air required is the same, this being handled at a less pressure and higher temperature and humidity.

18 Comparing the fan and natural-draft towers, there are few, if any, locations where high results are desired, where the natural-draft tower could be selected. A little calculating will convince any engineer that the draft is principally due to the wind velocity over the tower. Study of the meteorological tables will show that in most power centres, except in very few locations, the wind velocity is much greater in winter than in summer—just the opposite of our requirements. This is clearly demonstrated in the operation of any fan tower from the fan speeds permissible at different seasons. It must be remembered that with a tower of the same height the wind assistance is the same for either type of tower. There are many locations where a so-called combined tower can be used if the additional expense is warranted, but strictly speaking, the operation cannot be combined. It must be used either as a natural-draft tower or as a fan tower, but if the fan is operated at all, all the air must pass through it, whether the fan is located above or below the filling.

19 I do not see how there can be any induction in the tower shown in Fig. 14. The object of the tower is to get sufficient pressure below the filling to force through the requisite amount of air, but this pressure must be uniform in the entire space below the filling in order to obtain complete surface efficiency, and under such conditions air would leave rather than enter the tower through any additional openings to the outside air.

20 The Worthington Company make a so-called combined tower which permits of two water levels in the cold well. At the lower level

the air enters through the fan at rest and below the lower plates of the tower shell above the water. At the higher level the lower plates are sealed and all the air enters through the fans, which can be operated at the speed necessary to supply the additional pressure required by the low wind draft. This is also accomplished by the use of additional draft doors.

PROF. WILLIAM D. ENNIS. Will Mr. Bibbins explain in more detail the derivation of the curves in Fig. 7? The tower must provide cooling sufficient to absorb the heat liberated with the exhaust steam, viz., 939 B.t.u. per pound. The amount of cooling in each case would then be 939 divided by the weight of circulating water per pound of steam. On this basis, the maximum temperatures of entrant air agree closely with the curves at 27-in. and 28-in. vacuum, but are about 1 deg. higher than the curves indicate at 29 in., and 2 deg. or 3 deg. higher at 26 in. The curves should apparently be more nearly straight.

2 The paper gives unusually complete and valuable data on many phases of cooling-tower operation, but it is to be regretted that the matter of loss of water has not been dealt with in more detail. This is perhaps the most vital question. Manufacturers are sometimes asked to guarantee a limit of loss, but it would be just as logical to ask for a guarantee as to the value of π. A rough estimate often offered is that the loss will not exceed the amount of boiler feed water.

3 Mr. Bibbins gives data from three plants: that at Duquesne, in which the makeup water was from 10 to 20 per cent; the Potosina plant, in which the loss of vapor by windage was occasionally as much as 10 per cent of the volume of water passing through the tower; and the Detroit natural-draft plant, in which the vaporization loss was 2 per cent of the water passing through; practically equal to the weight of boiler feed. The average cooling per hour was $(293{,}530 + 5910.6) \times 16.23 = 4{,}860{,}018$ B.t.u. Each pound of water vaporized, if we neglect the cooling effect of the air must then have absorbed $\frac{4{,}860{,}018}{5970} = 816$ B.t.u. This is the nearest to a reasonable result I have ever seen in a cooling-tower test.

4 Usually, and this apparently applies to the two other cases cited by Mr. Bibbins, the loss of water is far greater than theory indicates as necessary. The cooling of the water is accomplished by (*a*) the absorption of heat by the air and (*b*) the evaporation of a portion of the water. When the minimum temperature of the air equals or

exceeds the maximum temperature of the water, the first effect becomes zero. When the air is initially saturated, the second effect becomes zero, except as the air is heated during its passage. Under the limiting condition at which there is no direct transfer of heat to the air, the necessary volume of air is increased, and the loss of water does all of the cooling; but the proportion lost need not exceed, in theory, the quotient of the range of cooling by the heat of vaporization, and the use of screens enables us even to reclaim some of the otherwise lost vapor. Why is it that almost invariably the make-up water greatly exceeds the amount thus computed as necessary? It is inferred from Par. 34 that Mr. Bibbins has considered this question of cooling by evaporation, in which case some exposition would be desirable.

HENRY E. LONGWELL. Very early in 1884, under the direction of John C. Dean, of Dean Brothers Steam Pump Works, I made drawings for a cooler that was built for the Kane Milling Company, Kane, Ill. I am told that it was the first one erected in the United States, and it is, at any rate, a well-authenticated case of a very early installation. The plant was operated for only two years, being then destroyed by fire, but so far as I can remember the installation acted in a very creditable manner, especially considering the primitive state of the art at that time.

2 There are probably many engineers who will take issue with the author if he means that the cooling-tower field is yet comparatively unexplored. For ten years or more the cooling tower has been on a strictly scientific basis. Its design and construction constitute a branch of engineering that is just as distinct and as well developed as any of those which deal with other specialties such as gas engines, steam turbines and the like. When we consider that one builder alone has constructed about 2000 cooling towers which in the aggregate are capable of cooling condensing water for about 3,000,000 horsepower, we must admit that this device has progressed a long way beyond the rudimentary stage.

3 It is not excessive cost or lack of knowledge that has restricted the use of cooling towers in the United States. It is because nature has been so good to us that the conditions in which cooling towers are desirable or necessary are comparatively rarer than in the less favored and more congested European countries, where these devices have reached the highest state of development.

4 I regret that the author has not presented in exactly the same

form the two tests of the cooling tower described. In Table 4 is given a complete log of the principal observations made at approximately hourly intervals; in Table 5 we have only the average of all the observations made over a period of 24 hours. The two tests were made under such widely different conditions that they afford no proof as to whether the performance of the tower was any better or even as good with its full complement of cooling surface, as it was with only three-fifths of it. During the test with only three-fifths of the cooling surface installed, the average load was nearly 80 per cent greater, and the average quantity of water circulated per hour was nearly 35 per cent greater than on the test with all of the surface installed.

5 Referring to Fig. 11, the indications are that the added cooling surface served no useful purpose. Indeed if the diagram means anything at all, it means that for the same temperature head the product of the heat dissipated per square foot of surface per hour multiplied by the proportion of the cooling surface installed, is practically a constant; also, that for equal temperature heads, the number of degrees cooling is practically the same.

6 In Fig. 12, in which temperature head is plotted against degrees of cooling, the lines corresponding to three-fifths and five-fifths surface, coincide so nearly that one could hardly say that they depart from each other by more than the limit of the normal error of observation.

7 Fig. 13 at first sight seems to indicate that at hot-well temperatures below 120 deg. the cooling was considerably greater with five-fifths than with only three-fifths. But we know that on the test with only three-fifths of the surface, the amount of water circulated was very much greater than with five-fifths surface. Comparisons of this sort are misleading unless the quantity of water circulated per hour and the temperature of the incoming air are the same in both cases.

8 The inconsistency of the curves in Fig. 13 will become apparent if we extend the straight line curve for three-fifths surface until it cuts the line of zero cooling. This will indicate that at a hot-well temperature of a little above 85 deg. the water would not be cooled at all, although we know from Table 4 that the temperature of the incoming air was at no time higher than 35 deg. The inference would be that water entering the tower at a temperature below 85 degrees would be warmed by coming in contact with air at or near the temperature at which water freezes.

9 The indications are that the tower is too small for the work, and

that its performance is limited, not by the amount of cooling surface, but by the weight of air that can pass through it in a given time. After all, it is the air that carries off the heat, and the quantity of air passing through the tower is just as important a factor as is the area of the cooling surface.

10 The tower described occupies 200 square feet of floor space, and is rated at 900 h.p. Assuming 15 lb. of steam per h.p-hr., the tower would have to cool sufficient water to condense 13,500 lb. of steam hourly. A natural-draft tower designed by one of the most experienced builders of this class of apparatus, would for this same duty occupy a space about 29 by 24 ft., or nearly 3½ times as great as that occupied by the towers described. It would also be from 7 to 10 ft. higher, which would give a more powerful draft.

11 Referring again to Fig. 12, it will be seen that the temperature of the water leaving the natural-draft tower is from 40 to 70 deg. above that of the incoming air. On this same diagram are curves which purport to show the performance of the forced-draft tower briefly referred to in Table 3. It would appear from these curves that the forced-draft tower under favorable weather conditions cools the water to within 3 or 4 deg. of the atmospheric temperature. Under unfavorable weather conditions it appears to cool the water to within 15 to 35 deg. of the temperature of the atmosphere.

12 The cost of the forced-draft tower is given as $2.60 per h.p. as against $1.50 for the natural-draft tower. However, if the comparative results as shown in the diagram (Fig. 12) are dependable, it would appear that the forced-draft tower was well worth the additional cost, and a little bit more.

13 In Fig. 7, the author purports to show the maximum temperature of inlet air permissible for various vacua. This diagram really shows the maximum temperature of cooling water to produce a given vacuum on the assumption that we limit the number of pounds of cooling water per pound of steam condensed, to the arbitrary figures set down in the lower right-hand corner of the diagram. The temperature of the atmosphere is not necessarily the limiting temperature to which the water may be cooled. It is well known that with low humidities, cooling towers may reduce the temperature of the water to several degrees below that of the atmosphere. And there is no law of nature that stipulates that we may circulate no more or less than 100 lb. of condensing water per lb. of steam to produce a 28-in. vacuum, or 60, 40 and 30 lb. per lb. of steam to produce respectively vacua of 27, 26, and 25 in.

14 I would point out that the diagram Fig. 1 shows that on two days in June 1906 the *average* temperature exceeded 90 deg. According to Table 1 on the following page there was not a single day during that month on which the *maximum* temperature reached 90 deg., to say nothing of the average. If there were 10 days in the month of June 1906 on which the temperature exceeded 75 deg., it is difficult to see why there must not have been at least as many days on which it exceeded 70 deg. The quantities set down in the columns headed "Average for Month" require some explanation to make them intelligible.

15 The theory of cooling towers is simple, and any one who has a reasonable acquaintance with that branch of natural science which deals with heat, may easily know it a little or even very well. As far as the theory itself is concerned it would be hard to improve on the clear, concise and generally masterly presentation of the subject by F. J. Weiss, inventor of the well-known Weiss condenser, which appeared in a book entitled "Kondensation," published in Germany about ten years ago. But as in all branches of engineering, the coefficients by which theory is reduced to practice are the property of the few who by special application and practical experience have come to know the subject profoundly.

Barton H. Coffey.[1] The advent of the turbine with the high cost of fuel in steam plants and the increasing cost of water for cooling purposes in urban installations of refrigerating apparatus, are making the cooling tower a necessary means of economy.

2 As the author remarks, the literature upon the subject is scanty; in fact, with the exception of C. O. Schmitt's paper before the South African Association of Engineers in 1907, there is scarcely anything extant that I know of, worthy of the name.

3 I do not wholly agree with Mr. Bibbins' presentation of the meteorological conditions to be met by cooling towers, as given in Fig. 1 and Table 1. The comparison of average humidity and temperature, as given by the weather bureau, is a little misleading, as the humidity observations are made at 8 a.m. and 8 p.m. only. In lieu of hourly humidity measurements, I think it better to take the average aqueous pressure at 8 a.m. and 8 p.m., as it is known that this quantity changes slowly, and from this the hourly humidities can be calculated. It will then be found, of course, that as the temperatures advance toward midday, the humidity falls, thus tending to maintain average thermal

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conditions with respect to cooling towers and explaining the approximately uniform results actually obtained. The mean aqueous pressure for July, covering a number of years, works out about as follows:

TABLE 1 MEAN AQUEOUS PRESSURE

	Actual Aqueous Pressure
Boston, Mass	0.542 in. mercury
Philadelphia, Pa	0.614 " "
Salt Lake City, Utah	0.296 " "
St. Louis, Mo	0.648 " "

At St. Louis, therefore, where the mean maximum temperature for July is 88 deg., the relative humidity would be 49 per cent against a mean humidity of 66.1 per cent, as given by the tables, which is distinctly a more favorable condition for cooling towers.

4 While on meteorology, I would like to call attention to the statement in Par. 15*b*, that the tray or atmospheric type of tower cools only by means of "transverse air currents from the side", the obvious deduction being, that without wind this type of tower fails. In fact, in a dead calm the efficiency of all forms of tower falls off, but this condition is of small practical account, as in the interior region the percentage of calm rarely exceeds 2 per cent and on the seaboard is practically unknown. However, in a dead calm the towers still continue to work, due to an ascending column of warm air and aqueous vapor over the tower and a corresponding horizontal inflow of cool dry air. This condition must exist, otherwise the entire space surrounding any tower would become filled with warm saturated air and all cooling would cease. In a forced-draft tower for example, the fan would be simply circulating air having no capacity for absorbing heat. Apropos of this, I have records from an atmospheric tower on refrigerating work for the entire month of September 1907, taken with recording thermometers, in which the cooling water from the tower was maintained at an average of 75 deg., never exceeding 80 deg., with a cooling range of about 10 deg.

5 In Par. 13*b*, among the elements of design, Mr. Bibbins advises, "Avoid free falling water. It should be distributed so as to descend clinging to some form of wetted surface." I would like to know the basis for this statement, as probably by far the largest number of towers in use throughout the world employ the principle of finely divided falling water, as, for instance, the various forms of atmospheric and chimney towers in Europe, South Africa and this country.

6 As 75 to 85 per cent of the cooling is due to evaporation, which can take place only at the surface in contact with the air, the form of cooling surface is of great importance. In a cooling tower with free-falling water, the cooling surfaces consist of the hurdles or decks and the exposed surface of the falling water. Experiments show the weight of a drop of water to be about three-fourths of a grain, the diameter of the corresponding sphere being 0.178 in. A gallon of water properly distributed will therefore expose about 54 sq. ft. of surface. If we know the flow per second and the time of fall in seconds, properly corrected for atmospheric retardation, we can calculate the exposed surface in the water, which, added to the fixed wetted surface, gives the total cooling surface in the tower. The efficiency of the surface in the falling water is greater than the fixed surface, due to the greater velocity of the air relative to the water surface, due to the motion of the drops.

7 The question of type of surface, in my opinion, is one of expediency to be determined by the conditions of operation. Fixed surface is undoubtedly more compact and when skilfully designed opposes less resistance to air currents. On the other hand, it involves weight, greater difficulties in distribution, and where oil is present in cooling water, it becomes coated, the capillarity is destroyed and the water film is reduced to streams, thus greatly lessening the water surface exposed.

8 If the atmospheric form of tower is to be employed, it is hard to conceive of any form of surface, save drops, that would be exposed to the wind from any direction; and where space is available for sufficient surface, the temperature reduction called for can always be attained.

9 In a test by the speaker of an atmospheric tower circulating 440 gal. per min., with air at 93 deg. and humidity 34 per cent, the water was cooled from 80 deg. to 74 deg., or within 3 deg. of the wet bulb, which is the limit of atmospheric cooling.

10 With reference to Mr. Bibbins' remarks on the effect of temperature range on the size of the tower, I beg to submit a few figures on the volume of air required at 80 deg. and 80 per cent humidity to absorb 1000 B.t.u. when the air can be heated to the following final temperatures and saturated:

TABLE 2 VOLUME OF AIR REQUIRED TO ABSORB 1000 B.T.U.

Class of Work	Final Temp. Air	Cu. Ft.
Refrigeration	88 deg	985
Steam Condensing 27 in. vac.	100 "	429
Steam Condensing 26 in. vac.	110 "	267

This shows the enormously increased quantities of air required as the lower ranges of cooling are approached, and also shows the particular advantage of the atmospheric tower for refrigeration work, in saving the power necessary to handle this large volume of air.

11 For example, with air at 80 deg. and 80 per cent humidity, to cool 600 gal. of water per min. to 80 deg., would require about 70,000 cu. ft. of air per min., requiring about 17 brake horsepower in a fan tower. An atmospheric tower of like capacity, having 960-sq. ft. wind exposure, would receive 248,000 cu. ft. of air per min. at a velocity of 4 miles per hour. In steam condensing with a limited space, the forced-draft tower is, of course, the only available type.

Carl George de Laval. The author states that the present high prices constitute the greatest obstacle to the use of cooling towers, and, further appears to give the impression that the cooling tower is a makeshift and not a permanent apparatus.

2 There are three classes of towers, forced-draft, natural-draft and a combination of both, the last-named being used either way, depending on the season of the year. The selection of the type should depend on climatic conditions, cost, etc., a dry climate being best suited for a cooling tower.

3 The author states that the costs range from $4.80 to $6.93 per kw., which appear to be slightly higher than market prices, the reason perhaps being that the author had imposed severe conditions when asking for bids on cooling towers, thereby increasing the costs.

4 Let us assume a plant of 1000-kw., consuming 19,000 lb. of steam per hr., basing the condenser performance upon the ordinary 10-deg. difference in a counter-current jet condenser, and upon a 27-in. vacuum, with air at 70 deg., and 70 per cent relative humidity. A cooling tower with interlocking pipe filling can be built approximately 19 ft. by 35 ft., fitted complete with fans, for about $5 per kw., and a wood-filled tower about 21 ft. by 35 ft, for about $4.50 per kw.

5 The author is correct in stating that installations are not being sufficiently studied, and this, no doubt, is the principal cause for the failure of cooling towers and has prevented a more general adoption of them. It is not sufficient merely to obtain information as to maximum load, steam consumption, maximum temperature and humidity, but it is necessary to know whether these maximum load conditions must be met at the conditions of maximum temperatures and humidity, and if so, for how long a time.

6 Let us assume that bids are asked for a cooling tower for 8000

kw., the conditions named being an air temperature of 75 deg. and 75 per cent humidity, 27-in. vacuum, no time being stated when this load of 8000 kw. is likely to occur, and what its duration is. The real facts may be that this load comes in winter only, and that in summer probably not over 5000 kw. would be required during the evenings, while the summer mid-day load might not be over 2000 or 3000 kw. Under such conditions a tower calculated for a 5000-kw. summer load would be ample for an 8000-kw. winter load, and if the installation was made on the basis of 8000 kw. the year round, the cooling tower would be too large and expensive, and the cost per kilowatt of maximum load would be too great.

7 The maximum mid-day temperature and humidities likewise should not be the basis of consideration with maximum loads, as the electric lighting plant maximum during summer should instead be based upon 8 p.m. temperature and humidities. One sometimes sees the requirement to handle maximum loads at an atmospheric condition of 90 deg. and 80 per cent relative humidity—a condition that may never be reached in the particular locality where the tower is to be installed.

8 Most of the towers described by the author appear to be home-made or makeshift towers, for instance, the tower shown in Fig. 6, installed at Butte, Mont., having a cross-board filling and a wooden stack for natural draft. The design is such that it will lose much of its efficiency as it continues in service, and the boards, as well as the upper stack, will warp, admitting cold air above the filling and tending to kill the draft upon which such a tower depends for its efficiency. The warping of boards will also cause leakage through the sides of the tower, the leakage being carried by any strong breeze, and thrown against surrounding buildings and territory, where during winter it may freeze into a heavy mass.

9 Referring to preceding discussion on the design of towers for maximum atmospheric conditions, one will note in the temperature ranges in Table 2, for the Butte tower, that the atmosphere was over 80 deg. during less than 3 per cent of the total time of the year, so that such conditions can hardly be used as a basis for calculation. Atmospheric conditions at Pittsburg during the four months from May 15 to September 15 average approximately 70 deg. and 70 per cent, which appears to be about a standard basis for cooling towers.

10 Par. 9 refers to the use of cooling towers for handling jacket water of gas engines, the temperatures being about the same as those encountered in ice plants, and higher than in the case of steam con-

densation. Several installations show this temperature to be from 156 deg. to 111 deg., and 130 deg. to 80 deg.

11 Par. 10 and Par. 11 state that the difference between the theoretical steam temperature and the temperature of the circulating water varies from 10 deg. to 50 deg. The usual jet condensers and surface condensers give about 15 deg., and cooling towers for reciprocating engines are usually based on a 24-in. vacuum, with circulating water cooled from 125 deg. to 100 deg. and an air temperature of 70 deg. and 70 per cent relative humidity. Counter-current condensers give about 10-deg. difference, the circulating water being handled under the same conditions of vacuum, with a temperature range from 130 deg. to 105 deg., instead of from 125 deg. to 100 deg., which of course gives an easier condition for the cooling tower.

12 It is a well-known fact that an efficient condenser must be installed in order to get good work from a cooling tower, it being an advantage to the tower to have the temperature of the hot water and the cold water as high as possible. For instance, taking examples of the two conditions, both 1000-kw. plants consuming 19,000 lb. of steam per hr., at 24-in. vacuum, and an air temperature of 70 deg. and 70 per cent relative humidity, one plant being based on a 40-deg. difference between the exhaust steam and the outlet circulating water, which requires the water to be cooled from 100 deg. to 75 deg.; the other plant being based on a 10-deg. difference between the steam and the water, the water being cooled from 130 deg. to 105 deg. In the former case, for the same load, vacuum and air temperature, we require an interlocking pipe-filled tower, 22½ ft. in diameter by 35 ft. high, having four 96-in. fans; whereas in the latter case with only a 10-deg. difference we can do the same work with the tower 13 ft. 6 in. in diameter by 35 ft. high, having one 120-in. fan. The efficiency of the condenser therefore makes a very decided difference in the size of cooling tower.

13 Under *c* in Par. 13, the author apparently refers principally to towers with wood sides, having a wood structure within the outside boarding. It is very important that the filling must come close to the side of the tower. Particular care should be taken in erecting towers to see that pipes are first laid around the outside edge as closely to it as possible; otherwise, there will be a short circuit of cold air around the side of the tower and a loss of efficiency. This condition, while bad enough in the forced-draft tower, is much worse in towers of natural-draft type, because this air will seriously reduce the draft by mixing with and cooling above the filling the heated air upon which the draft depends.

14 As to height of working section, it is true, as the author states, that the height is important, and the distance of the elevation of the water should be kept as low as possible. A pipe-filled tower is 13 ft. 4 in. deep with a drop at the bottom of from 6 ft. to 11 ft., according to the size of the tower. With a distributor operating head of 5 ft., this gives the largest towers a maximum pumping head, plus friction in the piping, of 29 ft. 4 in. against approximately 38 ft. as required in the experimental natural-draft tower at Detroit, shown in Fig. 9. The horsepower necessary to pump the water the additional 8 ft. 8 in. in height will offset the usual fan horsepowers, making a natural-draft tower of this type more expensive to operate than a fan-draft tower.

15 As to the mat of wood swelling and being thrown out of place, I would state that towers have been built with a cross-board wood filling, and four of these have been in satisfactory operation since 1904. In these towers were used 2 in. by 2 in. verticals at intervals through the filling, with the boards nailed so as to hold the filling in place and prevent distortion or formation of large open gaps through warping of the filling.

16 The cooling towers illustrated in Fig. 3, are furnished with perforated pans and have free-falling water, the sides being screened. This tower depends for its efficiency upon a cross breeze and is very inefficient in still air as the air cannot rise within the tower on account of the pans. A strong breeze will blow most of the water out through the sides of the cooling tower, in spite of the screen. The tower shown in Fig. 4 occupies considerable space, and also requires additional space in the immediate vicinity because of loss of water through windage. The tower illustrated in Fig. 5 is evidently much less efficient than that in Fig. 4, because of the large amount of free-falling water. The free-falling or splashing of water is a very inefficient method of cooling. Water for proper cooling should always be brought down in contact with a surface so that it will descend slowly and thus have close and intimate contact with surrounding air.

17 In Par. 26, the author gives the total cost of the Detroit tower, erected complete and including filling, distributing pipes, foundations, etc., at $1350. It appears that the steel work, if made of at least No. 10 gage, would weigh approximately 20,000 lb., which at 6½ cents per pound, which is about as low a rate as mentioned, would require an expenditure of $1300 for steel work alone. The lath filling and the work of assembling and installing this tower, would cost about $400; the timber supports, distributing piping, etc.,

about $250; concrete foundations an additional $250; or a total cost of $2200. Assuming a load of 1000 h.p. with 19,000 lb. of steam per hr., vacuum 24 in., with a temperature difference of 10 deg., the circulating water being cooled from 130 deg. to 105 deg., air at 70 deg. and 70 per cent humidity, a pipe-filled cooling tower of the fan type, measuring 13 ft. 6 in. by 35 ft. could be installed for about $2500.

18 The results of the test given in Table 5, with atmospheric temperatures of from 18½ deg. to 30 deg., are not complete for a natural-draft tower, as such towers fall off in efficiency very rapidly when the air temperature is raised. The results at temperatures from 70 to 80 deg., would not be so favorable.

19 In Par. 30, the condition of scale covering the wooden filling would be experienced in any tower, and is usually encountered where well water is used to make up in cooling towers for refrigerating plants. The scale forms a protecting coating in a pipe tower and prevents possible rusting of the pipe filling.

20 In Par. 32, the author refers to possible advantages of a slotted pipe as compared with spouts on a distributor arm, in regard to clogging. The spouts used by some first-class designers are 1 in. in diameter and are consequently much less liable to clog than are pipes having a ⅛-in. slot in the top.

21 In Par. 15, the author refers to the use of sprays over a pond. This seems a very simple apparatus, but it must be realized that the sprays require from 15 to 20 lb. pressure at the nozzle and so consume more power than required for circulation through a cooling tower of the fan type, and in most cases as much power as is required both for the circulating of the water and for the driving of the fan.

22 The arrangement of cascade or cooling sprays on a roof as described by the author is not new. The installation was in use by J. H. Stut of San Francisco, previous to 1892, being placed upon the roof of a factory. Galvanized troughs, 5-ft. wide were arranged in tiers on a slight incline so that the water traveled back and forth a distance of about 2000 ft. before being returned to the condenser. An arrangement of falling from one trough to another, these troughs being spread out upon a roof, was used at the old Budweiser Brewery in Brooklyn previous to 1890. The sprays and roof troughs are alike open to the objection that if there is a strong breeze the water is carried all over the surrounding neighborhood and if there is no breeze, a heavy fog quickly collects at the point of spray and thus greatly reduces the amount of cooling.

23 Referring to various types of filling illustrated in Figs. 8*a* to 8*g*, Fig. 8*a* offers too serious an obstruction to the draft within the tower, closing more than 40 per cent of the space necessary for vertical circulation of air, as against 3 per cent covered by interlocking pipe filling or 25 per cent by wood filling. The cascades as illustrated must fall as shown in the sketch in order to operate efficiently, that is, the water must strike the pans on the next lower section of the filling; but this they will do only if the amount of water supplied is practically constant, otherwise it is liable to spill over several rows of filling, and result in quick descent and consequent loss of efficiency.

24 The filling illustrated in Fig. 8*b* is that used by Henry W. Bulkley, and depends upon a cross steam of air, as in the tower shown in Fig. 3. It is open to exactly the same objection as the latter tower.

25 The filling 8*c* will cause large quantities of free-falling water between the several courses and will result in inefficient operation. The filling 8*e*′, a wooden cross-board type, is apparently good. It requires additional expense in placing, but evidently will save something in fan horsepower. The filling 8*d* offers a bad obstruction to the draft on account of deflecting the air alternately to the right and left. The water also will evidently flow down the top side of the board, whereas the air impinges most strongly against the lower side of the board.

26 The filling 8*b* is the same as 8*b*′ and is good. The filling 8*g* is open to the objection of having no redistribution,—the water distributed at the top, however unequal it may be, must remain unequal from top to bottom. The filling 8*f* has one redistribution at the center. Otherwise it is open to the same objection as 8*g*.

27 In Par. 23, and also in the footnote, it is stated that ball bearings are difficult to keep in good condition. Ball bearings are not used in modern towers, a floating water-step bearing being used instead.

28 Referring to Par. 41 and Par. 42, a combination type tower may run with natural draft about eight months during the year. At a plant in Newark, N. J., a combination type tower with side doors operates on an 800-kw. load nine months of the year with natural draft, and requires 25 h.p. during the remaining three months of the year.

29 As to Par. 46, efficient condensers are more badly needed than efficient cooling towers. Cooling towers have reached as high an efficiency as can be expected, but most plants now operating with direct jet condensers delivering into the towers, could obtain much

higher vacuum or handle greater loads at the present vacuum, if condensers of the counter-current jet type, or the more efficient baffled-surface condensers were substituted for the condensers originally furnished.

30 As to Par. 48, the temperatures are practically the same as in ice plants. In order to get the temperature head mentioned, it is more economical to circulate the water for the ice plant first through the ammonia condenser and then through the steam condenser delivering to the cooling tower, than to have two towers handling separately the water of the ammonia condenser and that of the steam condenser.

31 The open wooden towers referred to in Par. 50 are not restricted to points of low humidity; but as already mentioned, they require much open ground, not only on account of their size, but also for wind effect, and that surrounding buildings may not be drenched with water blown from the tower.

32 As to Par. 51, the tower best adapted to natural-draft work is the one which offers the least resistance to the ascending current of air. In Par. 52, no temperatures are given to substantiate the statement of heat dissipation by lath mat construction.

33 As to Par. 53, one cannot endorse the fan booster or induction type when combination towers can be made that will give better results and that are surely preferable for overload conditions.

34 The largest number of towers in this country are of the forced-draft type, while European practice tends towards natural-draft towers. It is thus apparent that there can be no standard of type or size, because of difference in climates; each installation must be considered as a separate problem.

E. D. Dreyfus. In Par. 10, Mr. Bibbins says, "But in practice from 10 to 15 deg. difference exists, depending upon the type of condenser and the volumetric ratio of water to steam." I wish to supplement this by adding that it does not depend altogether on the volumetric ratio. Another important factor is the effectiveness of air removal. Lower vacuum makes it possible to operate with a diminished volumetric ratio as the temperature rise is increased.

2 Exception is to be taken to Mr. Foran's remark that perfect condenser operation entails much greater experience, which might be implied as generally applicable. This is true only of surface condensers. In cooling-tower practice, the conditions are extremely favorable to the use of the more simple jet type. The more efficiently

this latter type is operated, that is, the nearer the discharge water is brought to the temperature of the exhaust steam, the smaller is the volume of water necessary, since volume and temperature rise are component factors of the B.t.u. extraction. Therefore, with less volume of water handled, the size of the condenser may be reduced and consequently furnished at a smaller cost.

3 A remark made by the author in presenting the paper, that an inefficient condenser and an efficient cooling tower go hand in hand, bears further explanation, although the statement was somewhat modified subsequently. With an inefficient condenser, the vacuum is not likely to be very good, and therefore, with the higher temperature prevailing in the condenser, the water might pass to the tower at a higher temperature, making it easier for the latter apparatus to perform its work. On the other hand, the statements might be applied with equal, if not greater, force to efficient condensers which are able, for the same condensation, to create higher vacua, besides heating the discharge water up to the same final temperature head as the inefficient type, there being little or no terminal difference in an efficient design at its normal capacity. Moreover, considering the benefit accruing to the prime mover, a smaller volume of water may be used and worked at the same temperature as in the inefficient type of condenser, thus increasing the possibility of the tower. I would qualify the above statement to the extent that it deals with a comparison of condensers designed for the same vacuum, and evidently would not hold for a case where a very poor vacuum was admissible.

4 It might be well to state here that a near approach to the theoretical vacuum is not an impossible condition in actual operation. This implies, of course, that the character of the condenser design, the counter-current type with an efficient air pump, fulfills the requirements. In a test which I conducted last fall on a 1000-kw. low-pressure turbine equipped with a counter-current jet condenser, the following results were obtained: At three-fourths load with 83 deg. injection water, a vacuum of 28.20 in. (30 in. barometer) was maintained, and the water left the condenser at a temperature of 96.8 deg. The temperature corresponding to the vacuum was 97.6 deg., giving practically one degree terminal difference.

5 I have observed that temperatures of the water leaving the tower were several degrees colder than the atmospheric temperature in warm weather, the difference being as much as ten degrees at times.

6 With the increasing recognition the cooling tower is receiving,

it would be desirable to have the Society define a standard basis of measuring the efficiency of the apparatus. There is a conspicuous lack of harmony of opinions as to what constitutes the governing characteristics of tower performance.

T. C. McBride. In the earlier parts of the paper the author would lead us to believe that cooling towers have not received the scientific attention warranted. Reference to the literature on this subject and the work that is being done hardly confirms this statement. A considerable number of manufacturers have for some years past been supplying cooling towers designed on scientific lines, and the proposals submitted by them, particularly on fan-type towers, are intelligently framed and leave no points whatever open to guess work.

2 The paper very properly calls attention to the intimate relationship of condenser efficiency to cooling-tower performance, but in doing so is extremely unfair to the condenser—in fact, in speaking of different types of air pump, the author almost leads us to believe that some are so superior to others that the vacuum they create is of a kind superior to that created by other air pumps.

3 Condenser engineers now agree that the efficiency of condensers, with regard to the comparison of discharge-water temperature with theoretical vacuum temperature, is as much a question of the average temperature of the vapor in the condenser as its design. The average temperature of the condenser is necessarily determined by the amount of air present therein, and is a direct function of the ratio of the air-removing capacity of the air pump and the volume of air reaching the condenser with the steam. The merit of the air pump cannot therefore be determined either from the vacuum obtained or from the relation of the discharge-water temperature to the theoretical vacuum temperature, but is wholly a question of the capacity of the air pump to handle air at the least expenditure for power, maintenance, interest on first cost and depreciation.

4 It is true that the question of condenser efficiency and air-pump efficiency is somewhat involved with that feature of condenser design having to do with the reduction of air-pump suction temperature, but as all condenser designs should take care of this feature it may be eliminated from the comparison of types of condensers or types of air pumps. It is conceded that the author's division of condensers and air pumps into good, indifferent and bad classes, in accordance with the vacuum and discharge-water temperature obtained, follows lines which have been generally accepted in the

past; but a view from an engineering standpoint must consider the impurities in the steam in the shape of air and non-condensable vapors, before judging any particular type of condenser or air pump.

THE AUTHOR is exceedingly grateful for the interest shown in the paper and the practical nature of the discussion, which has served to clear up several ambiguities and to extend the subject into channels of inquiry representative of everyday commercial problems.

2 Mr. Ennis deprecates the loss by windage of considerable volumes of circulating water, in excess of that supplied by condensed steam. Theoretically, without windage loss, there should be practically no make-up water required, as an exact thermal balance has been established. But this loss does occur in both forced-draft and open-tray type towers, and often to a serious extent. However this is simply a point in favor of the closed natural-draft type of tower, in which the velocities are reasonably low and hence small tendency exists to abstract water from the cycle.

3 The high loss in the Duquesne Lighting Company plant, it should be explained, is not due to windage. The hot jacket water can only be partially cooled, consequently enough must be thrown away to lower the temperature by the addition of fresh cold water. The loss at Potosina, however, was entirely due to windage.

4 The curves in Fig. 7 may very possibly be slightly in error, as they were necessarily based upon arbitrary assumptions—hence no attempt was made at absolute accuracy.

5 Mr. Foran evidently has had in mind the surface condenser in discussing possible and probable temperature differentials, whereas the author has referred more particularly to the barometric or jet types, especially in Fig. 7. This should have been stated more clearly in the paper. Generally speaking, it is possible with the jet type to work with much lower differentials than with the surface type. Mr. Foran's deductions regarding the extent of surface required to meet special conditions are therefore entirely proper. This very difficulty which is experienced with surface condensers in meeting the conditions imposed by the best cooling-tower practice, only emphasizes in the author's mind the inherent advantages of the jet types.

6 The term "fixed cooling-tower performance" could not apply to the construction of the curves in Fig. 7, as it is here used in the sense of efficiency rather than size. The use of "performance" here was in reference to relative cooling effect (deg. fahr.)—not capacity for absorbing heat—for the sake of eliminating another variable

in the construction of Fig. 7. The size or capacity for a given condition is simply a function of a heat quantity (B.t.u.) absorbed from the exhaust steam. For a given type of surface and draft velocity, the rate of absorption is fairly constant—a parallel to the constant rate of heat transmission through the tubes, as cited by Mr. Foran.

7 In reference to the Detroit tests, Table 5, it should be noted that the condensing plant was not well adapted to the work in view, being an equipment temporarily retained in service from an old plant, too limited in surface and without means of operating air and water pumps individually, as required for economical working. The poor results from this particular plant were therefore distinctly attributable to the temporary nature of the installation, and not to an inherent fault in the type itself, as might be gathered from the reports.

8 In his closing remarks, Mr. Foran seems to confine the use of "natural-draft tower" to the open-tray type. It is quite true that this has no application where large capacities or the highest efficiency are necessary. The closed chimney type is not dependent to any extent upon lateral wind velocity, and may be designed to economize space effectively.

9 The point raised by Mr. Dreyfus in regard to the effect of low temperature differentials is well taken. The author's observation that poor vacuum and good cooling go hand in hand applies to a given equipment, but the highly efficient condenser with low differential of course finds the most direct application.

10 The author did not observe or infer that the cooling-tower field remains comparatively unexplored, but that certain conditions have tended to render the subject a closed book. This is not the case with engines, turbines, boilers, condensers, etc., so the fact that this condition obtains with cooling towers is not readily justifiable.

11 The two series of tests could not be presented in identical form, as the data were not available in such form. However, the curves, Figs. 11, 12 and 13, were drawn up to facilitate comparison. The first test covered day and peak loads only; the second, the entire 24 hours,—hence a low average load, as Mr. Longwell observes. Because the tower shows a low rate of heat dissipation with the entire surface installed, it should not be inferred that the actual work done was proportionately lower. Considering abscissae (B.t.u.) as equivalent to load (kw.) it must be apparent that for the same load a much higher cooling effect was obtained with the cooling surface complete.

12 For equal temperature heads, the cooling is bound to be the same except when the "lost head" differs, as it does slightly in Fig. 12. This opens up an extremely interesting line of inquiry—a survey

of rates of heat dissipation and humidity in each successive zone of the tower. Which part of the tower does the most work? Assuming air to be discharged exactly saturated at the temperature of exit, what spacing of mats is correct to produce a proper gradation of humidity from, say 70 per cent at entrance to 100 per cent at exit?

13 Regarding the inconsistency of Fig. 13, Mr. Longwell has forgotten to reckon the "lost head" shown in Fig. 12—approximately 40 deg. There is thus a very small discrepancy. However, it is hardly safe to interpolate in such a case. It is already pointed out in the paper that the tower is working at a disadvantage, owing to the extremely poor condenser performance, that imposes an extra burden on the prime mover as well.

14 The circulating water ratios adopted as a basis of the curves in Fig. 7, were so adopted to approximate average practice, otherwise a "family" of curves would replace each single curve shown.

15 Mr. Coffey favors the use of vapor pressure in lieu of relative humidity. The author entirely agrees to this method as more scientific. However, absolute humidity expressed in grams per cubic foot perhaps has a more direct bearing on cooling tower work.

16 The suggestion "to avoid free-falling water" should have been amplified in the paper, and Mr. Coffey justly directs attention to it. Compactness or maximum duty for a given size is so essential in restricted locations that the atmospheric type is handicapped, if not debarred, which he himself recognizes in the closing sentence. The paper is directed entirely along these lines of maximum duty, and especially toward the development of the natural-draft type.

17 Mr. de Laval advances the argument that a tower should not have to be designed, rated and purchased *entirely* on a peak load basis. This is entirely in agreement with the author's object in presenting the combined natural-forced-draft tower with fan auxiliary for use only during peak loads or during bad weather.

18 The objections of Mr. de Laval to the construction of the Butte tower are, however, not well taken, as the construction is more substantial than as described by him, and several years' service has not developed the defects he mentions.

19 The tests made at Detroit occurred, it is true, during the colder season, but in Par. 29 it is stated that the tower showed very little difference in operation in winter or summer—this on the advice of the chief operator.

20 Tables 4 and 5 present the temperatures asked for to substantiate the assertion of a safe rate of heat dissipation of 200 B.t.u. per square foot per hour for the lath mat construction.

No. 1260

GOVERNING ROLLING MILL ENGINES

By W. P. Caine, Ensley, Ala.

Associate Member of the Society

In considering the conservation of steam-power equipment for driving rolling mills, we must take into account the two methods of rolling: the two-high mill driven by a reversing engine, and the three-high mill driven continuously in one direction; and the relative amount of power required for each.

2 There is very little variation in the type used for each class of mill. Twin engines are used for two-high mills and single engines for three-high mills, usually tandem compounds.

3 The reversing engine for the two-high mill must be powerful enough to take care of the engine and mill friction and the maximum torque produced by the piece in the rolls in any position. As these engines are usually twin engines with cranks at 90 deg., each side must be capable of doing the work alone when the other side is on the dead center.

4 In determining the size and distribution of the metal in engines of this type it is the custom to make the dimensions a little larger and the parts a little heavier than formerly for the same work. Reciprocating parts are made heavier to stand the shocks, thereby increasing their inertia, and making necessary heavier frames, bed-plates, bearings and pins, as well as more rigid adjustments, which in turn require more attention.

5 As an example of the power sometimes used for an engine of this type, a certain engine may be cited which was fully capable of delivering 25,000 h.p. while the actual average work on the steel passing through the mill could not have required more than 2000 h.p. at the maximum capacity of the mill. The engine and mill friction, if the mill were driven continually in one direction, would not fall much short of 1000 h.p. Assuming that 500 h. p. would be required for the

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reversals, the total average work of the engine would thus be about 3500 h.p., or less than one-seventh of its capacity.

6 As the three-high mill is driven continually in one direction, the energy stored in the flywheel makes it possible to do the same work with considerably less than one-half the maximum power required in the former case, the amount depending upon the size and weight of the flywheel. The greater the amount of energy the wheel can store up, the closer can the maximum power required approach the average work of the mill, resulting in the more economical use of steam and a lower cost of equipment.

7 Mill designers do not always give sufficient consideration to this fact and operators have to deal later with high steam cost and difficulty in keeping the proper steam pressure. Of course, there are other features to be considered, but economical use of power is a very important item.

8 For driving a three-high mill a twin engine of the cross-compound type could be used with an intercepting valve such as is employed in locomotive practice, by which the engine could be started from any position and handled by a quick-acting throttle valve, so that it could be brought to a standstill as soon as a piece passed through the rolls, if another were not ready to enter the mill. Such an arrangement would go a long way toward answering one of the principal arguments in favor of the two-high mill: that its engine uses steam only when the piece is on the mill.

9 If an engine of the type described be furnished with a very heavy flywheel located between the engine and the mill, the shocks due to the piece striking the rolls will be taken very largely by the flywheel and not by the engine. Furthermore, if the engine were so designed that it could not work through the wide range of steam admission, as is the current practice, the abnormal amount of compression now necessary would be cut down to a large extent, the parts of the engine would be strained less, and the engine would run with greater steam economy owing to the cutting out of the high release pressure during heavy work and the reduction of the number of strokes during the period of negative work.

10 Going into the details of operation of an engine driving a three-high mill, we find that the engine first develops just enough power to take care of the friction of engine and mill; next, the piece strikes the rolls; and third, the piece leaves the rolls.

11 If the engine were running at a constant speed during this period, just enough steam would be admitted to the cylinder at each

revolution to do the work with the least possible variation of cut-off, resulting in the most economical use of the steam. The more the speed varies the greater the amount of steam required per horsepower developed.

12 A constant speed is also desirable for another reason; namely, the available energy stored in the flywheel is always normal under these conditions, whereas with a varying speed it would be below the normal about one-half the time. Should the piece strike the rolls when the steam pressure is low and the steel cold, the engine would be more liable to stall if the stored energy were below normal than if it remained constant.

13 The initial force of this blow is absorbed by the flywheel and the speed of the engine is reduced in consequence. When this has dropped, say four or five revolutions, the governor has probably so adjusted the steam valves that the engine is developing its maximum power and the valves will remain in this adjustment until the engine is nearly up to speed again. During this time the release pressure will be high, making it neccssary to carry a very high compression, conditions under which a non-condensing engine will make the most noise. Further, if the steam going to waste were utilized the engine would be capable of doing about one-third more work.

14 On many passes the engine is receiving the maximum amount of steam at the instant the piece leaves the rolls and the flywheel absorbs energy through the increase of speed above normal. In the writer's opinion, this is the time when nearly all of the failures of flywheels on rolling mill engines occur, and anything that can be done to cut down the amount of energy to be absorbed by the wheel at this time will increase the safety of the engine and decrease the repairs on the engine and mill.

15 Having in most cases reached the highest point of speed, the governor will shut off steam entirely from the cylinder, the engine will slow down to several revolutions below normal speed before sufficient steam will be admitted to increase this speed again and the result will be a wide range of the speed variation during the first period when the engine has simply to overcome its own friction and that of the mill.

16 The operations outlined in the foregoing are very complicated when two or more passes occur at the same time.

17 The results of a study of the above details of operation of the No. 1 rail mill engine driving the four-pass roughing rolls at the Ensley Rail Mill of the Tennessee Coal, Iron & Railroad Company, during

a series of tests show how great a difference in the performance of the engine the writer was able to obtain with one adjusting screw added to the governor.

18 The engine is a Reynolds Corliss, 52 in. by 72 in. non-condensing, equipped with a long-range cut-off valve gear.

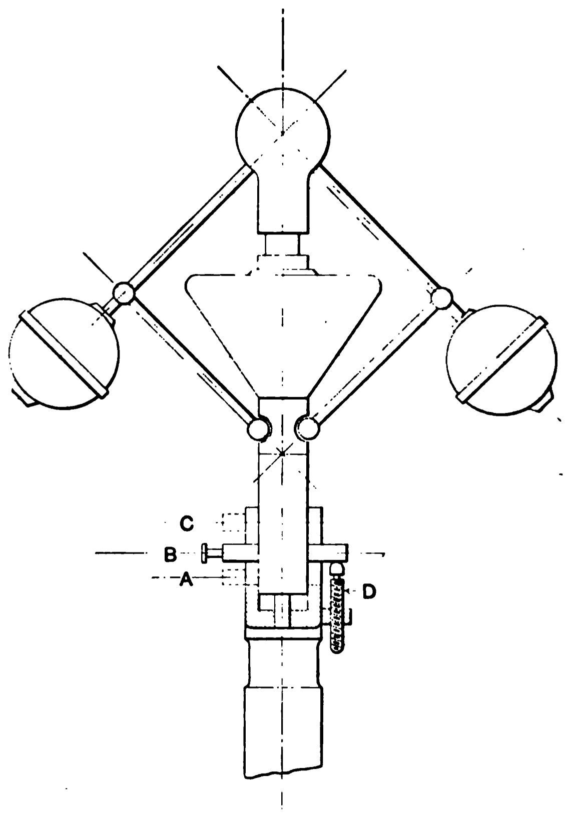

Fig. 1 Showing Location and Function of the Adjusting Screw *D* on the Governor

19 Fig. 1 shows the location of the adjusting screw *D* on the governor. Its purpose is to prevent the governor from dropping to position *A* which would allow the maximum amount of steam to reach the cylinder, as determined by the valve gear. In this case the steam was admitted nearly three-quarters of the stroke with the

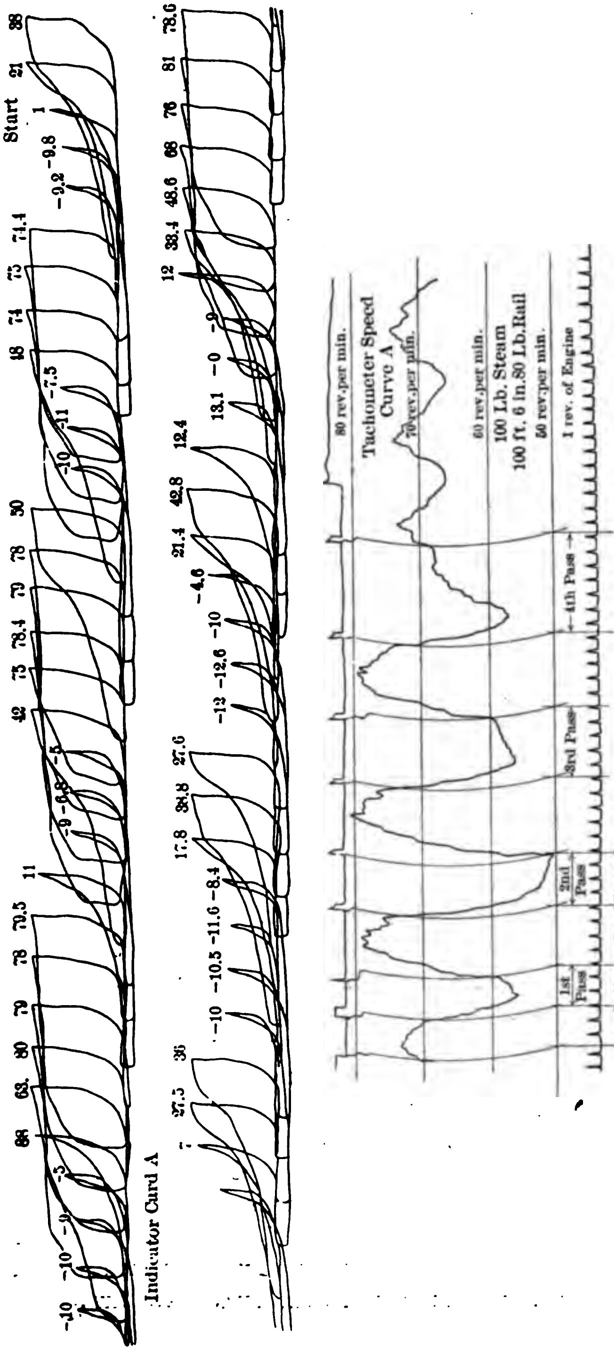

FIG 2 CONTINUOUS INDICATOR CARD A AND TACHOMETER SPEED CURVE A OF NO. 1 RAIL MILL ENGINE, 52 IN. BY 72 IN.

THE CARD AND CURVE WERE TAKEN AT THE SAME TIME AND SHOW ORDINARY PERFORMANCE. THE LOWER HALF OF INDICATOR CARD IS A CONTINUATION OF THE UPPER HALF.

governor in this position. When the pin is in position C the steam is entirely cut off. The screw D was adjusted very slowly while the engine was under load to determine the most advisable position B.

20 This position must be at a point such that the engine will carry an average load and also will not stop under a heavy load. It can readily be seen that by reducing the range of adjustment of steam distribution the engine will operate with more economical steam consumption and the greatest strains on the engine and mill will be reduced.

21 The degree of success attained by this adjustment may be judged somewhat by the accompanying continuous indicator cards and tachometer speed curves. Card and curve marked A (Fig. 2) were taken together before the attachment was used. Card and curve B (Fig. 3) were taken when the attachment was in use.

22 The speed curves were made by a recording tachometer which the writer rigged up from a Schaeffer & Budenberg indicating tachometer.

23 This indicating instrument was mounted on a plate, a bevel gear being fastened to the end of its driving shaft, which in turn drives a shaft with a worm at one end. The worm shaft is mounted on a bracket which will swing out of gear, thus disconnecting the indicating instrument if desirable. The worm wheel driven by the worm shaft runs loose on one of the paper-feed rolls and when a record is to be taken a small clutch causes the worm wheel to drive the rolls. Two rolls are geared together and a third acts as a press roll. The paper rolls used were such as are furnished for the Uehling pneumatic pyrometer.

24 The indicating needle on the original tachometer was replaced by a longer one which reached to the paper. A pencil was attached to the end of the new needle but there was so much friction that the records were of no value, so a small tin funnel was fastened to the needle and a linen thread passed through the hole in the bottom, protruding about $\frac{1}{8}$ in. The ink in the funnel worked down the thread to the paper and made a satisfactory record.

25 Below the paper-supporting plate is a vertical plate to which are fastened two electric bells with the gongs removed and pencils substituted for the clappers. One of these bells was operated by a contact made once in a revolution of the engine, the record being shown at the bottom of the curve A (Fig. 2). As it was very evident that the paper would always feed the same amount at each revolution it was not considered necessary to use this device each time.

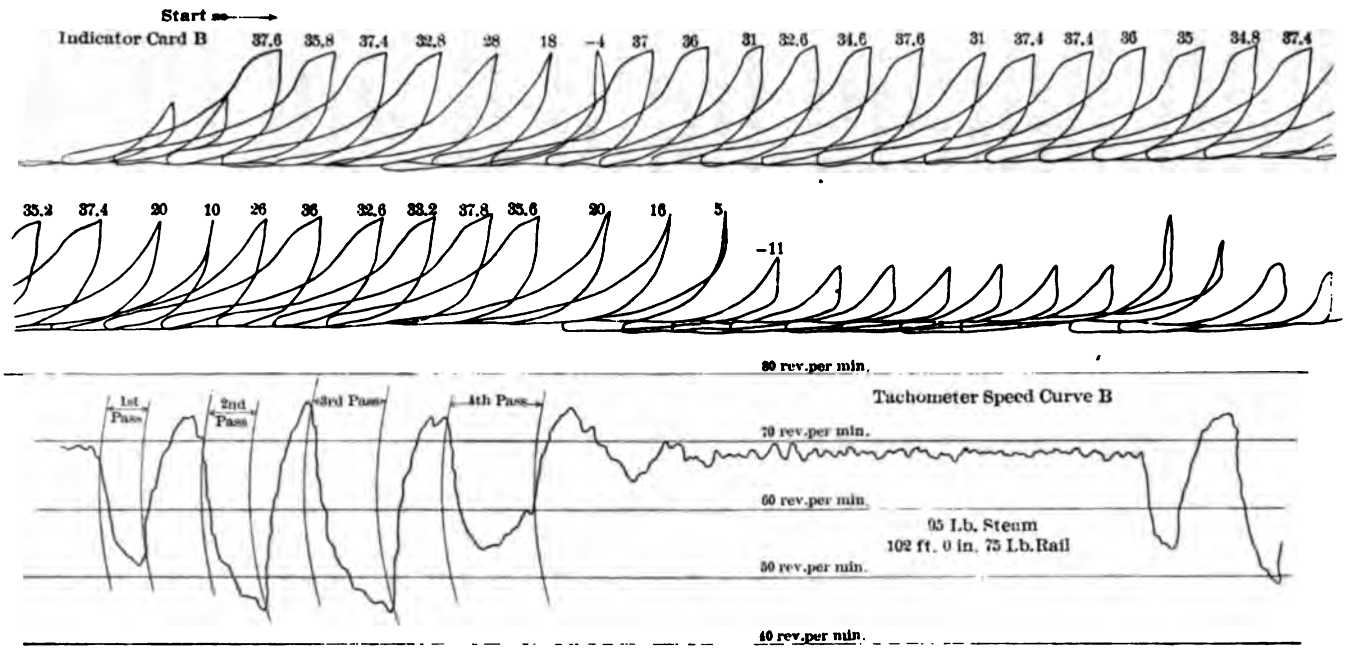

Fig. 3 Continuous Indicator Card *B* and Tachometer Speed Curve *B* taken when Running with the Caine Method of Engine Control

The lower half of the indicator card is a continuation of the upper half

26 The other bell was operated on a circuit that had two gaps in series, one of which was closed when the indicator cards were started and the other kept closed except at the instant an observer at the rolls indicated the start and stop of the various passes, by momentarily breaking the circuit by means of a push button. This last feature also appears unnecessary, as the speed curves have a pronounced change of direction at these instants.

27 From card A it will be noted that during the four passes there are 37 records showing that the engine is taking steam, and 22 records showing that it is not. These would lead one to think that if every card were a positive one, and the work were distributed throughout the entire period between pieces, and the mean effective pressure averaged, there would be a more economical use of steam, and possibly with a heavy flywheel the size of cylinder could be reduced. That would be the ideal condition, which cannot be realized, however, because of three changing functions: the varying time between pieces, the varying temperature of the steel and the varying steam pressure.

28 Card B, Fig. 3, shows the engine doing the same work as before, on a piece of the same length, and it can be seen that the work is distributed over 34 revolutions and only two negative cards. With the engine running as this card shows, it is a comparatively easy proposition to set the valves for economical steam distribution and it is also much easier to keep the rods and boxes properly adjusted. The low terminal pressure is the cause of the engine's running much more quietly than when card A was taken.

29 The indicated steam consumption of card A is about 43 lb. of steam per h.p. per hr., against 37 lb. in card B, a saving during rolling periods of over 20 per cent.

30 Of the speed curves, curve A shows that during the friction load the engine varies from 66 to 73 r.p.m., with an average of about 69 revolutions, and after the passes the speed becomes about 80 r.p.m., an increase of 11 revolutions above normal. Curve A shows also that the second pass is the heaviest one of the four, the speed dropping to 51 r.p.m.

31 Curve B indicates that during friction load the speed varies only about 3 r.p.m., and that the highest velocity is 75 r.p.m., or 7 above normal. Some changes were made on the rolls between the two records so that the third pass was as heavy as the second and the speed drops to about 45 r.p.m. These curves indicate that the engine could be speeded up to about 75 r.p.m. and not exceed the speeds

TABLE OF DATA FOR SUCCESSIVE PASSES

No. 1 Rail Mill Engine, E. P. Allis, 52 in. × 72, in. 4 Roughing Passes, Rolls 27$\frac{3}{8}$-in. Fitch Diameter, Product 100 ft. 6 in., 80-lb. Rail, Total Weight, 2680 lb.

		First Pass	Intervals	Second Pass	Intervals	Third Pass	Intervals	Fourth Pass	Intervals	Total
1	Area section after pass, sq. in.	54		33.5		22.9		19		
2	Length section after pass	15.95		25.7		37.6		45.2		
3	Revolutions	2.23	4.77	3.59	5.41	5.25	5.25	6.31	4.19	37
4	Speed at start } from curve	70.0	58	76	51	76.5	59	74	68	
5	Speed at finish } from curve	58.0	76	51	76.5	59.0	74	68	69	
6	Total m.e.p., pass and intervals	461.7		702.6		719.5		677.2		
7	Total energy pass and intervals, ft. lbs.*	5,755,000		8,758,000		8,968,000		8,440,000		
8	Total energy, ft. lbs.*	2,602,200	3,152,800	4,573,500	4,184,500	6,276,300	2,691,700	8,008,800	431,200	
9	Energy taken from flywheel, 1180 $(n^2_1 - n^2_2)$	1,812,500		3,747,000		2,798,000		1,005,000		
10	Energy added to flywheel		2,846,000		3,836,500		2,354,000		161,700	
11	Energy, total	4,414,700		8,320,500		9,074,300		9,013,800		
12	Energy in friction, 64,320 ft. lb. per rev.	143,400	306,800	230,900	348,000	337,700	337,700	406,000	269,500	2,380,000
13	Energy, total, less friction in piece	4,271,300		8,089,600		8,736,600		8,607,800		29,705,300
14	Torque per foot length of piece	268,000		315,000		232,000		191,000		

*From cylinders 12,464 × m.e.p.

used before, that more energy was stored in the flywheel and that the engine would not drop below the speed shown on Curve *A*.

32 Referring to the table, the constant used in items 7 and 8 is the foot-pounds of work for one stroke of the engine at 1-lb. mean effective pressure.

33 In the original calculation for item 9 only the flywheel was considered. This is 22 ft. in diameter, weighs about 130,000 lb. and has a radius of gyration of about 7.4 ft. Taking the formula

$$\text{energy} = \frac{Wv^2}{2g}$$

and altering it to get the energy stored up or given out at a change in velocity of one revolution per minute it becomes

$$E \text{ for 1 r.p.m.} = \frac{\text{wt. wheel} \left(\dfrac{\text{radius gyration} \times 2\pi}{60 \text{ seconds}} \right)^2}{64.32}$$

Substituting and solving with the values given above

$$E \text{ for 1 r.p.m.} = 1220 \text{ ft-lb.}$$

As the energy varies as the square of the velocity we would use the following to represent the amount of energy involved in a change of velocity

$$E = 1220\ n_1^2 - n_2^2$$

in which n_1 is the higher number of revolutions of the engine and n_2 the lower number. In checking this up against indicator card and speed curve results on friction alone, it became evident that 1180 was the proper constant to use to include the inertia effects of the mill and reciprocating parts.

34 The constant in item 12 was the average foot-pounds of work per revolution during the friction period as calculated from the indicator cards. The slide rule was used in making the calculations.

DISCUSSION

Henry C. Ord. The conservation of energy as applied to rolling mills has received very little attention until during the past five or six years. The power required to roll a given piece was not known until the continuous indicator and recording tachometer were applied.

The cards from these instruments furnished records from which the conditions for any stage of the operation could be calculated, giving complete information as to the variation in power and speed for different conditions and classes of work.

2 Rolling mill engineers have several reasons for preferring the two-high mill. As Mr. Caine says: "The engine uses steam only when the piece is on the mill." As there is considerable time between pieces in some classes of work, this is an important item. Should a piece not enter properly and stick in the rolls, thus stalling the engine, it is easy to reverse and back out the piece. This condition with a three-high mill would cause considerable trouble and delay. This is the reason why some of the modern three-high electric-motor-driven rolls are fitted with an emergency reversing device. The reversing feature can also be used as a quick safety-stop in case of accident.

3 The two-high mills are not so complicated as the three-high mills, and they have less rolls and no reversing mechanism for raising and lowering the table. In considering the two systems, this is an item of power that should be charged to the three-high mill. However, power is not the only consideration; it is usually a question of the maximum tonnage in minimum time with the least amount of power.

4 The 25,000 h.p. engine Mr. Caine refers to is a 42 in. and 70 in. by 54 in. twin tandem horizontal compound-condensing blooming-mill engine, designed by the writer about four years ago, and built by the Allis-Chalmers Company for a blooming mill requiring an average of about 6000 h.p., which is also about the economical load for the engine. It was designed for a maximum of 25,000 h.p. under the following conditions: steam pressure, 150-lb. gage; cut-off, ¾-stroke; vacuum 25 in. referred to 30-in. barometer; r.p.m. 200. This machine has been described as "the world's most powerful engine." I believe the piston speed, 1800 ft. per min., is the world's record.

5 If the engine Mr. Caine has experimented on was tested under the same conditions as regards pressure and work, with and without the function of the adjusting screw, we would expect different results than those shown. The controlling device has no control over the engine before the load is increased, until the speed falls to that fixed by the adjusting screw. At this speed and power the engine will be doing the maximum work allowed by the adjusting screw; consequently this control can be applied only to engines that have a longer-range cut-off than is required for the greatest loads they have

to carry. After the above conditions are studied, it will be evident that to prevent the engine's being stalled before reaching the latest cut-off for which it was designed, we would have to dispense with the services of the adjusting screw.

6 From a study of the speed curves in Fig. 2, assuming that the height of the governor varies approximately as the speed of the engine, it will be seen that had the adjusting screw been applied and adjusted for the maximum load or minimum speed, it would be in momentary control during the second pass, and from the speed curves given in Fig. 3, it is seen that it was in action for about the same length of time during the third pass. As it would take considerable time for sufficient change in the energy of the flywheel to produce the results claimed, we believe there are other reasons for the improved conditions shown. When the adjusting screw is in control, the engine will slow down much more quickly than without it, and the engine will be stalled by a lighter load; it will also take more time to do a given amount of work.

7 As engineers prefer to have engines with some power in reserve to take care of the abnormal load, I believe they would hesitate before using any method of control that eliminates the reserve power of the motor to which it is applied.

8 From a study of the Indiana Steel Company's plant at Gary, Ind., it is evident that conservation of energy as applied to steel plants has received considerable study. The rolls are driven by motors, current being supplied by gas-engine generators.

James Tribe. In Par. 5, Mr. Caine refers to a certain engine capable of developing 25,000 h.p. while the average load does not exceed one-seventh of its maximum capacity. I do not know what engine he refers to, but a blooming mill engine of unusually large dimensions and answering somewhat to the description given, was built by the Allis-Chalmers Company and installed less than two years ago at the Carnegie Steel Company's South Sharon plant. This was a reversing engine for rolling 28 in. by 28 in. ingots on a two-high mill. The maximum power, or rather, the maximum possibility, of this engine, was likewise 25,000 h.p., which was also far in excess of its average load, but it is doubtful if there is in existence a more efficient reversing blooming-mill steam engine equipment.

2 In Par. 6, Mr. Caine asserts that in a three-high mill driven continuously in one direction, the energy stored in the flywheel would make it possible to do the same work with considerably less than one-

half the power. There should therefore be some explanation to justify the installation of so large an engine, at so recent a date, and having so large a percentage of surplus capacity. There are two reasons for this: first, because of the stalling action at the moment the rolls bite the ingot; secondly, because of the probable increase of speed as the ingot is released.

3 The reversing engine, for well-known reasons, has no flywheel, consequently the momentum of the rotating parts is comparatively nothing. Therefore the stalling action at the instant of biting the ingot, due to the tremendous impact, which is followed immediately by an abnormally high tangential resistance at the rolls' surface, creates a demand for an exceedingly powerful engine. It is just at this moment that surplus power, or reserve energy, is of the most vital necessity in order to save time and heat which would otherwise be wasted while waiting for the engine to recover itself. At this critical moment the term "horsepower" does not explain the measure of effort necessary for overcoming this resistance; for as a less powerful engine would be almost, if not quite, brought to rest, two of the power elements, namely, time and space, are for the time being practically eliminated, and the engine reduced to a simple "force" acting on the crank pin. Hence, it becomes a question of a turning moment sufficient here to overcome the resistance, and of regaining normal speed in the shortest possible time: for loss of time means not only delay (which is very serious), but loss of heat, and loss of heat means additional power necessary.

4 In the second place, the engine must be so constructed as to be capable of permitting 25 per cent increase of speed above normal with perfect safety, for the reason that at the instant the ingot leaves the rolls, the slightest delay on the part of the operator in shutting off steam, all resistance except friction having been suddenly removed, results in an increase of speed and the safe limit is quickly reached. These two extreme conditions, full steam and abnormal speed, never occur at the same instant, in actual operation, but the engine must be capable of meeting them, and therefore such an engine may be said to be capable of several times its normal capacity.

5 So far as the gripping and the releasing of the ingot are concerned the effect is the same whether a reversing or a continuously running engine is employed; for the energy of a flywheel may to some extent prevent the stalling action, just as this is accomplished by the surplus capacity of the larger engine. But flywheel energy cannot be spent without a proportionate reduction in speed, and with loss

of velocity more time must be taken to regain it than would be the case where the force of the steam is applied entirely in the mill. Part of the steam energy would be spent in restoring the wheel energy and consequently, more time would be consumed in the pass than is the case in a sufficiently powerful engine without a flywheel. This loss of time and heat partly offsets the apparent gain in economy of the smaller engine. But the more serious loss would be experienced in a three-high mill, in both time and heat, as well as the additional power required for raising and lowering the ingot to the two different levels for each succeeding pass. Considering the shortness of the passes in blooming-mill work, this delay would be a very serious loss.

6 It therefore seems to me that but little, if any, substantial advantage can be gained in heavy blooming-mill work by the three-high mill so long as it is steam-driven. It also seems to me that the only hope of any improvement in economy over present practice will be in the use of the present two-high reversing mill, but driven electrically. In such an equipment, we would have the necessary power to avoid delay on gripping the ingot, the means for instantly throwing off the power at the release of the ingot, and also the continuously running steam engine with a sufficiently heavy flywheel at the generator.

E. W. YEARSLEY.[1] The value of the flywheel as a means to obtain constant load with intermittent work is well illustrated by Mr. Caine's experiments. This arrangement has been considerably developed in conjunction with electrically-driven rolling mills. Where considerable speed variation is allowable, and there is a suitable ratio of pause to operation time, the flywheel may be applied to many drives with economy.

2 Economical considerations are at present of great importance in the steel industry. Engines used for driving rolling mills are usually excessive steam consumers. There is no doubt that their performance in this respect can be greatly improved, especially for continuously running mills. In my opinion the electric motor will be found more reliable and satisfactory for this work, and it will be desirable to confine the refinements necessary for great economy of prime movers to an electric generating station.

3 Mr. Caine's method of regulating the governor is somewhat analogous to that used for controlling the rate of application and the

[1]Electrical Engineer, Midvale Steel Company, Philadelphia, Pa.

limit of electric current to a main roll motor, in order to obtain the similar results of more uniform load, less rapid speed variation, and protection of the driver. The tests show conclusively the improvement in steam consumption and performance resulting.

4 As the paper points out, the problem is considerably complicated by variation in the number of pieces passing simultaneously, also by variation of the interval between passes and its relation to the time of the pass, and in the temperature and composition of the material. A speed variation of from 12 to 20 per cent transferring from 23 to 36 per cent of the kinetic energy of the flywheel, has been found desirable. With a given torque, time of load, and interval, this speed change fixes the weight of wheel required. Data of power performance of rolling-mill drives are rapidly accumulating. This paper is an interesting addition to such information.

The Author. Mr. Ord seems to have the impression that there was a great difference in the work done in the two examples given. As a matter of fact, the area of the piece was the same in each case, on entering the first pass, and therefore, the total work for the four passes would be as their relative weights, 2680 lb. for Case A and 2550 for Case B; B having a slight advantage in weight, and A an advantage of 5 lb. in steam pressure, so that the work was practically the same in each case.

2 The valve setting was not altered between tests. The difference in the behavior of the engine was due to the adjusting screw alone; and now, three years after these tests were made, this screw is still in service. This method of engine control does not eliminate the reserve power; it does cut it down to a point where judgment says there is still sufficient reserve to answer all requirements.

3 Mr. Tribe asks the reason for building reversing engines with such a large surplus of power. Such engines are usually driving blooming mills, where it is no uncommon practice to roll about one-half of the total number of passes, from bloom to finished product, in one stand of rolls, the remainder being taken care of from three, four or more stands of rolls, so that the blooming mill must handle these passes in very rapid succession in order to get the tonnage. The engineer handles the throttle and reverse levers, and the roller, the screw-down and the table rolls. The screw-down adjusts the distance between the rolls; consequently it fixes the amount of reduction on the bloom and the load on the engine is proportional to the reduction.

4 The screw-down has no fixed limits for each pass, therefore it will be set in a very short period, according to the judgment, or lack of judgment, of the roller. The writer has timed these operations with a stop watch and found that quite often the adjustment was made in less than two seconds; that is, the time from the end of one pass to the beginning of the next. It is quite likely that the screw-down does not get located where the operator intended; if the reduction is less, the roller will make some other passes heavier because he does not wish to add two additional passes. From the calculated results, from continuous indication cards on an engine of this type, on a single bloom one pass was noted where no reduction was made, while another pass required nearly three times the average power. From this sort of operating conditions, coupled with the desire to get an engine that will not stall under any circumstances, it becomes very evident why there is a great surplus of power. This also calls attention to one of the features in favor of the three-high mills, namely, that the roll designer can distribute the work approximately equally on every pass, with the proper data at hand.

5 The fact that the reversing engine is man-governed is brought out. This practically places the speed limit at the rate at which it would run with a wide open throttle and nothing in the mill; which would far exceed 25 per cent of the normal. Speed curve A shows that our engines run at about 16 per cent above normal, and with curve B at but 10 per cent above.

6 Mr. Yearsley suggests that the principle involved might be applied to other than mill engines. The writer can cite an instance where this was done. Our company has two-crank flywheel hydraulic pumps which are started and stopped by an accumulator. When the accumulator would drop, the governing throttle valves would open wide and the pumps would run up to the speed determined by the fly-ball governors (50 r.p.m.), and when the accumulator reached the top limit it would shut off the steam, stopping the pumps very abruptly. This continual starting and stopping caused considerable trouble in keeping up the various adjustments, and pins ran hot at times. Upon my suggestion the engineer in charge adjusted the governing throttle valves so that they could be only partially opened, and as a result the maximum speed is just a little above the average, the pumps running almost continually at about 20 r.p.m., the trouble with hot pins is no longer experienced, the rod adjustments last several times as long, and it is my belief that the water valves must give less trouble.

No. 1261

AN EXPERIENCE WITH LEAKY VERTICAL FIRE-TUBE BOILERS

By F. W. Dean, Boston, Mass.
Member of the Society

In 1905 I made a design for a large vertical fire-tube boiler, two of which were built, to be placed on a brick fire box provided with a chain grate. In accordance with a great number of precedents the water leg was short, being in fact 2 ft. deep below the underside of the crown sheet. Unusual provision was made for easy circulation by wide spaces between tubes at every 45 deg. of the circumference instead of the customary 90 deg., or as in some cases at 180 deg. The distance from the top of the grate to the underside of the crown sheet was 7 ft. The following are the general dimensions of the boilers as they now are:

Inside diameter of smallest course of shell	**120 7/8 in.**
Inside diameter of largest course of shell	**122 11/16 in.**
Inside diameter of water leg	**112 in.**
Height of water leg	**7 ft. 2 1/4 in.**
Height of brick furnace	**5 ft. 8 in.**
Distance from grate to tube plate	**12 ft. 2 3/8 in.**
Outside diameter of tubes	**2 1/2 in.**
Length of tubes	**20 ft. 0 in.**
Number of tubes	**488**
Pressure for which the boiler was designed	**165 lb.**
Kind of grate	**B. & W. chain**
Size of grate	**8 ft. 6 in. by 9 ft. 0 in.**
Grate area	**76.5 sq. ft.**
Water heating surface, say	**4900 sq. ft.**
Superheating surface	**1181 sq. ft.**
Total heating surface	**6081 sq. ft.**

2 The boilers were designed for S. D. Warren & Company and were used in their paper mill at Cumberland Mills, Me. They were built by the Portland Company of Portland, Me. Each boiler was

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rated at 500 h.p., or substantially 1 h.p. for every 10 sq. ft. of water-heating surface, and was expected to work at 1000 h.p. a good portion of the time. Artificial induced draft was used and it was possible to obtain a draft of 2½ in. of water in the smoke box.

3 At the back end of the chain grate, instead of a water back or a brick back, a vertical or slightly inclined common grate was used, against which the unconsumed coke would accumulate, and under which the ashes would pass and fall upon the ashpit floor. Difficulty was found in making the coke accumulate uniformly and the ends of this grate were frequently bare.

4 The boilers were started gently and then operated at high capacity. After about two weeks a number of the tubes began to leak at the lower ends. They were expanded, but shortly began to leak again, and this process was repeated until the tubes were so injured that they could not be further expanded. They were then removed and new ends were welded on, but after a comparatively short time they leaked again. The leaks were more on the back half of the boilers than on the front. In winter when a nearby door in the building was open and cold air blew on the vertical grate, when the ends of the grate were bare, the leakage would increase. After learning how to keep the vertical grate covered, and keeping the door closed, the general trouble continued.

5 Knowing the sensitiveness to dirt on the crown sheet of vertical fire-tube boilers, and as the design permitted access to the interior, these crown sheets were examined and found to be clean. Thinking that possibly some invisible oil had entered the boilers in some way, one of the boilers was boiled out with caustic soda, but with no effect.

6 The opinions of several boiler experts were obtained, but they differed and were unsatisfactory. One thought that the workmanship was poor, another that the design was the worst he had ever seen. Another thought that the tube plates were too limber and even recommended riveting crown bars to them to stiffen them.

7 Spring Hill coal from Nova Scotia was used at first, followed by New River coal from West Virginia. With the Spring Hill coal the lower ends of the tubes quickly became incrusted with clinker, and were finally closed by it and a little later the clinker would hang in stalactites from the tube ends. Not all of the ends would be closed, but this was the case with a large proportion of them. With New River coal there was less trouble. Spring Hill coal was satisfactory under horizontal boilers and never plastered over the tubes; in the vertical boilers, however, the incrustation was so hard

that it had to be removed with chisels. This incrustation was of course moulten earthy matter injected by the draft against the tubes and tube plate and there congealed by the comparatively low temperature of the metal. In the horizontal boilers it falls to the bottom of the setting before it arrives at the tubes. In water-tube boilers it can be seen adhering to the lower tubes.

8 The existence of this incrustation probably furnishes the explanation of the tube leakage. As a large proportion of the tubes became stopped up the others had to pass all the hot gases, the water about their ends was probably driven away and they became very much overheated, causing them to over-expand, to become upset, and at some later time, when they became cooler, to be loose in the holes.

9 As a last resort, when it seemed as if the boilers must be consigned to the scrap heap, someone suggested that to lengthen the firebox and raise the boilers by the amount of the extension might cure the trouble. One boiler was thus altered, started August 31, 1908 and run at the estimated rate of 1100 h.p. 24 hr. per day for some three months, without the slightest leakage, although the tubes were very thin from over-expansion. The other boiler was then altered, and started February 25, 1909. When the first boiler was worked at the estimated rate of 1100 h.p. it consumed 84,000 lb. of New River coal in 24 hr., burning it at the rate of 46 lb. per sq. ft. of grate per hr. Neither boiler has leaked up to the time of presenting this paper.

10 The distance from the grate to the tube plate is now 12 ft. $2\frac{3}{4}$ in. There is some incrustation, but it is light, brittle and easily crushed, and can be blown off by a rotating multiple tube-blower in the smoke box. The tubes are conveniently and quickly blown in this way every three or four hours.

11 The boiler plant at this mill consists of Babcock & Wilcox boilers, 90-in. horizontal return tubular boilers, and the two vertical boilers described. The latter were intended to reduce the space occupied, both on the floor and above. The rear drum of the chain grate is exposed and the clinker is dropped at the back end, where it is easily removed without inconvenience to the fireman. Above is room for the smoke flues and economizer, which with other types of boiler would have been placed, in this case, with difficulty.

12 After these boilers had been operated long enough to show that they were reliable and a good investment, it was decided to test one of them, with the results given in Tables 1 and 2.

13 While the evaporation is good it is not satisfactory. The function of a boiler is to absorb the heat generated in a furnace. The

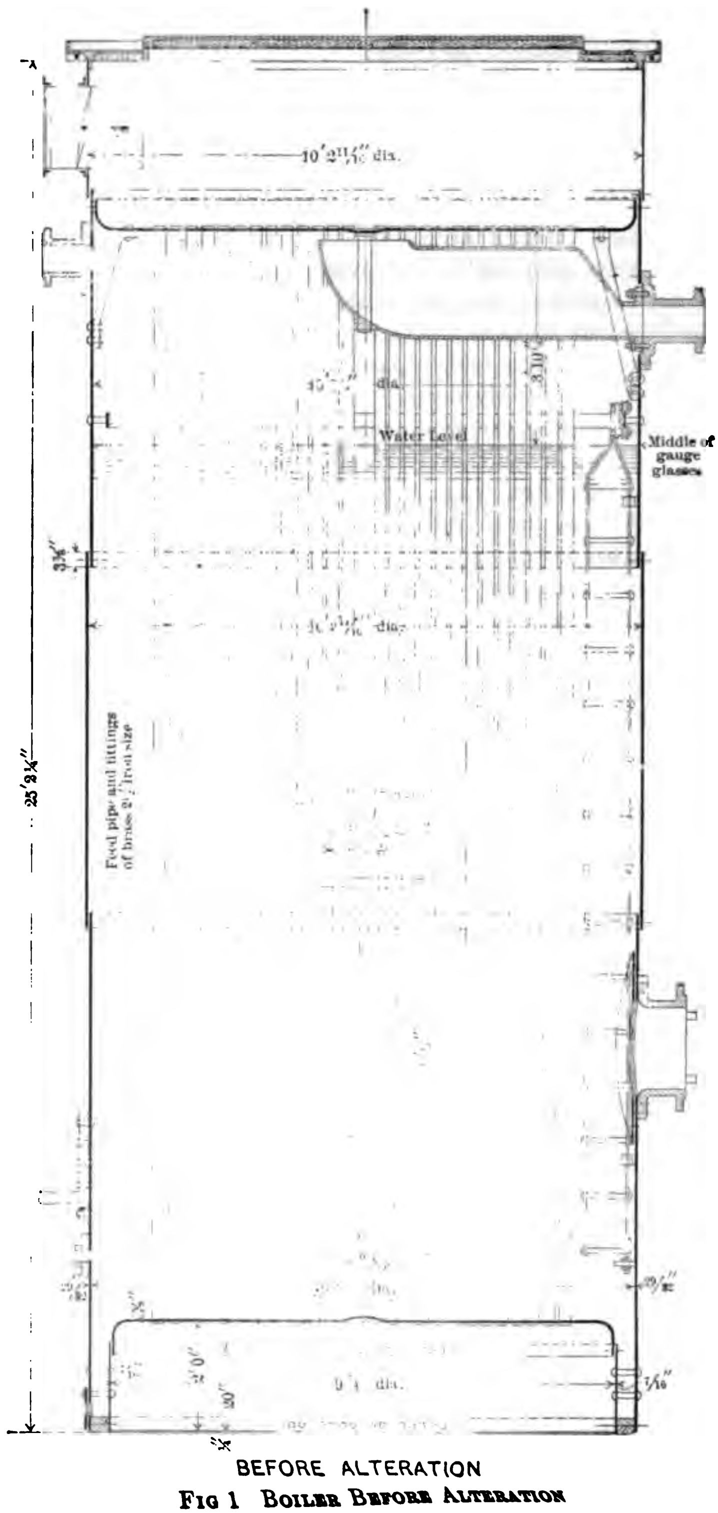

Fig 1 Boiler Before Alteration

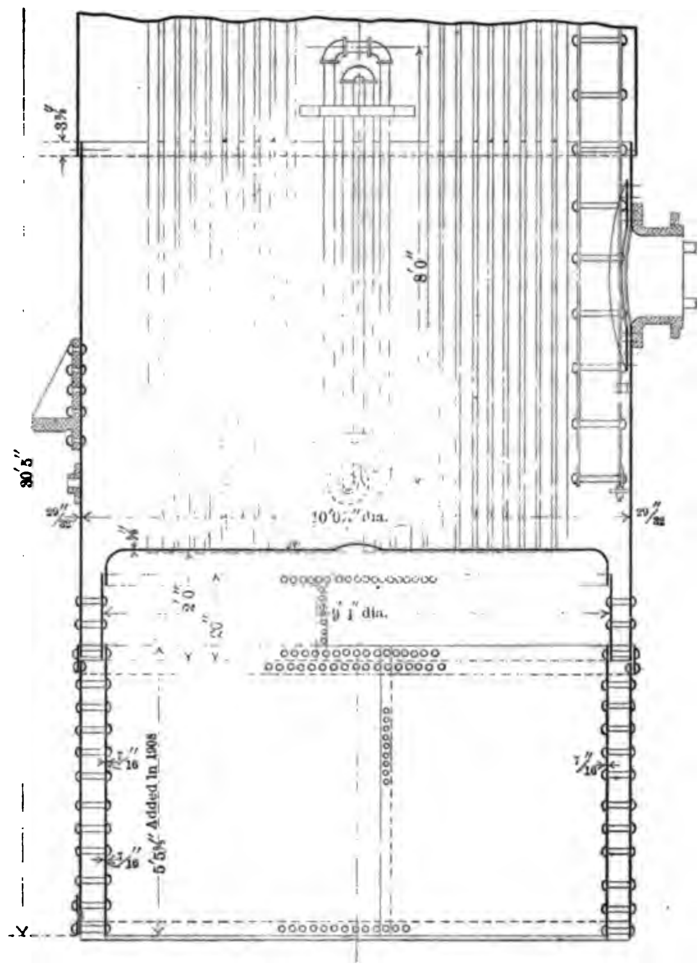

FIG 2 LOWER PART OF BOILER SHOWING COMBUSTION CHAMBER AFTER ALTERATION

furnace efficiency may be poor and the boiler efficiency good, and that was the case during these trials. The best furnace result occurs when the carbon is burned to CO_2 with as little surplus air as practicable. In these trials the CO_2 was low and some CO was nearly always found. It was impossible to get any better combustion for reasons which I do not know. Experiments will probably be made to ascertain this and overcome the trouble. It might disappear with another kind of stoker.

14 That the boiler efficiency was good is evident from the low temperature of the escaping gases, when developing over 1000 h.p., which is at the rate of less than 5 sq. ft. of water surface per horsepower. It will be noticed that the evaporation was best on combustible when the boiler was operated at double its rated horsepower. It was found impossible to keep the horsepower down to 500. This could only be done by reducing the grate area.

15 An interesting result of the tests is that the superheat was the same at all rates of power.

16 Returning again to the matter of clinker on tubes, it occurs on locomotives which burn anthracite coal, and I understand on locomotives of the Boston & Maine Railroad that burn coke. In the latter case coke-burning locomotives cannot be used on long runs, but whether a better quality of coke would be more successful I do not know. Professor Denton has informed me that clinker accumulated on the boiler tubes of a Transatlantic steamship on which he was a passenger to such an extent that men were sent into the combustion chambers to remove it. He also referred to a Manning boiler on which this trouble occurred.

DISCUSSION

Reginald P. Bolton. It appears to me that this design of boiler was an invitation to the trouble that followed, and it is only necessary to go back into the experience of other people to find out that others have suffered in the same manner. If the view of the boiler as presented in the paper is turned horizontally, and it is imagined that it is a locomotive boiler cut off short, it will be seen that there is no combustion chamber whatever in it. This boiler was to be put to a service which might call for a rate of combustion in the furnace demanding double its rated capacity output, so that the double aggravation of a very small combustion chamber and very large ate of combustion, was present.

TABLE 1 TESTS OF BOILERS

Time	Draft	Temperature	CO_2	O	CO	N
			April 29, 1909			
8.35– 9.00	0.75	575	7.6	11.2	0.6	80.6
9.05– 9.30	full	575	10.0	9.0	0.3	80.7
9.35–10.00	full	575	7.2	12.2	0.1	80.5
10.05–10.30	full	575	7.0	12.8	0.3	79.9*
10.35–11.00	full	580	7.8	11.2	0.1	80.9
11.05–11.30	full	600	7.0	12.8	0.2	80.0
11.35–12.00	full	620	7.8	11.6	0.1	80.5
12.05–12.30	full	620	8.2	10.7	0.2	80.9
12.35– 1.00	full	610	8.2	11.5	0.5	79.8
1.10– 1.30	full	585	7.7	12.3	0.1	79.9*
1.40–2.10	full	600	8.2	12.3	0.2	79.3
2.20–2.50	full	625	7.4	12.2	0.1	80.3*
3.00–3.25	full	585	7.0	13.3	0.0	79.7
3.35–4.00	full	600	4.8	14.8	0.0	80.4†
4.05–4.30	full	610	7.8	12.0	0.3	79.9
4.35–5.05	full	615	8.3	10.9	0.2	80.6
5.10–5.25	full	640	8.8	10.6	0.4	80.2
5.30–5.50	full	620	8.4			
			April 30, 1909			
8.05– 8.30	0.5	575	8.3	11.2	0.3	80.2
8.37– 9.05	0.5	540	8.3	11.1	0.1	80.5
9.15– 9.37	0.5		13.5	5.3	0.8	80.4‡
9.45–10.10	0.4	500	10.7	7.8	0.9	80.6‡
10.20–10.45	0.2	500	10.0	8.7	0.4	80.9‡
11.05–11.30	0.2	510	11.0	7.4	0.6	81.0
11.35–12.00	0.2	520	9.6	8.8	0.8	80.8
12.05–12 30	0.2	525	7.9	11.8	0.2	80.1§
12.35– 1.10	0.2	500	6.7	12.7	0.0	80.6
1.15– 1.45	0.2	525	7.9	12.2	0.2	79.7
1.50– 2.25	0.2	515	7.1	12.5	0.1	80.3
2.30– 3.00	0.2	485	7.0	11.6	0.1	81.3
3.05– 3.35	0.2	520	6.3	13.9	0.0	79.8*
3.40– 4.00	0.2	560	7.4			
			May 1, 1909			
8.00– 9.00		520	7.3	13.3	0.0	79.4
9.15–10.00	0.8	525	7.2	12.8	0.0	80.0
10.00–10.55	0.8		5.0	14.5	0.0	80.5
11.05 11.40	0.9	530	7.7	11.6	0.2	80.5
12.45– 1.30	0.7	530	9.0			
1.30– 2.30	0.7	560	8.6	10.4	0.3	80.7
2.40– 3.30	0.7	555	6.0	13.2	0.0	80.8

*Cleaned fire.
†Thin on backside.
‡Fire thick and banked.
§Fire thinner.

TABLE 2 RESULTS OF BOILER TRIALS

Running Start-and-Stop Method of Trial, Vertical Fire-Tube Boiler, New River Slack Fuel

1	Date of trial	Apr. 29	Apr. 30	May 1
2	Duration of trial, hours	7	8	8
3	Number of boilers in use	one	one	one
	DIMENSIONS AND PROPORTIONS			
4	Grate surface 8 ft. 6 in. by 9 ft. 0 in., square feet	76.5	76.5	76.5
5	Water-heating surface, square feet	4900	4900	4900
6	Superheating surface, square feet	1181	1181	1181
7	Total heating surface, square feet	6081	6081	6081
8	Ratio total heating surface to grate surface	79.5	79.5	79.5
	AVERAGE PRESSURES			
9	Steam pressure, by gage, per square inch, pounds	129.8	124.3	126.8
10	Atmospheric pressure per square inch, pounds	14.7	14.7	14.7
11	Absolute steam pressure per square inch, pounds	144.5	139.0	141.5
12	Force of draft in column of water between damper and boiler, inches	1.47	0.28	0.63
	AVERAGE TEMPERATURES			
13	External air	38	35	39
14	Feed-water before entering	50.6	63.6	58.2
15	Escaping gases after leaving	609.1	517.3	558.6
16	Steam	372	369	371
	FUEL			
17	Moist coal consumed, pounds	24,192	17,030	21,782
18	Moisture in coal, per cent	3.06	1.45	4.55
19	Dry coal consumed, pounds	23,452	16,783	20,791
20	Wood consumed, pounds	0	0	0
21	Total dry refuse, pounds	2361	1310	1650
22	Total dry refuse, per cent	10.02	7.8	7.94
23	Total combustible, pounds	21,091	15,473	19,141
24	Dry coal consumed per hour, pounds	3013	1934	2393
	QUALITY OF STEAM			
25	Moisture, per cent	0.00	0.00	0.00
26	Degrees superheated	17	17	17
	BRITISH THERMAL UNITS			
27	Number of heat units in a pound of dry coal, by oxygen calorimeter	14,759	14,759	14,759
28	Number of heat units in a pound of combustible, by oxygen calorimeter	15,760	15,760	15,760
29	Specific heat of superheated steam at constant pressure	0.61	0.61	0.61
30	Heat units absorbed per pound of steam generated	1182	1174	1167
31	Heat units imparted to boiler per pound of dry coal	9901	10,075	9893
32	Heat units imparted to boiler per pound of combustible	11,009	10,929	10,746
	EFFICIENCIES			
33	Efficiency based on dry coal, per cent	67.1	68.3	67.0
34	Efficiency based on combustible, per cent	70.0	69.3	68.2
	FACTORS OF EVAPORATION			
35	Factor of evaporation	1.224	1.215	1.209

TABLE 2 RESULTS OF BOILER TRIALS.—Continued

	Date of trial	Apr. 29	Apr. 30	May 1
	WATER			
36	Total water pumped into boiler, pounds	196,441	144,025	178,250
37	Water actually evaporated, corrected for quality of steam	198,185	145,321	179,854
38	Equivalent water from and at 212 deg. fahr., pounds	242,578	176,633	217,444
39	Equivalent water from and at 212 deg. fahr., per hour, pounds	34,654	22,079	27,193
	EVAPORATIVE PERFORMANCE			
40	Water actually evaporated per pound of dry coal, pounds	8.45	8.66	8.65
41	Equivalent per pound of dry coal from and at 212 deg. fahr., excluding economizer, pounds	10.34	10.52	10.46
42	Equivalent per pound of combustible from and at 212 deg. fahr., excluding economizer, pounds	11.50	11.42	11.36
	COMMERCIAL HORSEPOWER			
43	On basis of 34½ lb. of water from and at 212 deg. fahr., per hour, by boiler h. p.	1005	640	788
44	Heating surface of water surface per horsepower, square feet	4.87	7.66	6.22
45	Horsepower per square foot of grate surface, h.p.	13.14	8.36	10.30
46	Rated horse power	500	500	500
47	Percentage developed above rating	101	28	58
	RATE OF COMBUSTION			
48	Dry coal actually burned per square foot of grate surface per hour, pounds	39.1	25.3	31.3
49	Dry coal burned per square foot of water-heating surface, per hour, pounds	0.61	0.39	0.49
	RATE OF EVAPORATION			
50	Water evaporated per square foot of heating surface per hour from and at 212 deg. fahr., pounds	7.05	4.50	5.55

2 The design of the boiler is radically defective in two important points, namely, the tubes are entirely too long, and the combustion space was entirely too small. It is now very nearly half a century ago that the experiments of Geoffroy and Petiet demonstrated the futility of unduly lengthening fire tubes. These experiments demonstrated the rapid reduction in efficiency due to length of tubes, under various conditions of draft and rates of fuel consumption. Almost precisely the same conditions were tested as in the author's boiler, as follows:

3 A consumption of fuel exceeding 50 lb. per sq. ft. of grate, under an air pressure of 2.36 in. with the following results:

	Evaporation per Sq. Ft. Lb.
Fire-box plate	23.5
First three feet of tubes	5.4
Second " " " "	2.5
Third " " " "	1.33
Fourth " " " "	0.83
Fifth, three feet evaporated only	0.48
Sixth " " " "	0.3

The last two were found by extending the curve.

4 An examination of these results might have dissuaded the author from the mistake of designing the boiler with such a length of tube, involving not only inefficiency, but the evident concomitant of leakage as a result of expansion and contraction. Apart from the other defective feature, the boiler could have been shortened so as to reduce the tubes at least five feet in length, and would no doubt have given better efficiency as a result.

5 The general type of the boiler possesses nothing new or original unless we may so regard the restricted combustion chamber, by which the tube plate was brought within seven feet of the grate, allowing a total capacity of only 535 cu. ft. for the fire and for the gases of combustion.

6 A very simple computation of the results of the combustion of 40 lb. of coal per square foot of grate area, will show that the volume of products of combustion would be so great, that only an excessively heavy draft could force them through the combustion chamber and tubes, and that incomplete combustion was bound to result.

7 The addition of 5½ ft. to the height of the chamber, which was arrived at only after three years' experience with this boiler, nearly doubled the effective space for combustion, and also removed the ends of the tubes from the direct action of the blast. It may be

observed that a Dutch oven would have afforded equal results, at perhaps less expense.

8 The reason for the adhesion of molten clinker to the ends of the tubes, need have presented little difficulty, in the light of past experience, since the ends of the tubes were placed so close to the fire. This result developed in the fire-tube boilers of H. M. S. Polyphemus nearly thirty years ago, and when found in the boilers of locomotives is due to precisely the same cause.

9 It will be noticed that the best of the tests which were made after the change of combustion chamber was effected, is that in which the rate of fuel consumption is least.

10 I agree with the quoted conclusion of the second boiler expert, referred to in Par. 6, and am at a loss to understand why such an opinion, thus expressed, was regarded as unsatisfactory. It may be hoped that the paper may stand as a warning signal to other designers. It requires a great deal of courage to present a paper of this kind, and the author should be thanked for bringing forward a record of a failure so that we may profit by the facts.

William Kent. I join with Mr. Bolton in praising Mr. Dean's courage in bringing forward a report of his failure, and I regret that some eight or ten years ago I did not bring forward a record of another similar failure, not my own, but that of some other man, which might have prevented Mr. Dean's. The New York Steam Company bought a boiler for their Greenwich Street Station to go in a very small ground space. It was a very large plain vertical cylinder boiler, eight or ten feet in diameter, full of tubes about 20 ft. long, and was rated at 1000 h.p. It had not been in use more than a week or two when it began to leak. There was no way to clean the flat tube sheet or to clean the tubes of scale, and the boiler was condemned and taken out.

J. C. Parker. The reason that the tubes leaked was that when the boiler was set close to the grate the tube ends were subjected to wide fluctuations in temperature. The flow of air through a chain grate increases toward the rear end, and where the boiler was set higher there was more mixing of the hot and cold currents and, consequently, less fluctuation in temperature.

2 The clinkering of the tubes would naturally increase the trouble because of the concentration and increased friction of the gases in the tubes that remained clear.

Orosco C. Woolson. This discussion has brought out the important fact that perfect combustion should take place before the gases reach the tubes or shell of the boiler.

2 I have been somewhat surprised in my travels among the cotton and woolen mills of the eastern states where the management have large experience in cotton spinning but are limited in personal experience regarding what constitutes the production of the highest calorific value of a pound of bituminous coal. One man of large experience in mill work wanted his furnace fire directly under the tubes of his vertical boilers, and gave me his reasons. I told him that I would guarantee him better results if he would discard the idea that the area immediately under and against the tube sheet should act as a combustion chamber. Let combustion take place entirely before it reaches the tube sheet and the results will be much more satisfactory.

3 Secondly, as to the tubes filling with vitrified slag or any other residuum of combustion, I would suggest that such deposit should be made to take place under a fire arch, where it will adhere to the crown and serve a useful purpose by forming a refractory coating. This practice is becoming popular, and more so today than ever before. It is my opinion that by completing combustion under a properly constructed arch within a properly constructed combustion chamber, the products of this combustion will be sent to the boiler in the form of what we will term "caloric ether" and not a mixture of its original constituents which play no useful part, under the circumstances, in producing or maintaining heat, but rather are subject to ready condensation.

A. A. Cary. In my experience with vertical fire-tube boilers I once found a boiler containing shorter tubes and of a greater diameter than ordinarily found in the Manning type. The fuel used was a moist anthracite coal, and there was a natural draft of more than one inch of water in the smoke box over the boiler. The draft could not be regulated, due to the previous burning out of the steel plate butterfly damper. The partially burned furnace gases passed rapidly through the vertical tubes and ignited above the top tube sheet, thus causing the destruction of dampers and the steel breeching, to say nothing of the reduced evaporation in the boiler due to this waste of heat.

2 The trouble was remedied by placing the grates at a greater distance from the lower tube sheet and arranging baffles in the

combustion chamber so as to insure the more complete combustion of the gases before they entered the tubes. A cast-iron plate damper replaced the former one of steel plate, and no further trouble has since been experienced.

3 In another case, the question came up as to the advisability of applying a special automatic furnace, using bituminous coal and producing very high temperatures, under boilers of the Manning type. An arrangement which has been used in New York City for a number of years was suggested and successfully applied.

4 Fire-bricks, piled on edge with open spaces between the bricks, were arranged a short distance beneath the lower tube sheet. This checker work of bricks filled the entire space beneath the boiler, the openings between the bricks at the center being very much reduced, so as to cause a decreased flow of gases directly under the center of the overhead tube sheet. By this means, a very even distribution of temperature was secured over the entire area of the lower tube sheet with a slight reduction of heat delivery at its center, the most sensitive portion of the whole tube area.

5 The author mentions inefficient combustion, which is indicated by the comparatively low percentage of CO_2 and high percentage of O, shown in Table 1. As the higher temperatures are secured by the most complete combustion with the least excess of air, the question arises, why should such destructive results follow such inefficient furnace conditions?

6 Pyrometric testing with gas analyses have taught me that when a furnace is being operated inefficiently, very high temperature may be found in one part of the furnace while at the same time a comparatively low temperature may exist in another part. This may lead to the simultaneous impingement of gases of very different temperatures upon various parts of the lower tube sheet, setting up destructive strains and contributing to such troubles as have been described by Mr. Dean.

7 The lower tube sheets of boilers of the Manning type are very sensitive, especially towards the center of the sheet where the water seems to penetrate with great difficulty, thereby failing to keep this portion of the heating surface constantly wet.

8 Concentration of heat due to concentration of combustion and lack of space for this small volume of high-temperature gas to diffuse itself throughout the entire mass of furnace gases before they reach the tube sheet, is bound to cause trouble, especially when this highest temperature is concentrated against the center of the tube

sheet on the inner surface of which there is apt to be little or no water. After the center of this sheet loses the supporting effect of the center tubes, acting as stays, the surrounding tubes are very apt to follow.

9 Concerning the low efficiency of the furnace referred to in Par. 13, there should be no trouble in remedying this fault. A properly conducted furnace test (apart from the boiler) with pyrometers, gas-analyzing apparatus, etc., will show just where the trouble exists and will point out the needed changes as well as the limitations under which this type of stoker can be worked with the different grades of fuel used.

PROF. L. P. BRECKENRIDGE. One of the speakers said that the highest temperature in a boiler furnace is directly over the fire. This is not always so. We have measured the temperature twenty feet from the fire and found it higher. It depends on the volatile content of the fuel and whether the flame has been supplied with a sufficient amount of air early in the process of combustion. It is this that determines whether the high temperature point is ten feet or twenty feet away. Many times in our experiments in the St. Louis boiler trials we have seen that every time the furnace door was opened the temperature at the rear end of the combustion chamber went up, because when more air was admitted the combustion was better and the temperature increased.

2 For experiments concerning the transmission of heat through a boiler tube, it occurs to me that Mr. Dean has designed one of the most satisfactory laboratory boilers I have seen. There has been much discussion of late on the heat transferred through a boiler tube, as influenced by the velocity of the gases passing through the tube. This boiler with its large number of tubes would be just the type with which to make a test on this point. I wish Mr. Dean would burn a large amount of coal per square foot of grate in this boiler furnace, using, first, all the tubes, and secondly, only one-half the tubes. If the same amount of coal was burned in each case the velocity of the gases through the tubes would be twice as great in the second case, and it would be interesting to know the relative amounts of heat transferred.

3 I hope that some time we may take up the question of the burning of fuel, making a distinction between the economical performance of the boiler and of the furnace. We have reached a time when we can intelligently discuss these questions separately. Anthracite coal, on account of its high fixed-carbon content, is burned mostly on

the grate itself. When burning semi-bituminous coal, with 18 to 20 per cent volatile content, a large combustion chamber is required, and as the volatile content increases the size of the combustion chamber must be increased. When burning bituminous coal, with 40 per cent volatile content and 20 per cent ash, the fuel actually burned on the grate is small. The grate supports the fuel and some coal is burned there, but it is in the combustion chamber that we burn fully one-half the combustible part of our fuel. It is evident that more attention must be given the proportions of our combustion chambers when burning high-volatile coals, and especially at high rates of combustion.

Prof. A. M. Greene, Jr. In London Engineering for October 22 and November 5, 1909, appeared an article by Professor Dalby, in which he summarized a number of articles referring to heat transference through plates. I would commend the article to the attention of all the members of the Society interested in this matter.

2 In London Engineering for February 1909, Professor Nicholson described experiments showing clearly that only a small part of the possible heat transmission through plates is utilized. I mention this to call the attention of the members to the fact that some data are available on this subject. In this article are given the formulæ for heat transmission which may be compared with the results of German Experiments recently completed at Dresden (Zeit. des Verein Deutscher Ing., October 23, 1909).

William Kent. In another issue of London Engineering, a correspondent showed that the idea of high speed of the gases being favorable to combustion was negatived by the Lancashire boiler, in which the flues are very large and the speed of the gases low, yet the economy is as high as in any other boiler.

Reginald P. Bolton. It is mainly a question of the difference in temperature between the inside and outside of the heating surfaces. The lower the temperature of the feed water, and the higher the temperature of the fire, the greater will be the efficiency of the boiler.

E. D. Meier. I find myself in substantial agreement on some points with all the gentlemen who have spoken. I want to say for Mr. Dean, that he is correct in his conclusion that the precipitation which occurs at the bottom of the tubes has a great deal of influence on the overheating of the tube sheet. The other causes which were mentioned are also true, but there is no doubt an accumulation of carbon there. I do not know whether Mr. Dean preserved any of the precipitate or stalactites, but I believe a large part of it was unconsumed carbon, which will remain at a high temperature for some time.

2 I am reminded of an experience which I had with water-tube boilers at the Chicago World's Fair. I think there were ten different makes of water-tube boilers, most of them sub-horizontal, but some of the vertical-tube type and some of the bent-tube type. We were burning crude oil, and all the boilers suffered from the same causes,—every one lost tubes by burning out. Some were careful enough to shut down a boiler as soon as they noticed the blisters on the tubes.

3 The boilers which I had at Chicago were afterward placed in the midwinter fair at San Francisco, and were fired with California crude oil for seven months without a tube being lost. These boilers were afterwards sold with the condition that if the customer found any tube damaged it would be replaced, but not one was found to be burned. That bears on the subject mentioned by Mr. Dean. The trouble we found was this: The oil is supposed to be atomized in the burners, but this is not always the case. Little slugs of oil would fly up and adhere to the tube, and would spread and slowly carbonize. They would not burn, because no air could get to them. One little spot, a half inch in diameter, would become red hot in spite of all the circulation of water, and would ultimately burn out and make a blister.

4 When the boilers were installed in California, the oil burners were placed lower and were directed downward so that the jet would strike the bottom of the combustion chamber at a distance of six feet from the front, hence there was no chance of oil striking the tubes. Perfect combustion was obtained, and on one occasion one of the boilers was forced so hard that a picture was taken of the inside of the furnace by its own heat. I have that photograph still, to show what can be done. One can see a perfectly white heat and not a single blister on the tube. In Mr. Dean's case carbon was deposited and became incandescent, and gave an intense local heat on some point, which accounts for the failure of the tubes at such point.

5 In regard to the combustion chamber, I agree with Professor Breckenridge. I have always been a believer in a large combustion chamber, and one of my early experiences in that direction was when in charge of a plant having two horizontal tubular boilers, using

Illinois coal. At that time everybody in the Mississippi Valley believed in river practice. The boilers, engines and dimensions of pipes, etc., were according to river practice. The boilers were set with the grate twelve inches from the bottom of the she. Ill raised them to thirty inches, and I was told I would not get any heat, but I got better results, and the boilers lasted longer. The increase in the distance from the fire to the shell was a great advantage, and, of course, incidentally I increased the efficiency of the boiler.

DAVID MOFFAT MYERS. In my paper on Tan Bark as a Boiler Fuel, results of an efficiency test are given in Table 4 in which the temperature inside the furnace was 1100, the temperature in the combustion chamber, under the boiler, was 1475, the flue temperature was 493, and the thermal efficiency was 71.1 per cent.

2 These figures prove that under conditions of good efficiency it is quite possible to have a higher temperature at some distance from the fuel than close to it. The combustion of the gases is simply retarded to a later point of their travel. This might be caused by the combination of a high velocity of draft with a moderate air supply, so that the oxygen does not come into sufficiently intimate contact with the fuel gases in the primary combustion chamber, that is, in the furnace proper. In the case quoted, the CO_2 ran almost uniformly at about 12 per cent, the O between 6 and 7 per cent, with no determinable CO.

A. BEMENT. In the boiler which Mr. Dean describes, I like the scheme of having the rear end of the chain grate exposed so that it is accessible. The capacities obtained with these boilers are very large; the strength of draft, however, is somewhat too much for an ordinary chain-grate fire. It is my experience that chain grates are not proportioned so that it is possible to carry the requisite thickness of fire for a draft such as existed in this case. I think this will account for the low percentage of CO_2 in the combustible gases, and in this is found the reason why the efficiency was not higher.

2 I would attribute the leaking of the tube ends in the head over the fire to another cause than that given. Considerable experience in similar cases leads me to believe that the trouble is caused by excessive heating on the delicate tube ends in the flue sheet. There are two thicknesses of metal to be penetrated before the heat reaches the water; also the opportunity for water to enter among the tubes and to flow freely over the heated parts is rather restricted. When

the ordinary return tubular boiler is set with a fire under the shell, a large portion of the heat flows through the shell, with the result that the temperature of the gases is much reduced, so that by the time they impinge upon the tube sheet, their temperature is low enough so that no damage results.

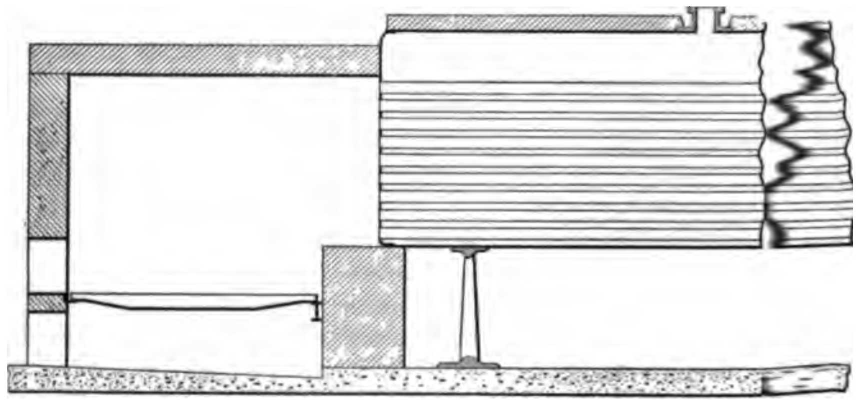

FIG. 1 SETTING OF A FIRE-TUBE BOILER IN WHICH THE TUBES LEAK:

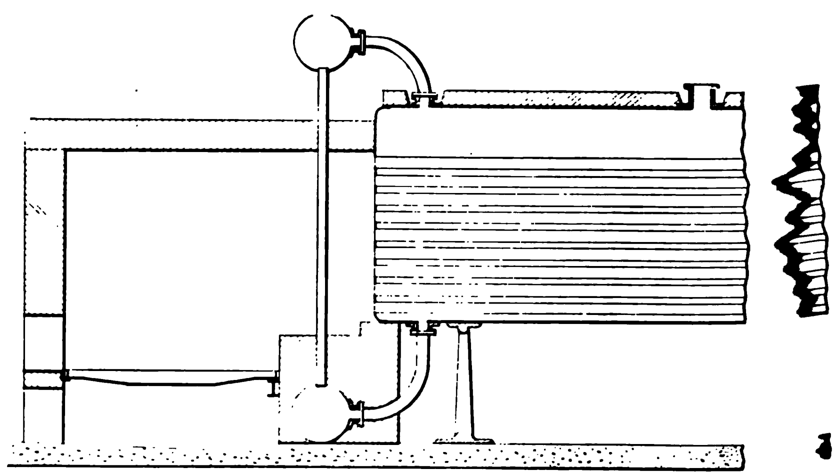

FIG. 2 SHOWING WATER LEG TO LOWER TEMPERATURE OF GASES IMPINGING ON TUBE SHEET

3 A case of trouble of this kind is illustrated by Fig. 1 and Fig. 2, the first showing a return tubular boiler set against an enclosed fire-brick furnace, in which the gases first impinged upon the tube sheet, passing through the tubes to the other end of the boiler, thence fin-

ing exit by way of a chimney attached thereto. When these boilers were put at work immediate and very serious trouble resulted with the tube ends; they leaked very badly, the bead getting out of shape and springing away from the sheet. By means of a little door in the side of the furnace one could see the water squirting from every tube, and running away from the setting on the floor in a large-sized stream.

4 A remedy was effected in this case, as shown by Fig. 2, by mounting above and below the furnace a drum which extended crosswise of the setting, and connected by vertical 4-in. boiler tubes as indicated; each of these drums being in communication with the boiler, allowed circulation of water and steam. With this scheme the gases first pass between these vertical tubes, which are set closely together, with the result that there is a considerable reduction in the temperature of the gases before they came in contact with the end of the boiler tubes.

5 Another case of this character was remedied by carefully cleaning off the end of the boiler and coating it with an asbestos cement, which was rounded over and into the boiler tube openings in such a way that the flue sheet was entirely protected. This covering lasted about three months, after which it was necessary to renew it. As it was a house-heating boiler, two renewals a season served until the boiler plant was dismantled. The cure of the trouble with the boiler having the extended water leg, as shown in Fig. 2, is due in my opinion to the added heat-absorbing surface in the deeper leg, as it operates to abstract a much larger quantity of heat from the gases before they came in contact with the tube ends, than did the boiler before alteration.

THE AUTHOR. Replying to Mr. Bolton's remarks, I have heard of the experiments which he quotes in regard to the rate of evaporation of different portions of the length of a tube, but I am not at all impressed with them as a guide. It is well known that the first surface that receives heat gives the greatest rate of evaporation and leaves less for the remaining surface to do. Attention to this to the extent apparently advocated by Mr. Bolton would lead to an absurd result, for one might go on indefinitely shortening tubes. It should be remembered that only 16 feet of the 20-ft. length of tubes are in contact with water, the remainder being for superheating.

2 Apparently Mr. Bolton believes that it is known how long tubes should be. I do not think that this is known, for the reason that a boiler must undergo a wide range of duty; a short tube would do

for light work and a long one would be necessary for heavy work. Many vertical boilers with 2½-in. tubes 20 ft. long have been used successfully for years and they are still being built. Mr. Bolton would evidently prohibit increasing the size of a boiler by increasing the length of tubes, and would recognize only an increase in diameter as a means of increasing size. To my mind this is illogical and not consistent with the teaching of successful practice.

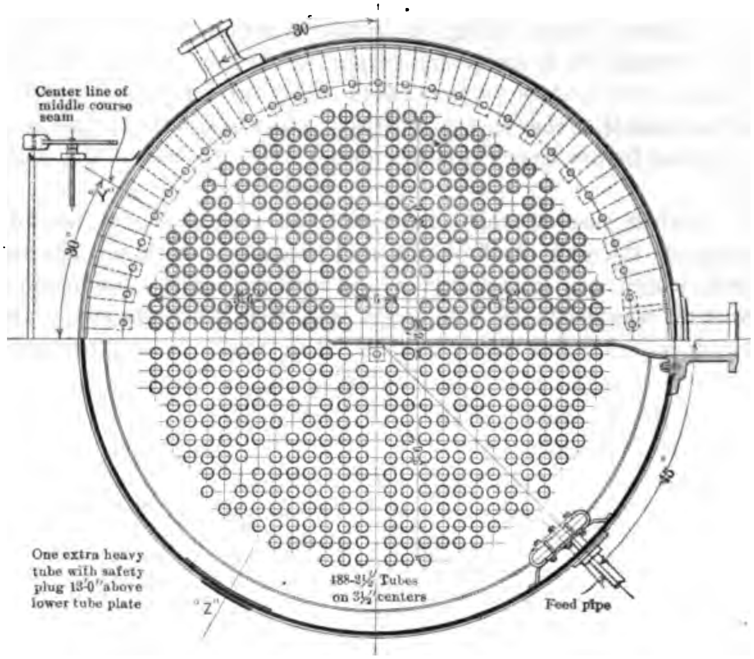

Fig. 1 Cross Section of Vertical Fire-Tube Boiler

3 Mr. Bolton speaks of the small combustion chamber as the boiler was first installed, but he ignores the hundreds, if not thousands, of vertical boilers with less combustion chamber space. I believe that I am the only person who designs vertical boilers with the crown sheet as much as 8 ft. above the grate, and this I have been doing for many years. In regard to the Dutch oven in front of these boilers, it would have wholly defeated the object of using vertical boilers. Besides it would have added undesirable brick work.

4 Mr. Bolton easily accounts for the lack of economy of the boiler, but ignores the perfection with which it absorbs heat. I believe the

lack of economy to be wholly due to want of air, and when this is supplied and properly distributed the economy will be satisfactory. This would be equally true if the combustion chamber were much longer. The locomotive boilers tested at the St. Louis Exposition by the Pennsylvania Railroad have very little combustion chamber

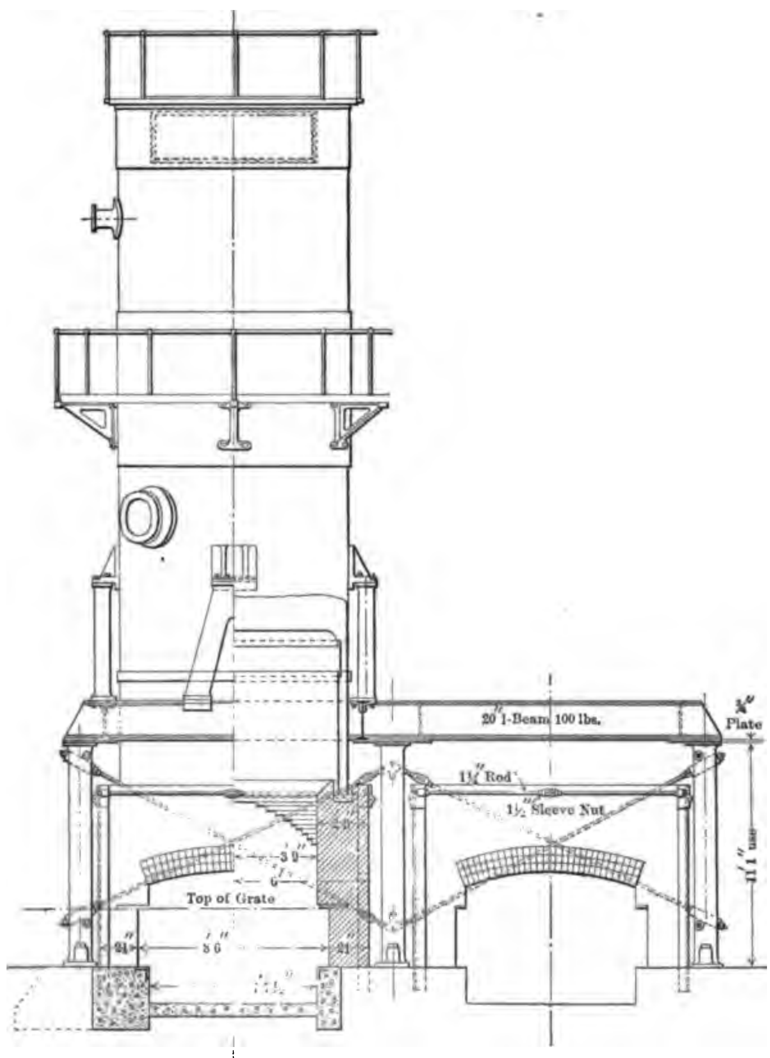

FIG. 2 SECTIONAL ELEVATION OF FURNACE OF THE AUTHOR'S FIRE-TUBE BOILER

space, and the excellent economy is due to the proper admission and distribution of air. In regard to the economy of the boilers under discussion, it should be remembered that it was good, only not as good as is sometimes the case.

5 Mr. Parker states that the tubes leaked for the reason that they were set close to the grate and were therefore subjected to wide ranges of temperature. This is true if we consider the closing of many of the tubes by clinker and the consequent overheating of those that were not closed.

6 I agree with Mr. Bement that some other kinds of stoker would probably not have precipitated the clinker on the tube ends, and this I stated in the paper.

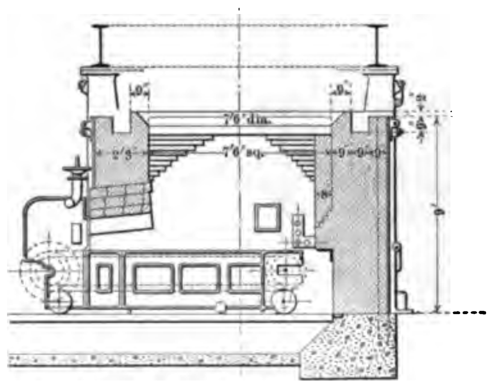

Fig. 3 Section of Furnace of the Boiler Shown in Fig. 2

7 Concerning the ability of the water to enter among the tubes, there are many large vertical boilers, some nearly as large as the one described, that have far less space for the passage of water among the tubes, and no trouble results. I know of some that have only one wide space across the crown sheet, while mine have four wide spaces entirely across, or eight reaching to the center.

8 I observe that Mr. Bement considers that the cause of the cessation of the leakage of the tubes of my boiler was the added surface of the water leg. I cannot feel that this is so. It is inconceivable to me that the heat near the center of the furnace should be sensibly reduced thereby. Moreover the absence of the clinker after the change seems to me ample cause of the improvement, for, as I have stated in the paper, a large proportion of the tubes were stopped up, and those that were in service must have been overheated. I

think that if the boilers had been raised without adding to the water leg the trouble would have ceased.

9 Whatever the cause of the leakage may have been, I find on January 17, 1910, the date of writing, that the tubes are not leaking; nor have those of one boiler leaked since August 31, 1908, nor those of the other since February 25, 1909, each boiler having been worked constantly to about 1000 boiler horsepower.

No. 1262

THE BEST FORM OF LONGITUDINAL JOINT FOR BOILERS

By F. W. Dean, Boston, Mass.
Member of the Society

It has been generally accepted in this country for a number of years, that the best form of butted longitudinal riveted joint for boilers is that in which the inside strap is wider than the outside, and which has one or more rows of rivets passing through the shell and the inside strap beyond each edge of the outside strap. The pitch of the first row of outer rivets is double that of the rows that pass through both straps, and if there are other outer rows they may or may not have a still greater pitch.

2 In England, where until comparatively recently boiler construction has been superior to ours, this form of joint appears to receive no recognition. It was first devised, as far as I know, by Dr. E. D. Leavitt, Past-President of the Society, and Edward Kendall, both of Cambridge, Mass., and was first used by Mr. Leavitt in some locomotive type boilers designed by him for the Calumet & Hecla Mining Company. I have a blueprint of this boiler dated 1879. It is, of course, hazardous to state that this joint was never used before and it is quite possible that it was used in England, and discarded and forgotten as poor construction, as I believe it is. It was first used on an American locomotive by the Baldwin Locomotive Works in a consolidation locomotive built by them for the Calumet & Hecla Mining Company, the drawing of this joint having been made by me when I was in Mr. Leavitt's employ.

3 While every boilermaker has for years been familiar with butt joints, this form made slow progress towards adoption in this country. One form of joint used to avoid the butt joint and get something as good, was a lap joint with an inside strap bent at the edge of the lap and riveted on each side of it. This was used on locomotives exclu-

Presented at the Annual Meeting, New York, (December 1909), of The American Society of Mechanical Engineers.

sively, and was of little or no value as it was simply a somewhat elastic bent tie connecting the two parts of the shell plate. Finally, and fortunately, this joint gave way to the butt joint first described.

4 I believe there has been no case of an explosion of a butt-joint boiler; at least one due to rupture of the joint. Recently, however, a boiler at Woonsocket, R. I., narrowly escaped explosion, a longitudinal rupture of the plate on one side of the joint, and within its limits, being discovered while the boiler was subjected to steam pressure. The steam pressure was rapidly reduced and no explosion occurred. An account of this is given in Power, January 26, 1909, and the joint itself is in possession of the boiler-inspection department of the Massachusetts district police at the state house in Boston.

5 It has been growing upon me for some years that a one-sided boiler joint, such as that first described, is poor construction, and may sooner or later cause a crack in the plate. The Woonsocket phenomenon has tended to confirm this opinion. It is evident that unless the outside rivets fill the holes they do very little good, and when they do fill them they form an overhung connection and to some extent possess in themselves the now recognized defect of the lap joint. Moreover the extended inside plate forms a bent connection between the different rivets at different distances from the center line of the joint.

6 In many cases designers have placed the outside rivets at a considerable distance from the edge of the outside strap and this is constantly overdone. It is obvious, on careful thought, that the outside rivets should be as near the edge of the outside strap as practicable, thereby diminishing the bent-tie effect. In order to diminish this effect still further, and also to render the overhung rivets more effective, the inside strap should be thicker than usual, and this feature can hardly be overdone. The inside strap should be at least as thick as the shell plate, and great care should be taken to have the holes match and the rivets fill the holes.

7 When a joint of this kind is tested to destruction in a testing machine, it will be found to fail somewhat in detail, the inside strap bending slightly and the outside rivets being the last to rupture after yielding a little. In a boiler the joint would be weaker than a flat specimen on account of the bent-tie feature. This could be prevented if it were practicable to calk the inside strap, as it would thereby be compelled to maintain the circular form. The theoretical efficiency of this joint is greater than of any other kind, but in practice I believe the efficiency is not realized and the defects that I have described render the joint, in my opinion, undesirable.

8 In order to avoid the defects of the one-sided butt joint, I have adopted and intend to use hereafter, a joint with both straps of the same width, as illustrated in Fig. 1. This has the merit of having all rivets in double shear and the strains all taken care of in the best manner. The efficiency of this joint can hardly be above 84 or 85 per cent while that of the one-sided joint can be theoretically 91 or 92 per cent; but the certainty that the efficiency of the former is realized in practice is ample compensation for the use of slightly thicker plates. The pitch of the outer rows of rivets is rather great, compelling the use of a thick outside strap in order to stand calking and remain steam-tight. I use an equally thick inside strap in order to diminish the bent-tie effect. This effect is small, however, as the

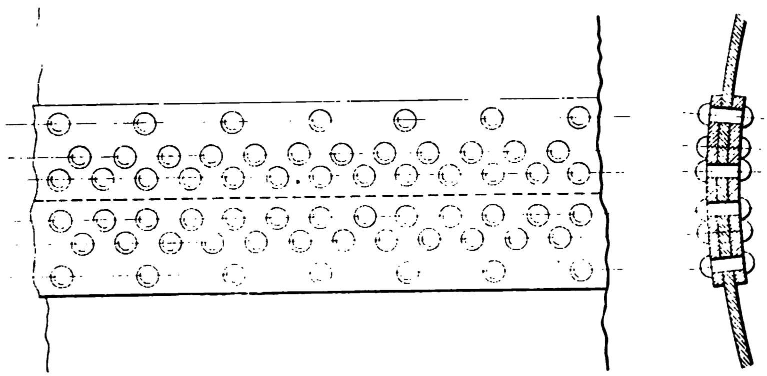

FIG. 1 RECOMMENDED FORM OF LONGITUDINAL JOINT

rivets are all near the center of the joint. It can be eliminated by calking the inside strap, which is practicable with this joint, and is done in the best marine practice. This assumes that the calking is effective and will remain so.

9 While this subject is under consideration, it is well to call attention to the perfection with which the longitudinal joints of boiler-plate cylinders can be welded, a fact which has been demonstrated for many years with corrugated furnaces and more recently with soda digesters. While the joints of corrugated furnaces are in compression those of digesters are in tension, and their proved safety should be sufficient to overcome any timidity concerning the perfection and safety of welded joints. Circumferential joints are not so easily welded as longitudinal, and it is of course of little importance in boilers that they should be welded.

DISCUSSION

Reginald P. Bolton. The form of longitudinal joint for boilers which Mr. Dean has described as the best is as old as the time of Brunel, and was tested by him in 1838, and again by Longridge in 1857. It is a double-welt triple-riveted joint, omitting alternate rivets in the outer strip, and it has the defect of undue distance for calking between the outer rivets. It is not so good a joint as it would be when the triple riveting is continued, instead of omitting the alternate outer rivet. The other form of joint to which Mr. Dean refers, in which the inside welt was wider than the outside welt, has stood the test of many years usage, and I do not know of any case of failure.

2 In discussing the longitudinal joint, we should not lose sight of the fact that the weak parts of every longitudinal joint are the ends, where the two shell plates unite and the circular seams meet the longitudinal joint. It is there that weakness develops in all joint construction. In explosion cases on which I have been engaged, I have found that trouble has developed at those points, and have noted that ruptures commenced there. Therefore, in dealing with the design of longitudinal joints, the essential feature seems to me to be its character where it meets the circumferential seam.

E. D. Meier. I think that the value of this joint depends largely on the diameter of the boiler that one has in mind. In a Scotch marine boiler, from 12 to 15 ft. in diameter, the joint would be an excellent one, especially with the scalloped edges mentioned by Mr. Dean[1]. That is a very troublesome thing to do, but in addition to the advantage of the scalloped edge which Mr. Dean cited, there is the further one, that it modifies the tendency, common to such joints, to buckle at the point where the sheets come together. The butt joint is stiffer there than any other part of the shell and with a change in the pressure and temperature the buckling ultimately tends to impair the joint.

2 With a small boiler, 36 in., 42 in., or 48 in. in diameter, the joint is too large a proportion of the total circumference, and this action would become worse. That buckling action is distributed by making the butt plates as thin as possible, and making the inside one longer than the outside one.

3 The welded joint will be an ideal one when we can be sure of

[1] This was referred to by the author in presenting his paper.

a weld that will give 95 per cent efficiency. The difficulty will be to test it. We do know, however, that when we rivet a joint and do it honestly, we have something that can be relied on. Much will depend on how the material is chosen and how the work of laying up and riveting is done. The joint should be made by carefully bending the butt straps at a red heat to the true curve, and rolling the plate itself true to template. This will make as perfect a joint as possible. For a large diameter of boiler, I think the joint advocated by Mr. Dean, especially if the edges are scalloped, is an excellent one, but for smaller diameters I prefer the old joint.

4 Two other points must be considered: first, how the calking is done, as in many sheets the initial fracture is caused by bad calking; second, what sort of metal was used, for unless the chemical analysis of the plates as to minimum of injurious metalloids is firmly insisted on, trouble is sure to follow even in the best proportioned joints.

Prof. A. M. Greene, Jr. Mr. Dean is probably aware that in the 1893 report of the Chief of the Bureau of Steam Engineering of the Navy, it is shown that the boilers intended for the New York, the Columbia and the Minneapolis, were all designed on the same plan as that which Mr. Dean recommends. The illustration in the paper is almost exactly similar to those in the report. These boilers were all installed and have given entire satisfaction.

2 Locomotive engineers, however, are using the unequal length butt strap quite extensively. I know of locomotives in which two rows of rivets were placed outside of the outer butt strap, and I do not know of any failure of such joints. If it is a case of getting increased efficiency, and still having the outer butt strap arranged for a calking distance, I do not see why we should depart from the method of unequal straps to use the equal strap arrangement which cannot give such high efficiencies.

William A. Jones. I wish to point out the tension which exists in the outer row of rivets and its effect on the drum shell. This should have an important part in determining whether the form of joint which Mr. Dean recommends is really better than if the outer butt strap were cut back one row of rivets on each side, so that the rivets at their calking edges would be close together.

2 We probably all agree that rivets are more reliable in shear than they are in tension; that the more closely and firmly the edge of the outer butt strap is held down, the less calking will be required and

the less possibility there will be of injuring the shell plates by calking the butt strap in the shop, and the more remote will be the probability of subsequent leaks, prompting inexpert men to calk them again later.

3 If we assume that the inner rivets are about 3 in. apart, then the outer rivets shown in the joint which Mr. Dean recommends will be about 6 in. apart, and each rivet will be holding an area of butt strap of from 15 to 20 sq. in., which, at 200-lb. pressure, will require from 3000 to 4000-lb. tension per rivet. In addition, each of these rivets will be required to hold the calking for an edge about 6 in. long, and the calking will have an advantage over the rivet of about 2 to 1, due to the leverage which it has because the rivets are back from the edge. It does not require much thought to see that these rivets would be better able to do this work if they were twice as close together.

4 The joint which Mr. Dean has shown has five rivets in double shear on each side, in a length equal to the pitch of the outer rivets, so that ten times the area of one rivet is the total area in shear in this length. If, on the other hand, the outer butt strap were cut back so that the rivets at its edge would be close together and the outer rivets were in single shear, then the total area in shear would be only one-tenth less, and the proportion of the circular tension transmitted by the rivets in single shear could not be more than 11 per cent of the total in this case.

5 I understand that it is in an effort to improve the action of this 11 per cent of the force involved that this wide outer butt strap is recommended, and that where four rows of rivets are used instead of six, this proportion may rise to 20 per cent. In any case, the slight bending in the shell plate is less, I believe, than the bending tendency which the tension would produce in the rivets, due to pressure on the wide outer butt strap.

6 Let us consider the forces acting upon a rectangular area of plate in a drum shell under pressure. The circular tensions acting tangentially at the edges of this area are equal in intensity, but act at an angle to each other, so that each has a component normal to the chord of the area considered. These normal components exactly balance the pressure acting on that chord. When the area considered embraces a half-circle, the normal components become equal to the circular tension.

7 In the case of the outer butt strap, if all the circular tensions of the drum could be transmitted to the outer butt strap by rivets at its extreme edge, the shear of these rivets alone would hold the outer butt

strap to the drum, and the components of the shears normal to the chord would just balance the steam pressure on that chord, so that no tension in the rivets would be necessary, except for calking. Moving the rivets back from the edge of the butt strap makes the shear act more nearly parallel to the chord, while it does not diminish the chord, so that shear alone will no longer hold the butt strap in place, and tension must be developed in the rivets to make up the difference.

8 Transmitting part of the circular tension through the inside butt strap further increases the tension on the rivets, due to pressure, but the additional tension in this case maintains the curve in the inner butt strap by stitching it to the surface which receives the pressure, and the reaction of the tension at the inner ends of these rivets is thus provided for.

9 In the case of the outer rivets of the joint which Mr. Dean shows, reaction of this tension at the inner ends of the rivets must be absorbed by an abrupt change in direction of the circular tension at those points, tending to produce corners in the drum shell in order to satisfy the triangle of the three forces formed by the tension on the rivet, the tangential tension to the right, and the tangential tension to the left. If we assume a 42-in. drum, 200-lb. steam pressure, 6-in. pitch of outer rivets, each of which takes in tension the pressure of 20 sq. in., we have 4000-lb. tension in each rivet due to steam pressure, the inner ends of the rivets being anchored by an abrupt change in direction of about 9 deg. of 25,200-lb. circular tension.

10 Evidently, this abrupt change of direction of the total circular tension may readily distress the plate more in the form of joint which Mr. Dean recommends than in the usual form of joint with the narrow outer butt strap, even though a very small part of the circular tension is transmitted through a rivet in single shear.

11 Mr. Dean's statement that he believes there has been no case of failure of butt-strap joints, would indicate that there was nothing wrong with the established form using the narrow outer butt strap. Certainly the remedy proposed seems more objectionable than a rivet in single shear.

Sherwood F. Jeter.[1] It seems that all engineers design joints with reference to their weakest point, that is, provided the joint were to be ruptured in a machine. Of all the explosions that to my knowledge have been due to ruptures, none have occurred in the theo-

[1] The Bigelow Co., New Haven, Conn.

retically weakest part of the joint. Most explosions due to rupture of the sheet have occurred near the joint and were apparently due to flexure of the metal, which had destroyed its life at the particular point of rupture.

2 I believe that there is a great need for an investigation as to what causes the rupture of the plate, and for other than machine tests of different kinds of joints. An account in Power states that there have been four ruptures of butt-strap joints of a nature similar to what was previously alluded to as a "lap cracking" of the joint. From the great number of lap joints in successful use for twenty-five years or more, it may be judged that something besides a mere lapping of the plates causes such defects.

THE AUTHOR. There is very little for me to say in closing, as my views have been fully set forth in the paper. I am interested in the history of this joint as stated by Mr. Bolton. I first knew of it in 1889; it is shown in Thomas W. Traill's book on Boilers, and a table of sizes of parts is there given.

2 Several of the speakers express doubt as to the tightness of the joint on account of the wide spacing of the outer row of rivets. There should be no doubt of this kind, for too many of them are in use. I know of one joint with $1\frac{1}{4}$-in. rivets in 1-in. straps on a pitch of $9\frac{1}{8}$ in., and another with $1\frac{1}{16}$-in. rivets in a $\frac{7}{8}$-in. strap on a pitch of $8\frac{3}{8}$ in.

o. 1263

TESTING SUCTION GAS PRODUCERS WITH A KOERTING EJECTOR

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Non-Member

The method of testing the suction gas producer herein described, and the forms for computation given in the Appendix to the paper, have been used by the writers to advantage in their gas-producer tests in the mechanical engineering laboratory of the University of Illinois. The method of testing has reduced the labor of running such tests to a minimum, and the forms for computation have greatly reduced the labor and tedium of the calculations.

2 The tests were made on an Otto suction gas producer rated at 60 h.p. and 8000 cu. ft. of gas per hour. The plant as originally installed consisted of the producer *A* (Fig. 1), the wet scrubber *B*, the gas receiver *C*, and a 22-h.p. engine. In order to facilitate the testing of the plant the connection to the 22-h.p. engine was blanked and a Schutte-Koerting steam ejector of 12,000-cu. ft. hourly capacity was placed in the gas main at *F*. This ejector was used to draw the gases from the producer and deliver them to the wet scrubber *G*, where the steam used by the ejector was condensed.

3 The condensed steam and condensing water passed out at the overflow *M*, while the gases passed out through the separator *N* and into the dryer *H*, constructed from a gas bell, or holder, filled with straw, and used to separate the suspended moisture from the gases before they entered the meters *I* and *J*. The meters were of 8000 and 3500-cu. ft. hourly capacity respectively, and were connected

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Presented at the Annual Meeting, New York, (December 1909) of The American Society of Mechanical Engineers.

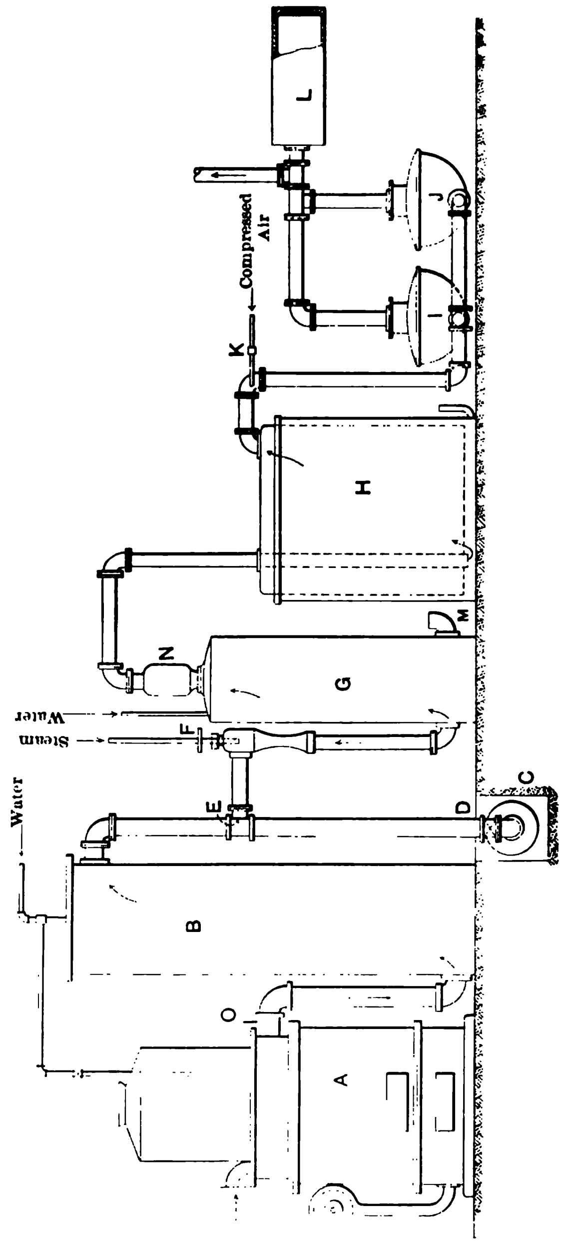

Fig. 1 General Arrangement of Suction Gas Producer Plant Tested with Koerting Ejector

in parallel for capacities greater than 8000 cu. ft. per hour, the larger meter alone being used for lower capacities. From the meters the gases were discharged into the atmosphere above the roof of the laboratory.

4 A gage box L was adapted to receive thin plates with orifices, and was used in calibrating the meters, by means of air. The meters having been blanked from the gas main, compressed air was admitted at K, and expanding passed through the meter to be calibrated and out at the orifice in L. The data for the orifice was taken from the paper, on the Measurement of Air Flowing into the Atmosphere through Circular Orifices in Thin Plates and under Small Differences of Pressure, by R. J. Durley[1]. After the calibration, the inlet to the box was blanked.

5 The producer is of the contained vaporizer type, with grate and without charging bell, the specifications stating that it is only to be used twelve hours at a time. During some of the earlier tests the cast-iron vaporizer was cracked. A steam jet was then used to sup-

TABLE 1 TEMPERATURES IN FUEL BED

Time	Zone No.	Temp. 3 in. From Near Wall F°	Temp. at Center, F°	Temp. 3 in. From Far Wall F°
10:05–10:10 a.m.	1	2100	2037	2025
10:25–10:30	2	2350	2225	2275
10:43–10:55	3		2200	2400

ply the moisture, and the vaporizer was blanked off. The weight of steam was measured by passing the jet through a calibrated orifice in a thin plate.

6 The test was started with the producer full and with a clean fuel bed. The coal fired during the test was weighed and at the end of the test the fire was cleaned, the fuel bed being brought as near to the starting condition as possible, and the producer filled. In order that the error in determining the weight of coal fired in this manner might be known, the producer when cold was filled a number of times, and the weight of coal required was noted. The average of these weights was taken to be the true weight of coal required to fill the producer, the probable error in filling with a given weight of coal being estimated from these results. In running it was endeavored to make

[1] Trans., Vol. 27, No. 1098.

the tests of such duration as to bring the probable error of filling down to about two or three per cent.

7 The temperature of the gas leaving the producer was taken at *O* by means of a platinum-rhodium thermo-couple and a Siemens & Halske milivoltmeter, calibrated to read direct in degrees centigrade. The temperatures in the fuel bed were taken with Hoskins thermo-couples and galvanometer, the latter reading in degrees Fahrenheit. Other temperatures were taken with mercury thermometers.

8 The temperatures in the fuel bed were taken in three horizontal zones 10 in., 18 in. and 24 in., respectively, above the grate. In each zone readings were taken 3 in. from the lining on each side, and in the center of the fuel bed. The results are given in Table 1.

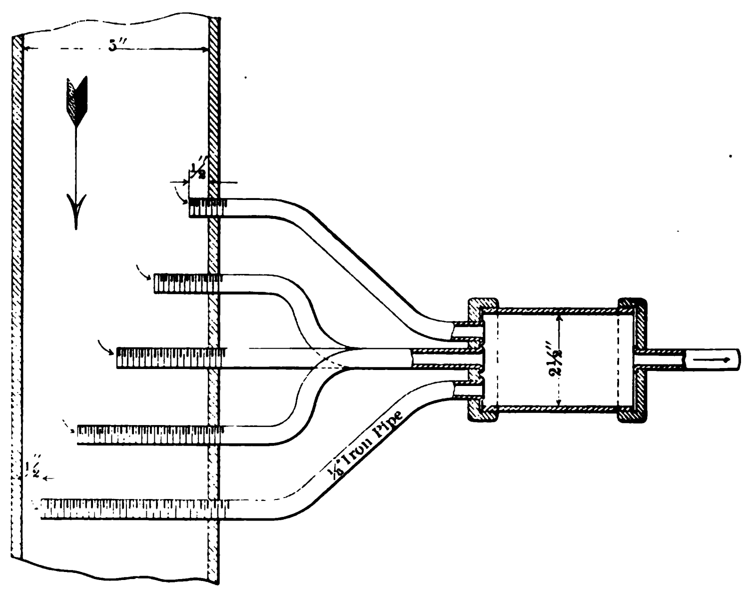

FIG. 2 SAMPLING TUBE FOR TAKING SAMPLES OF GAS CONTINUOUSLY

9 By means of the sampling tube illustrated in Fig. 2, samples of gas were taken continuously for test by a Junkers calorimeter and for analysis by Hempel apparatus. The results of the analyses are given in Table 2.

10 As already stated, the weight of steam fed to the producer was determined by the use of a calibrated orifice. By means of a small laboratory aspirator, a sample of the gas leaving the producer was drawn successively through a calcium chloride tube and a small

gas meter, the weight of moisture being determined by the calcium chloride tube and the volume by the meter. The per cent of moisture determined by this method was used merely as a check, the percentage used in the computations being obtained by calculating the weight of water decomposed from the analysis of the gases and the analysis of the fuel. The difference between this quantity and the total weight of moisture carried into the producer, gives the weight of the moisture in the gas leaving the producer.

11 The volume of gas generated by the producer, and measured by the meters, was also checked by computing the volume of the gas generated from the analyses of gas and coal. In the anthracite producer, where the loss of carbon in soot and tar is small, probably not over 1 per cent, this offers an excellent means of checking the gas volume, and also of computing the weight of air used. The gas analysis, where continuous samples are taken by the form of sampling tube illustrated, should be accurate within 1 per cent. The greatest error is likely to be made in the sampling of the coal. With a fine coal, such as pea or buckwheat, and a sample representing from 10 to 20 per cent of the total weight of coal fired, the error in sampling should not exceed 2 per cent. The maximum error in determining the gas volume and the weight of air used should not exceed 5 per cent, if the error in filling the producer is 2 per cent. The probable error is therefore much less. In most of the tests, the volume of gas computed from analysis has checked within 5 per cent the volume determined by the meters. The meters are known to be accurate well within 2 per cent.

TABLE 2 GAS ANALYSIS BY VOLUME

No.	Time	Per Cent CO_2	Per Cent O_2	Per Cent CO	Per Cent CH_4	Per Cent H_2	Per Cent N_2	B.T.U.
1	6:23– 9:05 a.m.	5.7	0.5	22.9	2.1	12.2	56.6	134
2	9:10–10:37	4.1	0.2	27.9	1.6	11.1	55.1	142
3	10:39–12:25 p.m.	3.3	0.1	28.4	1.5	9.5	57.2	137
4	12:30– 2:52	4.3	0.2	26.9	1.8	10.6	56.2	139
5	3:00– 4:35	3.6	0.1	28.6	1.8	9.0	56.9	139
6	4:40– 6:05	4.1	0.3	27.4	1.8	10.0	56.4	138
Average		4.20	0.23	27.01	1.77	10.40	56.40	138.1

12 In the testing of large producers of the bituminous type, it is often difficult to measure the gas volume by any mechanical means.

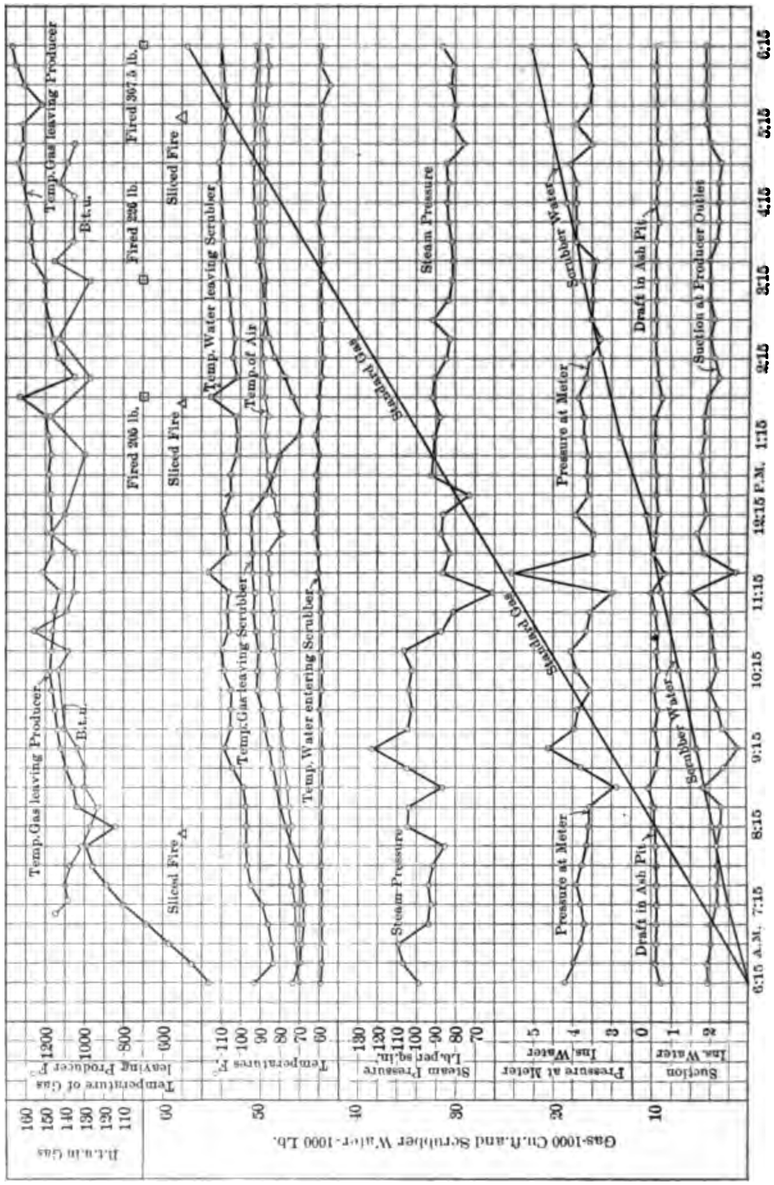

FIG. 3 GRAPHICAL LOG OF TEST OF SUCTION GAS PRODUCER

In such cases, if the carbon lost in the soot and tar is estimated from a sample of the soot and tar, and this amount deducted from the total weight of carbon in the coal, the volume may then be computed from the analyses of the gas and coal, and may be relied upon within 5 per cent, provided the sampling is accurate.

13 In order to facilitate computations, we have prepared three separate forms, or rather two forms and a guide sheet. Form 1 is used only for the presentation of the results of the tests. Form 2 contains all items used in the computations, while Form 3 is the guide sheet containing all of the formulae and their derivation. The items of Form 3 are arranged in the order of computation. In following out this method, the average corrected quantities are taken from the original data sheets and placed on Form 2. The computations are then made by following Form 3. After Form 2 is completed, the results are transferred to Form 1.

14 Referring to Form 1, Item 46, it will be noted that the total ash and refuse is much less than the weight of ash alone that would be obtained by computing from the analysis. This is due to the difficulty in cleaning the ash out of the fuel bed, and partly to the loss of ash in the form of dust, which is carried over into the scrubber. In this particular coal, which had very little tendency to clinker, the ash was soft and fine so that it packed in and filled the interstices between the live coals. A small amount of clinker was formed on the sides.

15 This tendency of the ash to pack in the fuel bed, while it prevents the accurate determination of the actual weight of ash, does not, it is believed, materially affect the determination of the weight of coal as fired, for the reason just given; that is, while the fuel bed may contain as much carbon at the start as at the close, the bed is much more compact due to the ash. The weight of ash and refuse is valuable principally for the determination of the unburned carbon lost through the grate.

16 Item 66, dry coal per sq. ft. of grate area per hour, is high; while the producer was operating only at about 4800 cu. ft. per hour capacity, this was considerably above its actual capacity. If the fuel had contained a fusible ash the results as shown on Form 1 and the graphical log Fig. 3 would have been practically impossible.

17 The heat balance, Form 1, shows the unaccounted-for loss to be 4.4 per cent. This includes radiation and conduction, which for this test probably amounts to between 2 and 3 per cent. By referring to Form 2, Item 126, it will be seen that the volume of standard

gas, computed from the analysis of the gas and the analysis of the coal, checks within about 2.3 per cent of the volume of standard gas as given by the meters, Item 125.

18 The graphical log sheet (Fig. 3) illustrates the uniformity of conditions that were maintained throughout the test.

19 Permission for running the producer tests was obtained through Prof. L. P. Breckenridge, the results being presented through the courtesy of Dean W. F. M. Goss, of the University of Illinois.

APPENDIX

FORM 1 RESULTS OF GAS PRODUCER TRIALS

1	Test number	25
2	Made by.....C. M. Garland and A. P. Krats	
3	At.....University of Illinois	
4	Kind of producer.....Otto	
5	To determine.....,Efficiency	
6	Principal conditions governing trial.....Uniform load	
7	Kind of fuel.....Scranton-Anthracite	
8	Kind of grate.....Plain	
9	Method of starting and stopping test.....Alternate	
10	Type of producer.....Suction	
11	Form of blower-ejector.....Schutte & Koerting	
12	Date of trial	5-29-1909
13	Duration of trial	12 hr.

DIMENSIONS AND PROPORTIONS

14	Dimensions of grate, ft	1.25 by 1.33
15	Grate area, sq. ft	1.663
16	Mean diameter of fuel bed, ft	1.545
17	Depth of fuel bed, ft	2.21
18	Area of fuel bed, sq. ft	1.877
19	Height of discharge pipe above grate, ft	2.875
20	Approximate width of air spaces in grate, in	0.5
21	Area of air space, sq. ft	0.722
22	Proportion of air space to whole grate area, per cent	43.3
23	Area of discharge pipe, sq. ft	0.165
24	Water heating surface in vaporizer, sq. ft	
25	Outside diameter of shell, ft	2.833
26	Length of shell from base to top of magazine, ft	7.125
27	Ratio of water heating surface to grate area, — to 1	
28	Ratio of minimum draft area to grate area, 1 to	48.8

AVERAGE PRESSURES

29	Draft in ashpit, inches, water	0.61
30	Suction at producer outlet, inches, water	2.04
31	Pressure at meters, inches, water	3.76
32	Corrected barometer reading	29.15
32.1	Steam pressure, lb. per sq. in. gage	90.5

AVERAGE TEMPERATURES

33	Of fire room, deg. fahr.	82.2
34	Of steam leaving vaporizer, deg. fahr.	212
35	Of feed water entering vaporizer, deg. fahr.	
36	Of overflow from vaporizer, deg. fahr.	
37	Of water entering scrubber, deg. fahr.	57.8
38	Of water leaving scrubber, deg. fahr.	103.6
39	Of gases leaving producer, deg. fahr.	1108
40	Of gases leaving scrubber, deg. fahr.	84.3
41	Of gases entering meter, deg. fahr.	68.0

FUEL

42	Size and condition ... Pea–Clean	
43	Weight of coal as fired, lb.	798.5
44	Percentage of moisture in coal	2.75
45	Total weight of dry coal fired, lb.	776.5
46	Total ash and refuse, lb.	85.0
47	Quality of ash and refuse	
48	Total combustible consumed, lb.	614
49	Percentage of ash and refuse in dry coal	10.9

PROXIMATE ANALYSIS OF COAL

50	Fixed carbon	78.45
51	Volatile matter	5.99
52	Moisture	2.75
53	Ash	12.81
54	Sulphur, separately determined	1.10

ULTIMATE ANALYSIS OF DRY COAL

55	Carbon, C	79.84
56	Hydrogen, H_2	2.67
57	Oxygen, O_2	2.37
58	Nitrogen, N_2	0.82
59	Sulphur, S	1.13
60	Ash	13.17
61	Moisture in sample of coal as received	2.75

ANALYSIS OF DRY ASH AND REFUSE

62	Carbon, per cent	38.80
63	Earthy matter, per cent	61.20
	a SiO	
	b $\left\{\begin{array}{l} Al_2O_3 \\ Fe_2O_3 \end{array}\right.$	
	c MgO	
	d CaO	

FUEL PER HOUR

64	Dry coal fired per hr., lb.	64.7
65	Combustible consumed per hr., lb.	51.2
66	Dry coal per sq. ft. of grate area per hr., lb.	38.8
67	Combustible per sq. ft. of grate area per hr., lb.	30.7
68	Dry coal per sq. ft. of fuel bed per hr., lb.	34.5
69	Combustible per sq. ft. of fuel bed per hr., lb.	27.3
70	Rate of descent of dry coal through fuel bed, lb. per ft. per sq. ft. per hr.	15.6
71	Rate of descent of combustible through fuel bed, lb. per ft. per sq. ft. per hr.	12.4

CALORIFIC VALUE OF FUEL

72	Calorific value by oxygen calorimeter per lb. dry coal, B.t.u.	13,040
73	Calorific value by oxygen calorimeter per lb. of combustible B.t.u.	15,700
74	Calorific value by analysis per lb. dry coal, B.t.u.	13,125
75	Calorific value by analysis per lb. of combustible, B.t.u.	15,800

WATER

76	Total weight of water[1] fed to vaporizer, lb.	267.8
77	Total weight of overflow from vaporizer, lb.	
78	Water[1] actually evaporated in vaporizer, lb.	267.8
79	Total weight of water fed to producer, lb.	341.5
	a From vaporizer[1]	267.8
	b In air	51.7
	c In coal	22.0
80	Total weight of water decomposed	218.2
81	Total weight of water in gas leaving producer, lb.	123.3
82	Ratio of water decomposed to water supplied	0.639
83	Weight of water decomposed per lb. gas generated, lb.	0.0558
84	Weight of water decomposed per lb. of dry coal fired, lb.	0.281
85	Weight of water decomposed per lb. of combustible consumed, lb.	0.355
86	Weight of water decomposed per lb. of air supplied, lb.	0.0702
87	Weight of water supplied per lb. of dry coal fired, lb.	0.440
88	Weight of water supplied per lb. of combustible consumed, lb.	0.556
89	Weight of water supplied per lb. of dry air used, lb.	0.1097
90	Total weight of scrubber water, lb.	22,200

WATER PER HOUR

91	Water evaporated per hr. in vaporizer, lb.	
92	Water evaporated per hr. per sq. ft. of water heating surface in vaporizer, lb.	
93	Weight of water decomposed per hr., lb.	18.2

[1] Steam fed to vaporizer.

94	Total weight of water fed to producer per hr., lb.	28.5
95	Weight of scrubber water used per hr., lb.	1850

QUANTITY OF AIR

96	Per cent of moisture in air, per cent of dry air	1.66
97	Total weight of dry air, lb.	3112
98	Total weight of dry air per hr., lb.	259.2
99	Weight of dry air used per lb. of dry coal fired, lb.	4.01
100	Weight of dry air used per lb. of combustible consumed, lb.	5.07
101	Weight of dry air used per lb. of dry gas generated, lb.	0.79[illegible]

GAS

102	Per cent moisture in gas leaving producer, per cent of dry gas	3.15
103	Per cent of soot and tar in gas leaving producer	
104	Calorific value of standard gas from analysis (high value) B.t.u. per cu. ft.	138.1
105	Calorific value of standard gas from calorimeter (high value) B.t.u. per cu. ft.	137.3
106	Specific weight of standard gas, lb. per cu. ft.	0.068[illegible]
107	Specific heat of dry gas leaving producer	0.328[illegible]
108	Carbon ratio C/H	14.07
109	Total volume standard gas, per cu. ft	57,500
110	Volume of standard gas per hr., per cu. ft.	4,795
111	Volume of standard gas per lb. of dry coal	74.1
112	Volume of standard gas per lb. of combustible	93.7
113	Total weight of standard gas, lb.	3912
114	Weight of standard gas per hr., lb.	326
115	Weight of standard gas per lb. of dry coal fired, lb.	5.03
116	Weight of standard gas per lb. of combustible consumed, lb.	6.37

GAS ANALYSIS BY VOLUME

117	Carbon dioxide, CO_2	4.20
118	Carbon monoxide, CO	27.01
119	Oxygen, O_2	0.23
120	Hydrogen, H_2	10.40
121	Marsh gas, CH_4	1.77
122	Olefiant gas, C_2H_4	
123	Sulphur dioxide, SO_2	
124	Hydrogen sulphide, H_2S	
125	Nitrogen, N_2, by difference	56.40

EFFICIENCY

126	Grate efficiency, per cent	95.3
127	Hot gas efficiency, based on high heating value, per cent	90.9
128	Cold gas efficiency, based on high heating value, per cent	78.3

EFFICIENCY BASED ON COMBUSTIBLE

128a	Hot gas efficiency, based on high heating value	95.4
128b	Cold gas efficiency, based on high heating value	82.2

COST OF GASIFICATION

129	Cost of fuel per ton delivered in producer room	
130	Cost per 1000 cu. ft. of standard gas, cents	
131	Cu. ft. scrubber water per 1000 cu. ft. gas	6.18

POKING

132	Method of poking	From top, slicing from bottom
133	Frequency of poking	Three times during run

FIRING

134	Method of firing	Hand
135	Average intervals between firing	Twice during run
136	Average amount of fuel charged each time, lb.	250

HEAT BALANCE

Debit	B.t.u.
a Total heat supplied per lb. dry coal	13,040
b Total heat supplied by air per lb. dry coal	19
c Total heat supplied by moisture in air per lb. dry coal	
d Total heat supplied by moisture in coal per lb. dry coal	
e Total heat supplied as sensible heat in coal per lb. dry coal	
f Total heat supplied by water[1] in vaporizer per lb. dry coal	385
Total	13,444

Credit	B.t.u.	Per Cent
a Heat contained as sensible heat in dry gas	1725	12.8
b Heat contained in moisture	262	2.0
c Heat contained in dry gas (heat of combustion)	10,240	76.2
d Heat in unburned carbon	618	4.6
e Heat contained in ash and refuse as sensible heat		
f Heat lost in overflow from vaporizer		
g Heat lost in radiation and conduction	599	4.4
Total	13,444	100.0

[1] Supplied in steam.

FORM 2 RESULTS OF GAS PRODUCER TRIALS

NO. OF TEST 25. DATE 5/29/09. TIME OF START 6.15 A.M.
TIME OF STOP 6.15 P.M. DURATION OF TRIAL 12 HR.
KIND OF FUEL SCRANTON-ANTHRACITE

DIMENSIONS AND PROPORTIONS

1	Dimensions of grate, ft.	1.25 by 1.33
2	Grate area, sq. ft.	1.663
3	Mean diameter of fuel bed, ft.	1.545
4	Depth of fuel bed, ft.	2.21
5	Area of fuel bed, sq. ft.	1.877
6	Height of discharge pipe above grate, ft.	2.875
7	Approximate width of air spaces in grate, in.	0.5
8	Area of air space, sq. ft.	0.722
9	Ratio of air space to whole grate area, per cent	4.33
10	Area of discharge pipe, sq. ft.	0.165
11	Water heating surface in vaporizer, sq. ft.	
12	Outside diameter of shell, ft.	2.833
13	Length of shell from base to top of magazine, ft.	7.125
14	Ratio of water heating surface to grate area	— to 1
15	Ratio of minimum draft area to grate area	1 to 48.8

AVERAGE PRESSURES

16	Average barometer reading, inches Hg.	29.258
17	Average corrected barometer reading, inches Hg.	29.152
18	Draft in ash pit, inches water.	0.61
19	Suction at producer outlet, inches water.	2.04
20	Absolute pressure at producer outlet, inches Hg.	29.00
21	Suction[1] at orifice, inches water	90.5
22	Absolute pressure[1] at orifice, inches Hg.	104.8
23	Pressure at meters, inches water.	3.76
24	Absolute pressure at meters, inches Hg.	29.43
25	Vapor pressure at meters, inches Hg.	0.685
26	Dry gas pressure at meters, inches Hg.	28.75
27	Suction at meter for dryer, inches water.	2.04
28	Absolute pressure at meter for dryer, inches Hg.	29.00

AVERAGE TEMPERATURES

29	At barometer, deg. fahr.	78.0
30	Of fire room, deg. fahr.	82.2
31	Of fire room, deg. abs. fahr.	542.2
32	Of steam, deg. fahr.	212
33	Of feed water entering vaporizer, deg. fahr.	
34	Overflow from vaporizer, deg. fahr.	
35	Rise in vaporizer, deg. fahr.	

[1]Steam ressnrep

36	Of water entering scrubber, deg. fahr	57.8
37	Of water leaving scrubber, deg. fahr	103.6
38	Rise in scrubber, deg. fahr	45.8
39	Of gases leaving producer, deg. fahr	1108
40	Of gases leaving producer, deg. abs. fahr	1568
41	Of gases leaving first scrubber, deg. fahr	84.3
42	Of gases leaving first scrubber, deg. abs. fahr	544.3
43	Drop in temperature of gases in scrubber, deg. fahr	1023.7
44	Of gases entering meters, deg. fahr	68.0
45	Of gases entering meters, deg. abs. fahr	528
46	Of gas at meter at dryer, deg. fahr	80.0
47	Of gas at meter at dryer, deg. abs. fahr	540

FUEL

48	Size and condition .... Pea, Clean	
49	Weight of coal as fired, lb	798.5
50	Percentage of moisture in coal	2.75
51	Total weight of dry coal fired, lb	776.5
52	Total ash and refuse, lb	85.0
53	Quality of ash and refuse	
54	Total weight of combustible, lb	614
55	Percentage of ash and refuse in dry coal, per cent	10.9

PROXIMATE ANALYSIS OF COAL

56	Fixed carbon, per cent	78.45
57	Volatile matter, per cent	5.99
58	Moisture, per cent	2.75
59	Ash, per cent	12.81
60	Sulphur, separately determined, per cent	1.10

ULTIMATE ANALYSIS OF DRY COAL

61	Carbon, C, per cent	79.84
62	Hydrogen, H_2, per cent	2.67
63	Oxygen, O_2, per cent	2.37
64	Nitrogen, N_2, per cent	0.82
65	Sulphur, S, per cent	1.13
66	Ash, per cent	13.17
67	Moisture in sample of coal as received, per cent	2.75

ANALYSIS OF DRY ASH AND REFUSE

68	Carbon, per cent	38.80
69	Earthy matter, per cent	61.20
	a SiO_2	
	b $\left\{\begin{array}{l}Al_2O_3\\Fe_2O_3\end{array}\right.$	
	c MgO	
	d CaO	

FUEL PER HOUR

70	Dry coal fired per hr., lb.	64.7
71	Combustible consumed per hr., lb.	51.2
72	Dry coal sq. ft. of grate area per hr., lb.	38.8
73	Combustible per sq. ft. of grate area per hr., lb.	30.7
74	Dry coal per sq. ft. of fuel bed per hr., lb.	34.5
75	Combustible per sq. ft. of fuel bed per hr., lb.	27.3
76	Rate of descent of dry coal through fuel bed, lb. per ft. per sq. ft. per hr.	15.6
77	Rate of descent of combustible through fuel bed, lb. per ft. per sq. ft. per hr.	12.4

CALORIFIC VALUE OF FUEL

78	Calorific value by oxygen calorimeter per lb. dry coal, B.t.u..	13,040
79	Calorific value by oxygen calorimeter per lb. combustible, B.t.u.	15,700
80	Calorific value by analysis, per lb. dry coal, B.t.u.	13,125
81	Calorific value by analysis, per lb. combustible, B.t.u.	15,800

WATER

82	Total[1] weight fed to vaporizer, lb.		267.8
83	Total weight of overflow, lb.		
84	Water[1] actually evaporated in vaporizer, lb.		267.8
85	Weight of water fed to producer,		
	a From vaporizer[1]	267.8	
	b In air	51.7	
	c In coal	22.0	
	Total		341.5
86	Total weight of water decomposed from analysis, lb.		218.2
87	Total weight of water decomposed as used in calculations, lb..		218.2
88	Total weight of moisture in gas leaving producer, lb.		123.3
89	Ratio of water decomposed to water supplied		0.639
90	Weight of water decomposed per lb. of gas generated, lb.		0.0558
91	Weight of water decomposed per lb. of dry coal fired, lb.		0.281
92	Weight of water decomposed per lb. of combustible consumed, lb.		0.355
93	Weight of water decomposed per lb. of air supplied		0.0702
94	Weight of water supplied per lb. of dry coal fired, lb.		0.440
95	Weight of water supplied per lb. of combustible consumed, lb..		0.556
96	Weight of water supplied per lb. of air used, lb.		0.1097
97	Total weight of scrubber water, lb.		22,200
98	Total weight of water absorbed by dryer, grams		15

[1] Steam fed to vaporizer.

WATER PER HOUR

99	Water evaporated per hr. in vaporizer, lb..................	
100	Water evaporated per hr. per sq. ft. of water heating surface in vaporizer, lb....................................	
101	Weight of water decomposed per hr., lb.	18.2
102	Total weight of water fed to producer per hr., lb.............	28.5
103	Weight of scrubber water used per hr. lb..................	1850

QUANTITY OF AIR

104	Relative humidity of air, per cent........................	73
105	Per cent of moisture contained in air, per cent by weight of dry air..	1.66
106	Total weight of dry air by analysis, lb.....................	3112
107	Total weight of dry air by orifice, lb.......................	
108	Total weight of dry air as used in calculations, lb............	3112
109	Weight of dry air per hr. from total used in calculations.......	259.2
110	Weight of dry air used per lb. of dry coal fired, lb............	4.01
111	Weight of dry air used per lb. of combustible consumed, lb. ..	5.07
112	Weight of dry air used per lb. of dry gas generated, lb........	0.796

GAS

113	Volume of gas passing through meter at dryer, cu. ft.........	31.06
114	Volume of standard gas passing through meter at dryer, cu. ft.	28.0
115	Total weight of gas passing through dryer meter, lb..........	1.9
116	Percentage of moisture in gas leaving producer, from dryer, per cent dry gas....................................	1.74
117	Percentage of moisture in gas leaving producer, from water fed to producer, per cent dry gas..........................	3.15
118	Percentage soot and tar in gas leaving producer.............	
119	Calorific value per cu. ft. of standard gas from analysis B.t.u. (high value)..	138.1
120	Calorific value per cu. ft. of standard gas from calorimeter, B.t.u. (high value).................................	137.3
121	Specific weight of standard gas, lb. per cu. ft................	0.0680
122	Specific heat of dry gas leaving producer...................	0.3281
123	Carbon ratio C/H....................................	14.07
124	Total volume of gas from meters, cu. ft....................	60,630
125	Total volume of standard gas, from meters, cu. ft...........	57,500
126	Total volume of standard gas, from analysis cu. ft..........	56,200
127	Total volume as used in calculations, cu. ft.................	57,500
128	Volume of standard gas per hr. from total used in calculations.	4795
129	Volume of standard gas per lb. of dry coal from total used in calculations, cu. ft..................................	74.1
130	Volume of standard gas per lb. of combustible from total used in calculations, cu. ft...............................	93.7
131	Total weight of standard gas from total used in calculations, lb...	3912

132	Weight of standard gas per hr., lb.	326
133	Weight of standard gas per lb. of dry coal, lb.	5.03
134	Weight of standard gas per lb. of combustible, lb.	6.37

GAS ANALYSIS BY VOLUME

135	Carbon dioxide, CO_2	4.20
136	Carbon monoxide, CO	27.01
137	Oxygen, O_2	0.23
138	Hydrogen, H_2	10.40
139	Marsh gas, CH_4	1.77
140	Olefiant gas, C_2H_4	
141	Sulphur dioxide, SO_2	
142	Hydrogen sulphide, H_2S	
143	Nitrogen, N_2 by difference	56.40

GAS ANALYSIS BY WEIGHT

144	Carbon dioxide, CO_2	7.16
145	Carbon monoxide, CO	29.25
146	Oxygen, O_2	0.29
147	Hydrogen, H_2	0.81
148	Marsh gas, CH_4	1.12
149	Olefiant gas, C_2H_4	
150	Sulphur dioxide, SO_2	
151	Hydrogen sulphide, H_2S	
152	Nitrogen, N_2, by difference	61.37

EFFICIENCY

153	Grate efficiency, per cent	95.3
154	Hot gas efficiency, based on high heating value, per cent	90.9
155	Cold gas efficiency, based on high heating value, per cent	78.3

EFFICIENCY BASED ON COMBUSTIBLE

155*a*	Hot gas efficiency, based on high heating value, per cent	95.4
155*b*	Cold gas efficiency, based on high heating value, per cent	82.2

COST OF GASIFICATION

156	Cost of fuel per ton delivered in producer room	
157	Cost per 1000 cu. ft. of standard gas, cents	
158	Cu. ft. scrubber water per 1000 cu. ft. standard gas	6.18

POKING

159	Method of poking .... From top, slicing from bottom	
160	Frequency of poking .... Three times during test	

FIRING

161	Method of firing .... Hand	
162	Average intervals between firings .... Twice during run	
163	Average amount of fuel charged each time	250

HEAT BALANCE

Debit	B.T.U.	
a Total heat supplied per lb. dry coal..................	13,040	
b Total heat supplied by air per lb. dry coal.....	19	
c Total heat supplied by moisture in air per lb dry coal....		
d Total heat supplied by moisture in coal...............		
e Total heat supplied as sensible heat in coal............		
f Total[a] heat supplied in vaporizer water...............	385	
Total..	13,444	

Credit	B.T.U.	Per Cent
a Heat contained as sensible heat in dry gas..........	1725	12.8
b Heat contained in moisture..........................	262	2.0
c Heat contained in dry gas (heat of combustion).......	10,240	76.2
d Heat in unburned carbon...........................	618	4.6
e Heat contained as sensible heat in ash and refuse		
f Heat lost in overflow from vaporizer...		
g Radiation and conduction, by difference..............	599	4.4
Total..	13,444	100.0

FORM 3 GUIDE SHEET CONTAINING ALL FORMULAE AND THEIR DERIVATION

The item numbers refer to the items of Form 2, and are arranged in the order of computation.

Item 4. "Depth of fuel bed" is to a certain extent arbitrary. In order that the term may have a fixed and definite meaning we will define it as the distance between the upper edge of the ash zone and that section of the fuel bed from which the gases separate and leave the fuel. The upper edge of the ash zone can ordinarily be readily determined by inspection.

Item 16. This reading is the average of the barometer readings for the test and is not corrected.

Item 17. Item 16 corrected. The following formula may be used:

Let H = corrected barometer reading.

t = temperature, deg. fahr.

h = barometer reading corresponding to temperature t.

Then $H = h\,(1.00254 - 0.000079t)$

Item 17. = Item 16 (1.00254 − 0.000079 × Item 29)

Item 18. = Observed.

Item 19. = Observed.

Item 20. = Item 17 − Item 19 × 0.0735

Item 21. = Observed.

Item 22. = Item 17 − Item 21 × 0.0735

[a] Supplied in steam.

Item 23. = Observed.
Item 24. = Item 17 + Item 23 × 0.0735
Item 25. = Taken from steam tables using temperature in Item 44, 1 lb. per sq. in. = 2.04 in. Hg.
Item 26 = Item 24 − Item 25
Item 27. = Observed.
Item 28. = Item 17 − Item 27 × 0.0735
Items 29 to 48. The observed temperatures should be corrected from the calibration curves before being placed in Form 2. The absolute temperature = the observed temperature + 460 deg.
Item 39. This item is observed in deg. cent. and should be transferred into deg. fahr.

$$\text{Deg. fahr.} = \frac{9}{5}\,\text{deg. cent.} + 32$$

Each observation must be transferred.
Item 50. Taken from Item 67.

$$\textit{Item}\ 51.\quad \text{Item } 49\left(1 - \frac{\text{Item } 50}{100}\right)$$

Item 52. Taken from ash sheet, correction being made for any moisture taken up in the ashpit.
Item 54. In these tests the total weight of combustible consumed will be taken as the total weight of dry coal fired.
The weight of ash computed from the analysis − the weight of nitrogen − $\frac{9}{8}$ × the weight of oxygen − the weight of carbon contained in the ash and refuse =

$$\text{Item } 51 - \frac{\text{Item } 51 \times \text{Item } 66}{100} - \frac{\text{Item } 51 \times \text{Item } 64}{100} - \frac{\frac{9}{8}\ \text{Item } 51 \times \text{Item } 63}{100} - \frac{\text{Item } 52 \times \text{Item } 68}{100}$$

Therefore,

$$\text{Item } 54 = \text{Item } 51\left[1 - \frac{\text{Item } 66 + \text{Item } 64 + \frac{9}{8}\ \text{Item } 63}{100}\right] - \frac{\text{Item } 52 \times \text{Item } 68}{100}$$

$$\textit{Item}\ 55. = \frac{\text{Item } 52 \times 100}{\text{Item } 51}$$

Items 56 to 69. From chemist.
Items 69, *a*, *b*, *c*, *and d*. The ultimate analysis of the ash will be made only in special cases to obtain data on the formation of clinker.

$$\textit{Item}\ 70. = \frac{\text{Item } 51}{\text{hours}}$$

$$\textit{Item}\ 71. = \frac{\text{Item } 54}{\text{hours}}$$

Item 72. $= \dfrac{\text{Item 70}}{\text{Item 2}}$

Item 73. $= \dfrac{\text{Item 71}}{\text{Item 2}}$

Item 74. $= \dfrac{\text{Item 70}}{\text{Item 5}}$

Item 75. $= \dfrac{\text{Item 71}}{\text{Item 5}}$

Item 76. "The rate of descent of dry coal through the fuel bed," or "the dry coal per cu. ft. of fuel bed per hour," which is the same, offers a means of comparing the rate of gasification in different producers that seems to be better adapted for the purpose than the expressions taken from boiler pratice, viz: "coal per sq. ft. of grate area," or "coal per sq. ft. of fuel bed," the latter having been used in producer practice.

Item 76. $= \dfrac{\text{Item 74}}{\text{Item 4}}$

Item 77. $= \dfrac{\text{Item 75}}{\text{Item 4}}$

Item 78. Taken from chemist's report.

Item 79. $= \dfrac{\text{Item 78} \times \text{Item 51} - \text{Item 52} \times \text{Item 68} \times 145.40}{\text{Item 54}}$

Item 80. $= \{\text{Item 61} \times 145.40 + \text{Item 65} \times 40.00 + [\text{Item 62} - \tfrac{1}{8}\text{ of Item 63}] \times 620.00\}$

Item 81. $= \dfrac{\text{Item 80} \times \text{Item 51} - \text{Item 52} \times \text{Item 68} \times 145.40}{\text{Item 54}}$

Item 113. Total volume of gas passing through meter at dryer. Observed.

Item 114. Total volume of standard gas passing through meter at dryer, neglecting the effect of moisture.

Let p_1 = absolute pressure in inches Hg. at dryer meter.
t_1 = absolute temperature, deg. fahr. at dryer meter.
v_1 = total volume of gas passing through meter.
P, V, and T, be the condition of standard gas.
P = 30 in. Hg.
$T = 460 + 62 = 522$

Then

$$\frac{p_1 v_1}{t_1} = \frac{PV}{T}$$

$$\text{or } V = \frac{p_1 v_1 T}{P t_1} = \frac{p_1 v_1 \times 522}{30 t_1} = \frac{17.4\, p_1 v_1}{t_1}$$

from which the value of Item 114 follows.

Item 114. $= 17.4\ \dfrac{\text{Item 28} \times \text{Item 113}}{\text{Item 47}}$

Item 118. Not considered in these tests.

Item 119. One cubic foot of standard gas, that is, gas at a temperature of 62 deg. fahr. or 522 deg. abs. and a pressure of 30 in. Hg., gives up on combustion, when the products of combustion are brought back to this temperature and the moisture is condensed, the following heat quantities

H_2 = 328 B.t.u. per cu. ft. of standard gas.
C_2H_4 = 1480 B.t.u. per cu. ft. of standard gas.
CO = 319 B.t.u. per cu. ft. of standard gas.
CH_4 = 1010 B.t.u. per cu. ft. of standard gas.

Item 120. This quantity is the average of all the calorimeter determinations. Each separate determination by the calorimeter must be computed and the heating value obtained. The following formula may be used. The calorimeter readings are taken in centigrade units with the exception of the meter readings and pressure.

Let t_2 = temperature of entering water, deg. cent.
t_1 = temperature leaving water, deg. cent.
r = rise in temperature of water, deg. cent.
W = weight of water used during the intervals = 8 litres for all tests.
G_1 = cu. ft. of gas used from meter.
t^g = temperature of entering gas, deg. cent.
p^g = pressure entering gas inches Hg. absolute, corrected for vapor pressure of water (see Item 25).
H = heating value per cu. ft. of standard gas (62 deg. fahr. or 16.7 deg. cent. and 30 in. Hg.)
t^s = temperature of standard gas = 62 deg. fahr. or 16.7 deg. cent.
p^s = pressure of standard gas = 30 in. Hg.
G^s = cu. ft. of standard gas.

$$\frac{G_1\, p_g}{t_g} = \frac{G_s\, p_s}{t_s}$$

$$G_s = \frac{G_1 \times p_g \times t_s}{t_g \times p_s}$$

Where t_g and t_s are in absolute deg. cent., $t_1 - t_2 = r$
Total heat per cu. ft. standard gas in B.t.u. = H
Total heat absorbed by water = $W \times r$

$$H = \frac{W \times r \times 3.968}{G_s} = \frac{W \times r \times 3.968}{\left(\frac{G_1 \times p_g \times t_s}{t_g \times p_s}\right)}$$

$$= \frac{8 \times r \times 3.968 \times t_g \times 30}{G_1 \times p_g \times (16.7 + 273)} =$$

$$\frac{t_g \times r \times 3.29}{G_1 \times p_g}$$

where 3.968 is the conversion factor.

In this formula it is assumed that the exhaust products are brought back to 62 deg. fahr. This is not strictly true but the error introduced is negligible, when the error in the use of the apparatus is considered. There is another error due to the exhaust products carrying out more or less vapor of water than was brought in by the entering gas and air.

This error will also be small and may either be positive or negative depending on conditions. The entering gas will in most cases come from direct contact with water and will therefore be saturated. The air ordinarily will not be saturated. On combustion, moisture will be formed by the union of the oxygen and hydrogen, there will be a contraction in volume of the gases due to the combustion, and also a contraction or expansion due to a change in temperature after combustion. In whichever direction the change in the weight of moisture in the out-going gas from that brought in by the entering gas may occur, this change may be considered very small; for the contraction on combustion will be comparatively small, and this contraction will partly offset the unsaturated condition of the air used for combustion. Also the change in temperature of the out-going gas from that of the entering gas will be small.

The heating values as given in Items 119 and 120 are the *high values*.

The values obtained from the analysis will be more accurate and will be used in all computations.

Item 121. The specific weights of the following gases at 62 deg. and 30 in. Hg. are

$CO_2 = 0.11610$	$CH_4 = 0.04278$
$CO = 0.07362$	$C_2H_4 = 0.07370$
$O_2 = 0.08418$	$SO_2 = 0.16380$
$H_2 = 0.00530$	$H_2S = 0.08682$

$$N_2 = 0.07400$$

Item 121. = [Item 135 × 0.1161 + Item 136 × 0.07362 + Item 137 × 0.08418 + Item 138 × 0.00530 + Item 139 × 0.04278 + Item 140 × 0.0737 + Item 141 × 0.1638 + Item 142 × 0.08682 + Item 143 × 0.0740] $\frac{1}{100}$

Items 144 to 152. Calculation of the gas analysis by weight from the analysis by volume. Assume that we have one cubic foot of gas at 62 deg. fahr. and 30 in. Hg. of the following composition:

Volumetric Analysis	Specific Weights	Analysis by Weight Per Cent
$CO_4 = a$ per cent	$0.1161 = W_a$	$A = \dfrac{W_a \times a}{W}$
$CO = b$	$0.07362 = W_b$	$B = \dfrac{W_b \times b}{W}$
$O_2 = c$	$0.08418 = W_c$	$C = \dfrac{W_c \times c}{W}$
$H_2 = d$	$0.00530 = W_d$	$D = \dfrac{W_d \times d}{W}$

Volumetric Analysis	Specific Weights	Analysis by Weight Per Cent
$CH_4 = e$	$0.04278 = W_e$	$E = \frac{W_e \times e}{W}$
$C_2H_4 = f$	$0.07370 = W_f$	$F = \frac{W_f \times f}{W}$
$SO_2 = g$	$0.16380 = W_g$	$G = \frac{W_g \times g}{W}$
$H_2S = h$	$0.08682 = W_h$	$H = \frac{W_h \times h}{W}$
$N_2 = i$	$0.07400 = W_i$	$I = \frac{W_i \times i}{W}$

Where $W = [a \times W_a + b \times W_b + c \times W_c + e \times W_e \ldots. + i \times W_i]\frac{1}{100}$ = Item 121.

Item 122. The specific heats of the gases vary according to the pressure and temperature. As the pressure used throughout the experiments is atmospheric we have only to consider the variation with the temperature. The following formulae taken from Zeuner, vol. 1, page 147, give the specific heat for constant volume C_v.

$$^{1}CO_2,\ mC_v = 6.50 + 0.00774t \ldots\ldots (1)$$
$$H_2O,\ mC_v = 5.78 + 0.00572t \ldots\ldots (2)$$
$$O_2\,H_2\,N_2,\ CO,\ mC_v = 4.76 + 0.00244t \ldots\ldots (3)$$
$$mC_p - mC_v = 1.9934 \ldots\ldots (4)$$

For the specific heat of marsh gas CH_4, our other constituent, we will use the value $C_p = 0.6$. This is approximate, but as the quantity of CH_4 is small the resultant error is consequently small.

In the above formula, m is the molecular weight of the gas, t the temperature in deg. cent., and C_v the mean specific heat between zero and t deg. cent. C_p is determined from formula (4). From the above formulae, the analysis by weight as determined below and the temperature of the gases leaving the producer, the specific heat of each constituent in a unit weight of the gas may be determined. The specific heat of the gas will be the sum of the specific heats of the constituents.

Substituting the value of mC_v from formula (4), and the value of m, and changing to deg. fahr. we have from the above formulae:

$$\text{For } CO_2,\ C_p = 0.19 + 0.0000977t \ldots\ldots a$$
$$H_2O,\ C_p = 0.426 + 0.000176t \ldots\ldots b$$
$$H_2,\ C_p = 3.355 + 0.000678t \ldots\ldots c$$
$$CO,\ C_p = 0.24 + 0.0000484t \ldots\ldots d$$
$$N_2,\ C_p = 0.24 + 0.0000484t \ldots\ldots e$$
$$CH_4,\ C_p = 0.6 \ldots\ldots f$$
$$O_2,\ C_p = 0.21 + 0.0000424t \ldots\ldots g$$

[1]Mallard and Le Chatellier's formulae.

In this formula it is assumed that the exhaust products are brought back to 62 deg. fahr. This is not strictly true but the error introduced is negligible, when the error in the use of the apparatus is considered. There is another error due to the exhaust products carrying out more or less vapor of water than was brought in by the entering gas and air.

This error will also be small and may either be positive or negative depending on conditions. The entering gas will in most cases come from direct contact with water and will therefore be saturated. The air ordinarily will not be saturated. On combustion, moisture will be formed by the union of the oxygen and hydrogen, there will be a contraction in volume of the gases due to the combustion, and also a contraction or expansion due to a change in temperature after combustion. In whichever direction the change in the weight of moisture in the out-going gas from that brought in by the entering gas may occur, this change may be considered very small; for the contraction on combustion will be comparatively small, and this contraction will partly offset the unsaturated condition of the air used for combustion. Also the change in temperature of the out-going gas from that of the entering gas will be small.

The heating values as given in Items 119 and 120 are the ***high values.***

The values obtained from the analysis will be more accurate and will be used in all computations.

Item 121. The specific weights of the following gases at 62 deg. and 30 in. Hg. are

$CO_2 = 0.11610$	$CH_4 = 0.04278$
$CO = 0.07362$	$C_2H_4 = 0.07370$
$O_2 = 0.08418$	$SO_2 = 0.16380$
$H_2 = 0.00530$	$H_2S = 0.08682$

$$N_2 = 0.07400$$

Item 121. = [Item 135 × 0.1161 + Item 136 × 0.07362 + Item 137 × 0.08418 + Item 138 × 0.00530 + Item 139 × 0.04278 + Item 140 × 0.0737 + Item 141 × 0.1638 + Item 142 × 0.08682 + Item 143 × 0.0740] $\frac{1}{100}$

Items 144 to 152. Calculation of the gas analysis by weight from the analysis by volume. Assume that we have one cubic foot of gas at 62 deg. fahr. and 30 in. Hg. of the following composition:

OLUMETRIC ANALYSIS	SPECIFIC WEIGHTS	ANALYSIS BY WEIGHT PER CENT
$CO_4 = a$ per cent	$0.1161 = W_a$	$A = \dfrac{W_a \times a}{W}$
$CO = b$	$0.07362 = W_b$	$B = \dfrac{W_b \times b}{W}$
$O_2 = c$	$0.08418 = W_c$	$C = \dfrac{W_c \times c}{W}$
$H_2 = d$	$0.00530 = W_d$	$D = \dfrac{W_d \times d}{W}$

Item **126. Calculation of the volume of the gas from the analysis of the gas and the analysis of the coal.** Evidently the total weight of the carbon appearing in the gas should be equal to the total weight of carbon in the coal minus the weight that is lost through the grate and the weight lost in soot and tar. This latter is small for the hard-coal producer and will be neglected.

Let P = per cent carbon by weight in dry coal.
W = total weight of dry coal.
W_1 = total weight of ash and refuse.
P_1 = per cent by weight of carbon in the ash and refuse.
W_2 = total weight of carbon that should appear in the gas, or the weight of carbon utilized in the producer.

$$W_2 = \frac{PW - P_1W_1}{100}$$

This carbon is contained in the CO_2, CO, CH_4, and C_2H_4.

The proportion by weight of C in CO_2 is 3/11, of C and CO is 3/7, of C in CH_4 is 3/4 and of C in C_2H_4 is 6/7.

Therefore the total weight of C contained in a unit weight of gas will be

$$W_3 = \frac{3/11\,A + 3/4\,E + 3/7\,F + 6/7\,G}{100}$$

Where A, E, F, and G are the per cent by weight of CO_2, CH_4, CO, and C_2H_4 from the gas analysis.

The per cent of this carbon contained in the gas as CO_2 is $\dfrac{3/11\,A}{W_3}$

The actual weight of this carbon will be $\dfrac{3/11\,A}{W_3 \times 100} \times W_2$. Since W_2 is the total weight of carbon utilized from the fuel.

One pound of carbon on burning produces $3\frac{2}{3}$ lb. of CO_2.

$$W_2 \times \frac{3/11\,A}{W_3 \times 100} \times 3\tfrac{2}{3} = \text{total weight of } CO_2 \text{ in the gas.}$$

Let W_S = the specific weight of CO_2 at 62 deg. and 30 in. Hg. See Item 121. The standard volume V_S of CO_2 will therefore be,

$$\frac{AW_2}{100 \times W_3 \times W_S} = V_S$$

Let this volume equal a per cent (from the volumetric gas analysis) of the total volume of gas delivered by the producer. The total volume of standard gas from the gas analysis is therefore

$$\frac{100\,V_S}{a} = V'_S$$

$$V'_S = \frac{A \times W_2}{a \times W_3 \times W_S}$$

Let a, b, c, d, e, and f, represent the mean C_p for the above gases between 32 deg. and t deg. fahr. Then the C_p of the producer gas = the sum of the products of the constituents of the gas by weight $\times$ the specific heat of the constituent.

That is,

$$\text{Item } 122 = [a \times \text{Item } 144 + c \times \text{Item } 147 + d \times \text{Item } 145 + e \times \text{Item } 152 + f \times \text{Item } 148 + g \times \text{Item } 146] \frac{1}{100}$$

Item 123

1	2	3	4
$CO_2 = O_2 + C$	$CO = C + O$	$CH_4 = C + 2H_2$	$C_2H_4 = 2C + 2H_2$
$44 = 32 + 12$	$28 = 12 + 16$	$16 = 12 + 4$	$28 = 24 + 4$

The total weight of carbon appearing in a unit weight of gas from the above =

per cent by weight $CO_2 \times \frac{3}{1100}$ + per cent by weight $CO \times \frac{3}{700}$ + per cent by weight $CH_4 \times \frac{3}{400}$ + per cent by weight $C_2H_4 \times \frac{6}{700}$

The total weight of H_2 appearing in a unit weight of gas = per cent by weight $\frac{H_2}{100}$ + per cent by weight $CH_4 \times \frac{1}{400}$ + per cent by weight $C_2H_4 \times \frac{1}{700}$

or Item 123 = [Item 144 $\times$ 0.273 + Item 145 $\times$ 0.429 + Item 148 $\times$ 0.75 + Item 149 $\times$ 0.858] + [Item 147 + Item 148 $\times$ 0.25 + Item 149 $\times$ 0.143]

Item 124. Observed.

Item 125. Let G = total volume of gas as measured by the meters.
p = absolute pressure of this gas in inches Hg. as observed.
T = absolute temperature in deg. fahr.
t = observed temperature.

The volume of gas G as measured by the meter is saturated with water vapor at the temperature t.

Let p_1 = pressure of this vapor in inches as obtained from the steam table.

Then as the pressure p is the total pressure of the mixture, the actual or partial pressure of the dry gas is $p - p_1 = p_2$.

Let p_s, G_s, and T_s. be the condition of standard gas. Then

$$\frac{G_s \times p_s}{T_s} = \frac{G \times p_2}{T} \text{ or } G_s = \frac{G \times p_2 \times T_s}{T \times p_s} = \frac{G \times p_2 \times 522}{T \times 30} = \frac{G \times p_2}{T} \times 17.4$$

Therefore Item 125 equals

$$\frac{\text{Item } 124 \times \text{Item } 26 \times 17.4}{\text{Item } 45}$$

Item 126. **Calculation of the volume of the gas from the analysis of the gas and the analysis of the coal.** Evidently the total weight of the carbon appearing in the gas should be equal to the total weight of carbon in the coal minus the weight that is lost through the grate and the weight lost in soot and tar. This latter is small for the hard-coal producer and will be neglected.

Let P = per cent carbon by weight in dry coal.
W = total weight of dry coal.
W_1 = total weight of ash and refuse.
P_1 = per cent by weight of carbon in the ash and refuse.
W_2 = total weight of carbon that should appear in the gas, or the weight of carbon utilized in the producer.

$$W_2 = \frac{PW - P_1W_1}{100}$$

This carbon is contained in the CO_2, CO, CH_4, and C_2H_4.

The proportion by weight of C in CO_2 is 3/11, of C and CO is 3/7, of C in CH_4 is 3/4 and of C in C_2H_4 is 6/7.

Therefore the total weight of C contained in a unit weight of gas will be

$$W_3 = \frac{3/11\,A + 3/4\,E + 3/7\,F + 6/7\,G}{100}$$

Where A, E, F, and G are the per cent by weight of CO_2, CH_4, CO, and C_2H_4 from the gas analysis.

The per cent of this carbon contained in the gas as CO_2 is $\dfrac{3/11\,A}{W_3}$

The actual weight of this carbon will be $\dfrac{3/11\,A}{W_3 \times 100} \times W_2$. Since W_2 is the total weight of carbon utilized from the fuel.

One pound of carbon on burning produces $3\frac{2}{3}$ lb. of CO_2.

$$W_2 \times \frac{3/11\,A}{W_3 \times 100} \times 3\tfrac{2}{3} = \text{total weight of } CO_2 \text{ in the gas.}$$

Let W_S = the specific weight of CO_2 at 62 deg. and 30 in. Hg. See Item 121. The standard volume V_S of CO_2 will therefore be,

$$\frac{AW_2}{100 \times W_3 \times W_S} = V_S$$

Let this volume equal a per cent (from the volumetric gas analysis) of the total volume of gas delivered by the producer. The total volume of standard gas from the gas analysis is therefore

$$\frac{100\,V_S}{a} = V'_S$$

$$V'_S = \frac{A \times W_2}{a \times W_3 \times W_S}$$

Item 126 therefore equals

$$\frac{\text{Item 144} \times (\text{Item 51} \times \text{Item 61} - \text{Item 52} \times \text{Item 68})}{0.116 \times \text{Item 135} \times (0.273 \text{ Item 144} + 0.75 \text{ Item 148} + 0.429 \text{ Item 145} + 0.858 \text{ Item 149})}$$

Item 127. Item 126 should be used as a check on Item 125. The difference between the two values should not exceed 5 per cent. Item 125 should be used in all computations.

Item 128. $= \dfrac{\text{Item 127}}{\text{hours}}$

Item 129. $= \dfrac{\text{Item 127}}{\text{Item 51}}$

Item 130. $= \dfrac{\text{Item 127}}{\text{Item 54}}$

Item 131. = Item 127 × Item 121

Item 132. $= \dfrac{\text{Item 131}}{\text{hours}}$

Item 133. $= \dfrac{\text{Item 131}}{\text{Item 51}}$

Item 134. $= \dfrac{\text{Item 131}}{\text{Item 54}}$

Items 135 to 143. From chemist.

Item 104. The relative humidity, or per cent saturation is observed by means of a hair hygrometer. This may also be obtained from a wet and dry bulb thermometer, and a set of psychrometric tables.

Item 105. See Kent, page 484, for weights of air and moisture.

Let p = per cent saturation, or relative humidity, Item 104.

n = weight of moisture contained in one cu. ft. of saturated air at the observed temperature, Item 29.

$\dfrac{pn}{100}$ = weight of moisture in 1 cu. ft. of air as used.

If m = weight of 1 cu. ft. dry air at the observed temperature, then

$$\text{Item 105} = \frac{pn}{100m} \times 100 = \frac{pn}{m} = \text{Item 104} \times \frac{n}{m}$$

This formula is in error due to neglecting the vapor pressure of water; this is however, negligible in the present case.

Item 82. Observed.

Item 83. Observed.

Item 84. = Item 82 − Item 83.

Item 86. The weight of water decomposed in the producer is evidently 9 times the weight of hydrogen formed, since 1 lb. of water on decompostion, yields 1 lb. of hydrogen and 8 lb. oxygen. The total weight of hydronge formed is equal to the totaweight of free hydrogen appearing in the gas, plus the total weight of hydrogen appearing in the CH_4 in the gas minus the total weight of hydrogen that is not in combination with oxygen in the coal.

Therefore, Item 86. =

$$9\left(\frac{\text{Item 131 (Item 147 + 0.25 Item 148) − Item 51 (Item 62 − }\frac{1}{8}\text{ Item 63)}}{100}\right)$$

Item 87. Owing to the difficulty in obtaining the weight of moisture in the gases leaving the producer with a proper degree of accuracy by the use of a dryer, it will ordinarily be better to use Item 86 for this item.

Item 106. Obtained from the gas analysis by weight, Items 144 to 152 inclusive.

Let A = per cent CO_2
B = per cent O_2.
C = N_2

Let D = per cent H_2
E = per cent CH_4
F = per cent CO

(1)	(2)	(3)
$C + O_2 = CO_2$	$C + O = CO$	$H_2 + O = H_2O$
$12 + 32 = 44$	$12 + 16 = 28$	$2 + 16 = 18$
$\frac{3}{11} + \frac{8}{11} = 1$	$\frac{3}{7} + \frac{4}{7} = 1$	$\frac{1}{9} + \frac{8}{9} = 1$

Fom equation (1), one lb. of CO_2 requires 8/11 lb. of O for its formation
From (2) one lb. CO requires 4/7 lb. of O for its formation.
The total amount of O appearing in 1 lb. of the gas is therefore

$$\left(\frac{8}{11}A + \frac{4}{7}F + B\right) \times \frac{1}{100}$$

This O comes from that contained in the air, that contained in the coal, and from the water decomposed. The oxygen contained in the coal, however, is supposed to be united with hydrogen, and is therefore contained in moisture which has been allowed for in the water decomposed.

Let W = total weight of gas.

Then the total weight of O used is

$$\frac{W}{100}\left(\frac{8}{11}A + \frac{4}{7}F + B\right)$$

Let W_2 = weight of water decomposed. From (3), 1 lb. of water decomposed liberates 8/9 lb. of O.

Weight of O supplied by decomposition of water = 8/9 W_2

Let W_3 = total weight of O supplied by the air.

From the above equation we have,

$$\left(\frac{8}{11}A + \frac{4}{7}F + B\right)\frac{W}{100} = \frac{8}{9}W_2 + W_3$$

$$\text{or } W_3 = \frac{W}{100}\left(\frac{8}{11}A + \frac{4}{7}F + B\right) - \frac{8}{9}W_2 \ldots\ldots\ldots\ldots (4)$$

The weight of air used is therefore $\frac{W}{0.23}$, since the proportion by weight of O in air is 23, or

$$\frac{W_5}{0.23} = W_4 = \frac{1}{0.23}\left[\frac{W}{100}\left(\frac{8}{11}A + \frac{4}{7}F + B\right) - \frac{8}{9}W_2\right] \ldots\ldots (5)$$

$$\text{Therefore Item 106} = \left[\frac{\text{Item 131}}{100}\left(\frac{8}{11}\text{ Item 144} + \frac{4}{7}\text{ Item 145} + \text{Item 146}\right) - \frac{8}{9} \times \text{Item 87}\right]\frac{1}{0.23} \ldots\ldots (6)$$

The above computation may be made from the weight of nitrogen appearing in the gas. The nitrogen comes from the air used and from the nitrogen introduced with the fuel.

Let C per cent per lb. = weight of N_2 from analysis

Let W as before = total weight of gas

Then $\frac{C\,W}{100}$ = total weight of N_2 in the gas.

The weight of N_2 supplied by fuel will be $\frac{W_1\,H_1}{100}$, where W_1 equals the total weight of dry coal and H_1 is the per cent by weight of N_2 contained in the coal. We have therefore,

$$\frac{CW}{100} = \frac{W_1H_1}{100} + W_4$$

where W_4 = total weight of N_2 in the air.

The weight of air supplied is therefore

$$\frac{W_4}{0.77} = \left(\frac{CW}{100} - \frac{W_1H_1}{100}\right)\frac{1}{0.77} = W_5 = \left(\frac{CW - W_1H_1}{77}\right)$$

$$\text{Or Item 106} = (\text{Item 131} \times \text{Item 152} - \text{Item 51} \times \text{Item 64})\frac{1}{77} \ldots\ldots (7)$$

The weight of air derived by formula (6) will be liable to error, due principally to the error in the determination of the total quantity of water decomposed, which may be large, and also to the neglecting of the SO_2 formed.

The weight determined by formula (7) will be in error, due principally to the taking of the weight of N_2 from the analysis by difference.

The results obtained from formulae (6) and (7) should check within 5 per cent.

The results obtained by (7) are believed to be more accurate and will be used in all computations.

Item **107.** This may be obtained direct from the calibration curve of the orifice. It should be compared with the two values obtained above.

Item 108. This will ordinarily be taken from Item 106

Item 109. $= \frac{\text{Item 108}}{\text{Hours}}$

Item 110. $= \frac{\text{Item 108}}{\text{Item 51}}$

Item 111. $= \frac{\text{Item 108}}{\text{Item 54}}$

Item 112. $= \frac{\text{Item 108}}{\text{Item 131}}$

Item 85. = Item 84 + Item 85*b* + Item 85*c*

Item 85*b* $= \frac{\text{Item 108} \times \text{Item 105}}{100}$

Item 85*c*. $= \frac{\text{Item 49} \times \text{Item 50}}{100}$

Item 88. = Item 85 − Item 87

Item 89. $= \frac{\text{Item 87}}{\text{Item 85}}$

Item 90. $= \frac{\text{Item 87}}{\text{Item 131}}$

Item 91. $= \frac{\text{Item 87}}{\text{Item 51}}$

Item 92. $= \frac{\text{Item 87}}{\text{Item 54}}$

Item 93. $= \frac{\text{Item 87}}{\text{Item 108}}$

Item 94. $= \frac{\text{Item 85}}{\text{Item 51}}$

Item 95. $= \frac{\text{Item 85}}{\text{Item 54}}$

Item 96. $= \frac{\text{Item 85}}{\text{Item 108}}$

Item 97. = Observed

Item 98. = Observed

Item 99. $= \frac{\text{Item 84}}{\text{Hours}}$

Item 100. $= \frac{\text{Item 99}}{\text{Item 11}}$

Item 101. $= \frac{\text{Item 87}}{\text{Hours}}$

Item 102. $= \frac{\text{Item 85}}{\text{Hours}}$

Item 103. $= \frac{\text{Item 97}}{\text{Hours}}$

Item 115. = Item 114 × Item 121

Item 116. $= \dfrac{\text{Item } 98 \times 0.2205}{\text{Item } 115}$

Item 117. $= \dfrac{100 \text{ Item } 88}{\text{Item } 131}$

Item 153. The grate efficiency is 100 times the ratio of the total B.t.u. in the fuel minus the B.t.u. in the fuel lost through the grate; to the total B.t.u. contained in the fuel. Therefore

$$\text{Item } 153. = \frac{\text{Item } 51 \times \text{Item } 78 \times 100 - \text{Item } 52 \times \text{Item } 68 \times 14{,}540}{\text{Item } 51 \times \text{Item } 78}$$

Item 154. The hot gas efficiency is 100 times the ratio of the total heat of combustion of the gas, plus the sensible heat of the dry gas, plus the total heat contained in the moisture, minus the heat given to the producer by the entering air, by the coal as sensible heat and by the moisture or steam in the air, or supplied from any outside source; to the heat of combustion of the dry coal. Therefore, Item 154 $= \{\text{Item } 119 \times \text{Item } 127 + \text{Item } 122 \times \text{Item } 131 (\text{Item } 39 - 62 \text{ deg.}) + \text{Item } 88 [1116 + 0.6 (\text{Item } 39 - 212)] - \text{Heat from external source}\} \times 100 \div \text{Item } 51 \times \text{Item } 78.$

The heat given to the producer by the air, moisture, coal, etc., may be neglected if the room temperature is within 20 deg. of the standard temperature 62 deg. This will ordinarily be the case. If steam is supplied to the producer by a steam nozzle taking steam from some outside source, the heat in this steam must be subtracted from the numerator of the above formula.

Item 155. The cold gas efficiency is 100 times the ratio between the total heat of combustion of the gases and the total heat of combustion of the dry coal. That is,

$$\text{Item } 155 = \frac{\text{Item } 119 \times \text{Item } 127}{\text{Item } 51 \times \text{Item } 78} 100$$

Item 157. $= \dfrac{\text{Item } 156 \times \text{Item } 49}{0.02 \times \text{Item } 127}$

Item 158. $= \dfrac{\text{Item } 97 \times 1000}{62.5 \times \text{Item } 127} = \dfrac{\text{Item } 97}{0.0625 \times \text{Item } 127}$

HEAT BALANCE

DEBIT

Item a. Obtained from Item 78.

Items b, c, d, e, f. Using as a standard the temperature of 62 deg. fahr., the heat given to the producer by the items 2 to 6 inclusive is in most cases negligible. The error at a temperature of 100 deg. fahr. is less than 1 per cent for a producer of the contained vaporizer type. However, the formulae will be given for computation of these items.

Item b. = Item 110 × 0.24 (Item 30 − 62°F)

Item c $= \dfrac{\text{Item 85b} \times (H - 1070)}{\text{Item 51}}$ where H = the total heat in 1 lb. saturated steam at the temperature of the fire room.

Item d $= \dfrac{\text{Item 49} \times \text{Item 50}}{100 \times \text{Item 51}}$ (Item 30 − 62 deg. fahr.)

Item e $= 0.24 \times$ (Item 30 − 62 deg. fahr.)

Item f $= \dfrac{\text{Item 82 (Item 33} - \text{62 deg. fahr.)}}{\text{Item 51}}$

CREDIT.

Item a = Item 122 × Item 133 × (Item 39 − 62 deg. fahr.)

Item b $= \dfrac{\text{Item 117} \times \text{Item 133}}{100}$ [(Item 39 − 212 deg. fahr.) × 0.6 + 1116]

Item c = Item 119 × Item 129

Item d $= \dfrac{\text{Item 52} \times \text{Item 68}}{\text{Item 51}} \times 145.40$

Item e This is very small and may be neglected.

Item f $= \dfrac{\text{Item 83}}{\text{Item 51}}$ (Item 34 − 62 deg. fahr.)

Item g = Sum of Items on debit side − (Item *a* + Item *b* + Item *c* + Item *d* + Items *e* and *f*.)

DISCUSSION

Prof. R. H. Fernald. In connection with the Government investigations, the feeling has prevailed ever since the beginning of the work in 1904, that gas producers could be tested on practically the same basis as steam boilers, i.e., without necessarily operating an engine in connection with the test. This would mean discharging the gas into the air in a manner similar to the discharge of steam in boiler test practice. This method of procedure has not been adopted at the Government testing station because so much prejudice has existed against the gas producer and gas engine. It has therefore been necessary that the gas generated at the testing station be utilized in an engine in order to avoid any discussion relating to the uncertainty of such operation. This has been particularly necessary owing to the large variety of fuels that have been handled and the variation in the quality of gas produced. It is true, however, that from the producer standpoint alone the engine is not essential, and the method suggested by Mr. Garland is ingenious and reasonably convenient.

2 There are a few points in connection with this paper upon which further information is desirable. In Par. 5 it is stated that the weight of steam was measured by passing the jet through a calibrated orifice in a thin plate. Methods of determining the quantity of steam used by gas producers seem to be varied and the results obtained somewhat uncertain. I believe it would be interesting to know the details of the method employed by Mr. Garland. In the testing station at St. Louis the steam used by the pressure producer was determined by means of a calibrated orifice, but the fluctuations in pressure were such that the readings obtained were not regarded as absolutely reliable. During tests covering a period of approximately two years the steam used varied from 0.28 lb. per lb. of coal fired to 1.13 lb. of steam per lb. of coal fired. The average for twenty consecutive tests showed 0.69 lb. of steam per lb. of coal fired. It should be borne in mind that the fuels used for the different tests were quite different in composition and that the amount of steam required by the different fuels may have varied considerably; but in spite of this fact the feeling which prevailed about the plant was that the method of determining steam by means of calibrated orifices was not entirely satisfactory unless the pressure of the steam passing into the producer, and the percentage of moisture in the steam, could be kept constant during the test.

3 At the Norfolk station, however, the steam required by the producer was supplied by an auxiliary boiler, so that all water passing into this boiler could be positively measured. Although the coals used for the six tests reported below were practically the same in composition, yet the records show the steam consumption per pound of coal fired to be decidedly variable, as follows:

(1) 1.12 lb. per lb. of coal fired	(4) 0.82 lb. per lb. of coal fired
(2) 1.14 " " " " " "	(5) 0.77 " " " " " "
(3) 1.04 " " " " " "	(6) 0.69 " " " " " "

This wide variation shown for these six tests is due entirely to the methods of operation, and not to uncertainties in measurement, as might at first be inferred. There is need of systematic and careful investigations relating to this question of steam per pound of fuel. At the Pittsburg station the method of determining the amount of steam used in the vaporizer is by means of a water tank calibrated in pounds, thus insuring accurate measurement.

4 In Par. 6 is presented the general method of determining the amount of fuel used. One phrase attracts especial attention: "at the end of the test the fuel bed being brought to as near the starting condition as possible." In boiler practice where the quantity of fuel on the grate at any one time is relatively small, it is undoubtedly possible, within a reasonable percentage of error, to determine the condition of the fuel bed and to make this condition practically the same at the beginning and close of an eight or ten-hour test.

5 However, the situation is totally different in gas-producer practice in which the initial fuel supply and the amount of fuel on the grate at any given time is large compared with the amount required by the plant during a run of a few hours only. Even though the conditions at the close of a producer test be made to duplicate those at the beginning, there is still considerable difficulty in determining the exact fuel consumption, owing to the lack of accuracy in determining the true thickness of the fuel bed. In a producer of 250 h.p. rating it is not uncommon to make an error of from four to six inches in the true depth of the fuel bed. In a producer of this size, this will cause an error of about 800 lb. of coal, or about 400 lb. of coke, according to the condition of the fuel bed at the time. It is imperative, therefore, that the tests of producer plants be continued to such length that these errors in measurement will be but a small percentage of the total fuel consumed.

6 Mr. Garland states that an endeavor was made to make the tests of such duration as to bring the probable error of filling down to about two or three per cent. It will be of interest to have explained in further detail the method of procedure used in determining the exact amount of fuel consumed. With a 250-h.p. plant in which the fuel consumption for a period of 8 hr. amounts to only 1800 lb. approximately, the error due to inaccurate measurement of the depth of bed and variations in fuel bed thickness may be as great as 1150 lb. The percentage of possible error in calculating fuel consumption for short periods is obviously great. With a period of 24 hr. and a fuel consumption of about 5400 lb., the percentage of possible error, although much less, is still over 20 per cent.

7 In the producer tested the effective fuel bed volume was approximately 4 cu. ft., which is equivalent roughly to 250 lb. of anthracite pea coal. It is probable that a large percentage of the gas value of this coal may be given off without materially decreasing the fuel volume, under certain conditions of fuel bed. In a run in which the fuel consumption for this producer amounts to only 800 lb. with an

initial bed of 250 lb., it is a question whether the percentage of error in fuel bed estimates may not amount to 10 or 12 per cent instead of 2 or 3 per cent.

8 In a recent paper on this subject published by the United States Geological Survey, the following conclusions were presented:

a Throughout a test the fuel bed should be maintained in uniform condition, with regard both to character of the fire and thickness of the bed.

b Failing in this, special care should be exercised to see that the fuel bed is in the same condition and of the same thickness at the close of the complete test or at the end of a test period, as at the beginning.

c A test should never be started when the producer has been standing idle for some time with banked fires, as the fuel bed will not be in the average condition under which it will be required to work during the test.

d If, as the appointed hour for closing the test approaches, the fuel bed is not in the proper condition, the time of ending the test should be postponed until the bed naturally assumes the proper thickness and character. No forcing of conditions should be allowed simply to bring the test to an end at a previously determined hour.

9 In Par. 12 it is suggested that the volume of gas may be computed from the analyses of the gas and coal and the statement is made that this "may be relied upon within 5 per cent, provided the sampling is accurate." This last clause "providing the sampling is accurate" seems to contain the essential point. This is an important subject and too much emphasis cannot be placed upon the fact that proper sampling is difficult to accomplish.

10 Reference to the packing of the ash in the fuel bed suggests another point which must be very carefully considered in making fuel bed measurements, viz., the swelling of any coals due to the application of heat. Frequently in the government tests, the measurements of the fuel bed have caused very misleading impressions due to the fact that the fuel had swollen materially during the operation of the plant.

11 In Form 1 a number of items appear under a heading Quantity of Air. Although it is quite possible to determine small quantities of air with some degree of reliability, yet methods for making such measurements of large quantities appear to be entirely lacking.

Further details of the methods pursued in this test will, I believe, prove of interest.

12 In items 128 and 128*b*, are presented the producer efficiencies based on dry coal and combustible. It is not apparent why there should be a difference of 4 per cent in the efficiencies shown.

G. M. S. Tait. The usual method of testing a plant for such a short period would be to operate the producer for two or three days beforehand so as to bring the fuel bed to an average working condition, that is, with an average amount of carbon in proportion to ash. Then a comparatively short run, provided great care was taken as to the fuel depth, would give fairly reliable figures. Otherwise, when drawing on a fresh fire and making a run of only twelve hours, it would be necessary to pull the entire fuel bed at the end of the run and analyze the contents for carbon and ash, in order that any sort of accuracy might be obtained.

2 In one of the author's tests, instead of 34 lb. of coal per sq. ft. of grate area, 8 to 10 lb. would be a normal figure, as 34 lb. of coal per sq. ft. is entirely impracticable for anything but a very short run on American fuels. In this test a large part of the coal originally in the producer was apparently burned to ash, and its consumption was completely left out of the test, causing very erroneous results.

3 Attention is called to the fact that the ash content in the ashpit is practically much less than the ash content of the fuel, as shown by analysis. The balance of the ash is undoubtedly in the fuel bed and its presence there entirely upsets the basis of calculation for this paper. It is safe to say that a two days' run would have given a reversal of the first day's figures.

H. H. Suplee. I would like to speak of the unreliability of an orifice as a means of measuring. My attention has recently been called by a member to the discharge of steam from a boiler in which the orifice and all conditions surrounding it were identical in several tests. The amount of steam generated was measured by carefully weighing the water, double-checking it in tanks, and yet there was a variation of ten to fifteen per cent in the results, the steam pressure and the temperature being kept as uniform as possible. This fact casts a doubt on the orifice as a means of determining flow.

L. B. Lent. The figures given show that the draft through the producer was practically 1½ in., and yet 38.8 lb. of dry coal was burned per sq. ft. of grate area. Still, with this consumption the producer efficiency seems to be very good. My impression is that this is a remarkable rate of consumption in producers of large type; and I would like to know if this is the common practice in smaller sizes of producers.

H. F. SMITH. Regarding the conditions of the fuel already discussed it seems to me that the author has presented all the necessary evidence to show that the conditions in the fuel bed were not the same at the end as at the beginning of the test.

2 In the graphical log in Fig. 3, it will be noticed that the temperature of the gas leaving the producer at the beginning of the run was 400 deg. fahr., and at the end of the run something over 1300 deg. fahr. The rates of gas production and fuel consumption were practically uniform. It is evident that there was some variation in conditions, otherwise this difference in temperature would not have occurred.

W. B. CHAPMAN. Perhaps I can answer Mr. Lent's question in regard to the quantity of coal gasified in producers. Producers for furnace work are usually rated at 10 lb. per sq. ft. of internal diameter on Pennsylvania coal, but only 7 lb. per sq. ft. on Illinois coal. The best record I have seen for hand operated bituminous coal producers was 16 lb. per sq. ft. Mechanically agitated producers gasify from 15 lb. to 30 lb. per sq. ft.

2 The question of the amount of anthracite coal that can be gasified is very interesting. Engineers from abroad say that two or three times as much can be gasified as is the custom in this country. Every gas producer manufacturer in this country having a foreign engineer in charge has designed his first producer very much too small. The more experienced manufacturers do not rate their producers at more than 10 lb. per sq. ft.

3 It is strange that we cannot get the results said to be obtained in foreign countries. The difference must be in the coal.

PROF. R. H. FERNALD. In reference to the rate of burning per square foot of grate area, I desire to call attention to the high figures shown by Mr. Garland. These figures seem to be very unusual for this type of producer even under the test conditions described. The highest rate with which I am familiar in commercial operation is that found in the case of a large installation using lignite as fuel. This plant shows a daily rate of 33 lb. per sq. ft. of fuel bed area per hour during 16 hours each day and 48 lb. during the remaining 8 hours. In this installation the producers are of the down-draft type, but even under these conditions this rate is, I believe, exceptional.

2 In reference to Mr. Chapman's remarks about the manufacturers abroad, I would say that apparently all of them stipulate the type of coal that shall be used in their producer. They specify that the coal must be of such and such a grade, non-caking and with only such and such a percentage of ash and tar. As nearly as I was able to ascertain, practically every manufacturer abroad has reached the conclusion that it is wise to designate definitely the coal to be used.

3 In one suction producer in Germany, operating on bituminous coal, I found that the successful manipulation of the plant was due to the fact that three kinds of coal, mixed in the proper proportions, were being used. In other words, this type of producer using bituminous coal as fuel was entirely feasible in the home plant of the manufacturer, but it would hardly prove a saleable article in this country, as it would be almost impossible to guarantee the three required grades of coal at all times. It would also be out of the question to secure operators at a reasonable compensation who would give the plant the required attention.

E. N. Trump. The rate of combustion in producers using anthracite coal depends very much upon the size of the coal. Seven tons per 24 hours, with a producer 7 ft. in diameter, is about the maximum for No. 1 buckwheat coal. This equals 15 lb. per sq. ft. of grate surface per hour.

2 Burning Western coals in producers, especially Hocking Valley coal, a high rate of combustion is obtained. I have operated producers continuously for a considerable period at the rate of 42 lb. per sq. ft. of grate surface. This is with a large percentage of steam in the air, also with mechanical ash extraction, the fuel bed being thus kept well agitated.

2 Venturi meters give very accurate results in the measurement of both gas and steam, much more accurate than the simple orifice.

The Authors. It will be well to emphasize thefact that the producer under discussion was designed and intended for intermittent service only; that is, it is not suitable for runs of greater than 12 to 18 hours duration. This is due to the small size of the producer, and the absence of charging bell, water-sealed ashpit and mechanical means for agitating the fuel bed.

2 Owing to the small size of the producer and the absence of means for thoroughly cleaning the fuel bed from time to time, as above noted, the accumulation of ash toward the end of 12 or 15 hours of continuous operation is so great as to necessitate such thorough cleaning as seriously to lower the heating value of the gas.

3 From the foregoing it will be evident that our test corresponds to the conditions under which the producer is normally operated. Owing to the thorough cleaning of the fires before starting the test, and the removal of the ash from the grate, a large quantity of green fuel is brought into the path of the outgoing gases, resulting in their being cooled. At this time, the temperature of the fuel bed is also lower, as indicated by the analysis of the gases over the first two hours of the test. The heating value of the gas is not lowered, for two reasons: first, the descent of the green fuel into the path of the gases results in the distillation of the CH_4 and other heavy hydrocarbons; secondly, an increase in the percentage of hydrogen results from the lower temperature of the fuel bed.

4 At the close of the test the fuel bed was evidently at a higher temperature than at the start. This resulted in increasing the unaccounted-for loss in the heat balance, but its extent (estimated from the results of a number of tests) is about one per cent for the present test. This, it is believed, explains the condition pointed out by Mr. Smith.

5 Professor Fernald and Mr. Tait call attention to the probable inaccuracy in determining the weight of coal fired on the test. We have recognized this source of error, and in Form 2 have included such items as give proof of the accuracy of the work through the stoichiometric relations. As the full import of these items has evidently not been realized, we will amplify them.

6 First, to determine approximately the purely mechanical error in estimating the weight of coal fired during the present tests, the producer was filled four separate times, and the weight of coal required was noted in each case. The average of the four weights was taken as the mean or true weight of coal required to fill the producer. The results are given in Table 1, herewith. It will be seen from this table that the maximum variation from the mean is 8.75 lb., or 1.7 per cent. This in the test under consideration represents an error of probably 1.1 per cent.

7 Mr. Tait seems to think that the presence of the ash in the fuel bed "upsets the basis of calculation for this paper." The total weight of ash in the dry coal is 776.5 lb.; 13.17 per cent = 102 lb., of which 52 lb. was taken out in the ash and refuse, leaving 50 lb. remaining in the fuel bed. This would seem to indicate an error of 6.3 per cent, due to failure to remove this ash. Since the ash is soft and fine it would pack into the interstices between the coals so that its volume would by no means displace the same volume of coal. If it displaced one-half its volume of coal it would cause an error of slightly over 3 per cent. It is probable that its presence caused even less error than this. In order to bring out the different errors we will analyze the conditions existing on the test.

TABLE 1 WEIGHT OF GREEN COAL REQUIRED TO FILL THE PRODUCER

Trial Number	Weight Lbs.	Variation from Average Weight	Per Cent Variation
1	669.25	−8.75	1.70
2	676.25	−1.75	0.26
3	683.25	+5.25	0.77
4	683.25	+5.25	0.77
Total	2712.00		
Average	678.00		

8 It is probable that the composition of the producer gas on leaving the scrubber, and at any two points in the cross section of the main, is the same. In order to eliminate such an uncertainty, however, we have taken the gas for our samples simultaneously from different points in the cross section of the main and at a point beyond the scrubber, by means of the sampling tube illustrated in the paper. These samples were taken continuously over the period of the test, both for analysis and for the calorimeter. The heating value of the gas, as computed from analysis, is 138.1 B.t.u. After corrections were made for the error in the meter, the error due to the vapor pressure of water, and the error due to radiation and conduction into the calorimeter, the heating value of the gas as determined by the Junker calorimeter was 137.3. Since the heating value as determined from two separate samples of gas, by two independent methods, and by two independent observers, checks within 1 per cent, it must be admitted that the sampling, the analysis and the heating value of the gas are probably correct within less than 1 per cent.

9 The volume of gas generated by the producer was measured by

a Westinghouse meter, guaranteed by the company to be accurate within 2 per cent. However, as a further precaution, the meter was carefully recalibrated and was found to be accurate within this limit. A calibration curve was plotted from the calibration, so that the error in determining the gas volume must have been within 2 per cent, and was doubtless even closer than this.

10 As shown by a number of tests on the present fuel, the coal was fairly uniform. A sample representing about 15 per cent of the coal fired was mixed and quartered until about eight or ten quarts remained. This was then ground, and again mixed and quartered until sufficient to fill a quart jar remained. The heating value from this sample as determined by the calorimeter was 13,040 B.t.u. per lb. The mean of eight determinations on this same fuel showed a heating value of 12,900 B.t.u. The probable error in the analysis and in sampling the fuel, judging from the heating value, is doubtless not greater than 1 or 2 per cent.

11 We have noted the volume of gas computed from the analysis of the coal and the analysis of the gases in Form 2, Item 126. This volume is 56,200 cu. ft. of standard gas, while the volume as actually measured by the meter, corrected for the vapor pressure of water, is 57,500, showing a discrepancy of about 2.3 per cent. The volume determined from computation was obtained from the formula of Item 126, Form 3. It is based on the fact that the weight of carbon in the coal fed to the producer must equal the weight of carbon appearing in the producer gas, plus the carbon lost in the ash, plus the carbon lost in soot and tar, plus the carbon lost by the absorption of CO_2 and CO by the scrubber water. The carbon lost in the ash is readily obtained, the carbon lost in the soot and tar is not over 1 per cent, while the carbon lost through the absorption of the gases by the scrubber water is also very small.

12 It may be well to compute the carbon in the gas and compare this with the carbon fed to the producer in the coal. We will compute the latter first. The total carbon in the coal is $0.7984 \times 776.5 = 620$ lb. The total carbon in the ash is $0.388 \times 85 = 33$ lb. The carbon that should appear in the gas is therefore 587 lb. The total weight of gas from Item 131, Form 2, is 3912 lb.

$$
\begin{array}{lll}
\text{Carbon in } CO_2 \text{ of gas} & = 0.0716 \times 12/44 \times 3912 & = 76.4 \\
\text{“ \quad “ } CO \text{ “ \quad “} & = 0.2925 \times 12/28 \times 3912 & = 490.5 \\
\text{“ \quad “ } CH_4 \text{ “ \quad “} & = 0.0112 \times 3/4 \times 3912 & = 32.8 \\
\hline
 & & 599.7
\end{array}
$$

Thus 599.7 lb. is the total weight of carbon appearing in the gas as measured by the meter. $599.7 - 587 = 12.7$ lb. of carbon unaccounted for $= \frac{12.7 \times 100}{599.7} = 2.1$ per cent. As already stated, there may be an error of 1 per cent in the meter by which the above volume of gas was determined, the error being either positive or negative. There may have been 1 per cent of carbon lost in the soot and tar, but not more than this; there may also have been an error in the analysis of the coal amounting to $1\frac{1}{2}$ per cent. We estimate the principal errors in the test as follows:

	Per Cent
Error in filling the producer	-2.1
Gas analysis or heating value of gas	± 0.7
Volume by meter	± 1.5
Coal analysis and sampling of coal	± 1.5
Carbon lost in soot and tar	-1.0
Loss in sensible heat in the fuel bed due to the lower temperature at the start than at the close of the test	-1.0

The total error in the results of the test that would affect the cold-gas efficiency of the producer, if all the above errors are assumed as accumulative, equals 0.8. The probable error is 2.7.

13 There are three other errors that may affect the heat balance, namely, the error in measuring the temperature of the outgoing gases, the error in the determination of the specific heat of these gases, and the error in the amount of steam fed to the producer. The error in measuring the temperature of the gases may be 2 per cent; the error in determining the specific heat of the gases may be 6 per cent; the error in determining the steam fed to the producer may be 25 per cent. If these errors are accumulative, the first two represent a total error based on the heating value of the fuel of about 2 per cent, and the third of about 1 per cent. Therefore, if all errors are accumulative, the total error in the heat balance is about 6.8 per cent; as some of these errors will be positive and others negative, the probable error in the heat balance is about 3.5 per cent. As the heat balance shows an unaccounted-for loss of 4.4 per cent, about 1 per cent being radiation and conduction, the actual error in measuring the coal delivered to the producer on this test could not have exceeded 3 per cent. We have therefore been able to run tests of such duration as to reduce the probable error in filling the producer to 2 or 3 per cent. Furthermore we believe the results indicate that they are above the average in

accuracy for this kind of work, as we have seen very few tests on producers that would stand the above analysis.

14 As Professor Fernald and Mr. Suplee have pointed out, we have found the use of the thin plate orifice for the measurement of steam not altogether satisfactory. As the heat supplied in the steam on most of our tests is small, a large error is permissible in the measurement. Our aim has been to vary the pressure on the orifice so as to keep the hydrogen content of the gas practically constant. It might be well to state that the orifice was used only while we were obtaining a new vaporizer for the producer.

15 We have found no tendency in the anthracite coal to swell. We believe that this is a property of bituminous coal containing large quantities of moisture and of hydrocarbons.

16 As the quantity of air does not enter into the computation of the more important quantities, it was computed from the nitrogen in the producer gas. The formulae for this computation are given in Form 3.

17 The difference in the efficiency based on dry coal and the efficiency based on combustible, as noted by Professor Fernald, is due to the fact that we have used the word combustible as defined in Form 3, Item 54. The efficiency based on combustible corresponds to the efficiency based on 100 per cent grate efficiency. It is used for the reason that it is often desirable to show relations between efficiency and other quantities that are independent of the grate efficiency.

18 The amount of coal burned per square foot of grate area is a very variable quantity and depends upon the size of the fuel, the kind of fuel, the nature of the ash, the amount of water supplied, the proportions of the producer, the operation and the length of run.

19 For intermittent work, such as the present producer is adapted for, and with coals containing an ash infusible at temperatures under 2300 deg. fahr., it is possible to operate at several times the capacity possible with coal containing a fusible ash which necessitates a low fuel bed temperature.

20 The rapidity and extent of the reaction of CO_2 on incandescent carbon depend upon the temperature and upon the catalytic action of the fuel. At a given temperature and an indefinite time of contact of gases with the incandescent carbon, a definite amount of CO_2 and CO will be formed. The lower the temperature the less the per cent of CO formed and the longer the time required for equilibrium, so that with low temperature in the fuel bed the time of contact of the

gases with the fuel must be greatly increased. The same is true for the reaction of water on incandescent carbon. Harries[1] passed water vapor over incandescent carbon at different temperatures and obtained the results given in Table 2. These results show the effect of temperature upon the water-gas reaction. Due to the low temperature, the CO_2 is high, the CO is low and the ratio of water decomposed to water supplied is small. The latter fact, in the case of the producer, results in lowering the efficiency, as the undecomposed water carries out a large quantity of heat.

21 The curves of Fig. 1, herewith, taken from Dr. Clements'[2] work on the rate of formation of CO in gas producers, illustrates the effect of the time of contact, expressed in terms of velocity in feet per second, upon the amount of CO formed in passing CO_2 over incandescent anthracite coal. At a temperature of 1100 deg. cent., and a time of contact corresponding to a velocity of 1 ft. per sec., 11 per cent of CO is formed. If the velocity is reduced to 0.1 ft. per sec., so that the time of contact is increased ten times, 70 per cent of CO is formed. If an indefinite time of contact is assumed, equilibrium is reached at

TABLE 2 EFFECT OF TEMPERATURE ON WATER-GAS REACTION

Temperature Deg. Cent.	H_2	CO	CO_2	H_2O
674	8.41	0.63	3.84	87.12
838	28.68	6.04	11.29	54.09
954	44.43	32.70	5.66	17.21
1125	50.73	48.34	0.6	0.303

this temperature with 90 per cent of CO formed. This illustrates why it is necessary to use a small rate of combustion per square foot of grate area, due to operating with coals requiring a low temperature for the prevention of clinker formation.

22 If in the example just cited the temperature had been 1300 deg. cent. in the fuel bed, 70 per cent of CO would have been formed at a velocity of 0.5 ft. per sec. The time of contact would have been reduced five times, so that the rate of combustion could have been increased almost five times without appreciably changing the composition of the gas or the depth of the fuel bed.

23 In the case of our tests with the Scranton pea coal, we have

[1] Habers, Thermo-dynamics of Technical Gas Reaction, p. 138.

[2] Bulletin No. 30, Engineering Experiment Station, University of Illinois.

been able to vary the coal per sq. ft. of grate area from about 10 lb. to 45 lb., without appreciably affecting the efficiency of the producer. At the higher rates of combustion, however, the producer requires much more attention. If it were not for the fusion of the ash, the weight of coal per square foot of grate area could be increased indefinitely by the use of a blast and a sufficiently deep fuel bed.

24 The term "coal per square foot of grate area," as used in producer practice, is not, we believe, a true basis of comparison for the operation of different producers, for the reason that the coal per

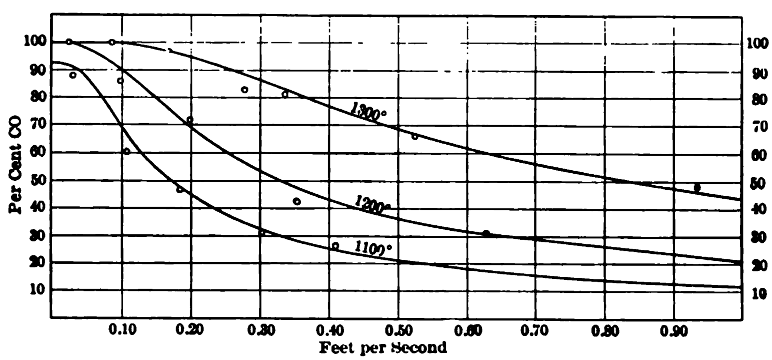

FIG. 1 VELOCITY OF GAS IN FEET PER SECOND, FUEL BED 1 FT. DEEP

square foot of grate area depends to a certain extent upon the depth of the fuel bed. For this reason, largely, we have used a term, "rate of descent of coal through the fuel bed," or "coal per cubic foot of fuel bed per hour," which appears under Items 70 and 71 in Form 1.

No. 1264

BITUMINOUS GAS PRODUCERS

WITH SPECIAL REFERENCE TO TESTS ON THE DOUBLE ZONE TYPE

By J. R. Bibbins, New York

Member of the Society

Several manufacturers have seriously applied themselves for years to perfecting the bituminous producer. The problem has been difficult and success elusive; but the improvements of the last two or three years have been material, and likely to lead to a type universally acceptable as standard. Outside of the question of pecuniary reward, much credit is due to these manufacturers for persevering against material obstacles and personal prejudice, and at an expense ruinous to any but those possessing large resources.

2 It is the object of this paper to record the results of the most recent achievements in this direction, and to interpret them in the light of personal experience. No attempt is made to discuss the commercial aspect, and in this respect the results presented will largely be left to speak for themselves. These results are drawn from resources accurate and reliable in so far as commercial tests can be made to approximate scientific investigation. Beyond this no claims can be made for refined accuracy.

ESSENTIAL REQUIREMENTS

3 Successful operation of a modern gas engine generating station prescribes certain requirements in the producer plant:

a Continuous operation, 365 days per year. Any departure from this condition means reserve equipment, additional capital outlay and idle plant. Producer designers cannot escape at this advanced stage of the art a condition parallel to that of steam boiler practice. For this continuous service the water-seal has proved adequate, but

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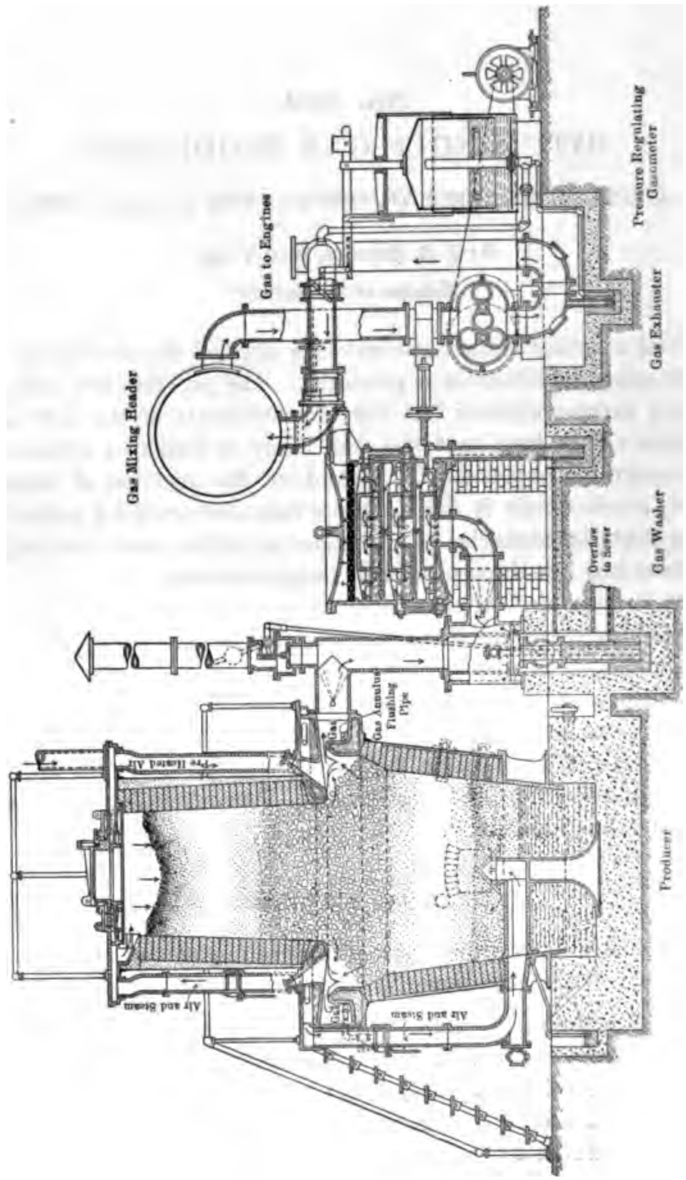


FIG. 1 SECTIONAL VIEW OF PRODUCER AND AUXILIARY APPARATUS

some means of mechanically removing ash should be developed.

b Plant suited to various kinds of fuels without remodeling, such as change of grates, etc. Fortunately gas producers are unusually flexible in this regard.

c Gas clean and free from tar. No engine design except perhaps some one of the simple valveless types can withstand the action of viscous tar deposits on the valve seats. Mechanical extraction can hardly be considered an acceptable remedy in this regard.

d Moderate labor requirements. No design will last which requires excessive attendance and large periods of shutdown for cleaning or repair.

e Prevention of clinker formations. Both labor cost and the uniformity of gas production are affected seriously by clinker. The obvious remedy is relatively low fuel bed temperatures.

f Automatic gas regulation. Large and expensive gas holders should be unnecessary. Quantity and quality regulation of gas may be made substantially automatic by proper design. An essential requisite is to relieve the producer attendant of all possible adjustments, as it is almost impossible to obtain at the prevailing wage the grade of intelligence otherwise necessary. The power-driven exhauster has removed a great proportion of the disabilities of the steam blown producer.

g Minimum auxiliary apparatus. It is manifestly inadvisable to nullify the high efficiency of the producer by wasteful auxiliaries. For this reason the suction principle has come into favor. Internal vaporizers provide automatic regulation quite adequate to the usual fluctuations in demand for gas, thus dispensing with the small boiler.

DESCRIPTION OF POWER PLANT, ETC.

4 The tests herein presented pertain principally to the double ne type of producer. As a complete description of this type was corporated in the last report of the National Electric Light Associa- on, 1909, constructional details may be dispensed with. Fig. 1

shows the arrangement in sections, comprising the following essential parts:

Water sealed ash pit,
Lower coke gasifying zone,
Central belt evaporator,
Upper coking zone for green fuel,
Air cooled top (preheating air blast),
Charging funnel open to atmosphere,
Vapor control valves for top and bottom fires,

FIG. 2 525-H.P. BITUMINOUS PRODUCER PLANT, WESTERN CHEMICAL COMPANY, DENVER

Radial poke holes for raking top and bottom walls,
Static cellular washer,
Positive rotary exhauster,
Automatic by-pass regulator valve,
Regulating gasometer for maintaining constant delivery pressure to engine.

5 This system obviously works entirely by suction with the charging top at atmospheric pressure. The sole adjustment is the relative position of the vapor control valves, which are set *permanently* for any given fuel and require no change for ordinary variations in power load. These valves determine the relative rates of combustion in

the upper and lower zones, the temperatures, and the rate of settling of the two fuel beds. While the producer is not supersensitive, intelligent adjustment is necessary to secure the most uniform gas. But the gas holder is dispensed with entirely, as the production is directly proportionate to the demand, giving a constant delivery pressure at the engine.

SCHEDULE OF TESTS

6 This producer plant has been under test in commercial sizes (175 h.p.), at East Pittsburgh, since December 1907, with various fuels and under various conditions of load. Up to July 1909, a total of over 2040 hours of operating tests had been run, operating from a minimum of 47 to a maximum of 514 hours continuous tests and on both 10- and 24-hour runs. Over 266,000 lb. of coal were gasified, the fuels ranging from low grade lignites to the best Pocahontas semi-bituminous coal. Some trials were also made on meadow peat. All the gas made was tested by means of a standard three-cylinder engine of 140 h.p., operating also continuously against the resistance of a prony brake. The gas was measured by wet meters at both[1] the producer and the engine. Determinations were made regularly for calorific value by means of the Junker calorimeter; for impurities by the Sargeant filter paper method; for composition and heat value by chemical analysis. Coal was weighed on scales—not measured. Table 1 shows a complete schedule of tests; of these special Tests F and G were run to determine accurately the normal standby loss; Test H to try out the type of washer shown in the sectional drawing, Fig. 1. It is apparent that this series of tests is unusually valuable in indicating results under various conditions of service. The important results follow, and are discussed *seriatim.*

DISCUSSION OF RESULTS

7 It should be noted by Table 2 that fuels containing as high as 25 per cent of water were successfully used for power purposes. The efficiency curve (Fig. 3) fully establishes the fact that the efficiency of heat conversion is practically as high with lignites as with the cheaper fuels.

8 In the test with Texas lignite an important fact was brought out, which has especially puzzled theorists for some time, viz., that with a poor fuel the rate of combustion can be increased sufficiently to permit the same rating of the producer as with better fuel. This re-

[1]For check purposes, meter calibrated by positive holder fall.

TABLE 1 SCHEDULE OF TESTS

Test	Date	Fuel	Duration Hours	Hr. per day	Load on Producer b.h.p.	Load on Producer Max.	Remarks
	1908						
A	4/ 2- 4	So. Am. Lignite	72	11	....	....	Purged Gas.
B	4/16-30	Col. Lignite	314	24	121.7	156.9	Continuous test.
C	5/ 8-23	Pittsburgh Coal	298	24	158.3	129	" "
D	7/16-31	" "	370	10	158.5	206	Intermittent test.
E	8/ 4-25	" "	514	10	170.8	190.7	" "
F	9/ 1-19	" "	432	Standby	.....	.....	Standby test. Fires blasted 1 hr. once in 24 hr. day
G	10/12-19	" "	168	"	.....	.....	
H	11/ 9-14	" "	...	10	137 to	204	Washer test and capacity test
	1909						
I	6/ 1- 2	Pocahontas Coal	46½	22½	75.6		Continuous test.
J	6/ 3- 4	" "	48	24	101.4		" "
K	6/ 5- 6	" "	48	24	126.5		" "
L	6/ 7- 9	" "	72	24	150		" "
M	6/30- 2	Texas Lignite	72	24	128	135	" "
N	7/ 7- 8	" "	42	24	157.2		" "

moves a heavy restraint on the development of producers for the enormous lignite fields of Texas, Wyoming, Colorado, Montana and the Pacific States. In Test N, Table 3, a charging rate of 27.2 lb. per sq. ft. per hr. was maintained with Texas Lignites and 15 lb. with Pocahontas coal, both at 150 h.p. load; with Pittsburgh run of mine it was slightly higher (18.1).

9 An economy of less than 1 lb. per brake horsepower-hour is probably below previous results with bituminous producers. This

TABLE 2 TYPICAL PROXIMATE ANALYSES OF FUELS TESTED

Class of Fuel	Moisture	Volatile	Fixed carbon	Ash	Sulphur	B.t.u. per lb. as fired	B.t.u. per lb. dry
Meadow Peat—Massachusetts..........	38.10	40.54	17.86	3.50	1.05	6410	10340
Lignite—South America...............	20.05	34.44	30.85	14.66		8035	10045
Lignite—Northern Colorado....	16.63	33.78	42.22	7.37		9589	11500
Lignite—Texas......................	24.08	38.55	28.76	8.61	0.57	7974	10503
Bituminous—Pittsburgh run of mine....	2.03	34.98	56.22	6.77	1.29	13305	13590
Semi-Bituminous—Pocahontas run of mine............................	1.39	16.01	74.28	8.32		13983	14170

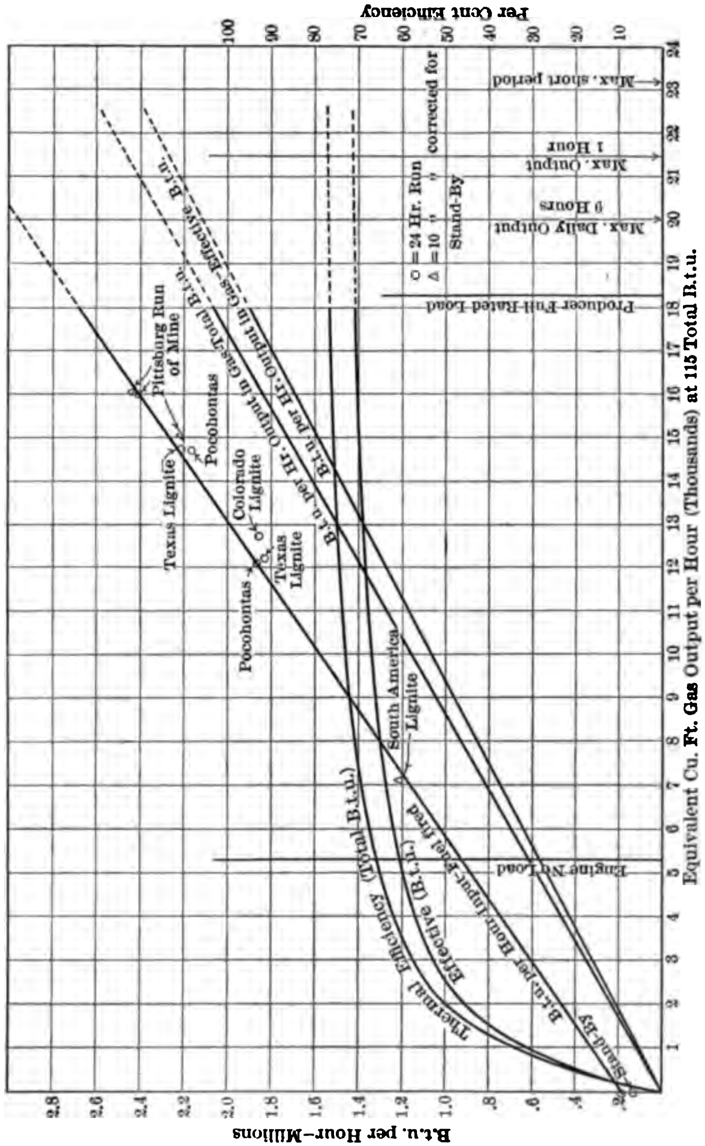


FIG. 3 EFFICIENCY TESTS, TYPE T-35 BITUMINOUS PRODUCER, WESTINGHOUSE MACHINE COMPANY

corresponds to less than 1½ lb. per kw-hr. in an electric generating station. An interesting point is the low standby fuel consumption, which averages in over a week's run 1 lb. per sq. ft. of fuel bed area per hour. In Test G it was reduced to this amount from 1.49 lb. (Test F) simply by reducing the natural up-draught through the idle producer, by closer adjustment of the valves.

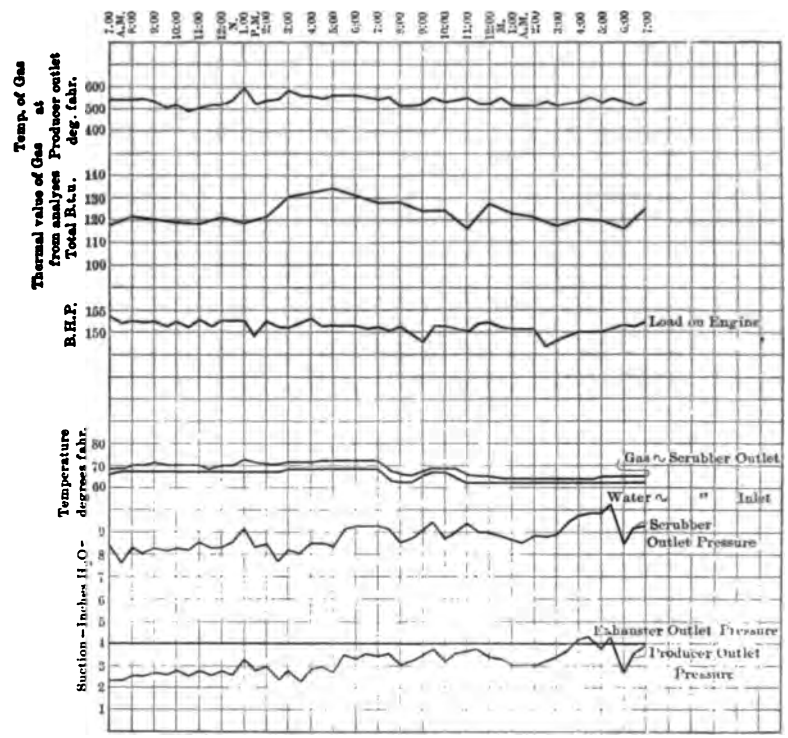


FIG. 4 TYPICAL LOG, POCAHONTAS COAL, LAST DAY OF RUN, 21½ HR.

10 Test H, which was a capacity test, shows an 18 per cent overload on gas production maintained for nine consecutive hours with Pittsburgh run-of-mine. Test C with the same coal shows nearly 30 per cent overload.

11 In heat value the gas is not high: but, which is more important, it is fairly uniform as shown by the typical log, Fig. 4, 5 and 6. The heat value seems to bear a certain relation to the fuel bed temperature. It is found that if a certain temperature of the gas off-

TABLE 3 FUEL DATA

Test	Lbs. charged incl. standby	Lbs. Charged Per Hr. Aver.	Lbs. Charged Per Hr. Max.	Lb. per[1] sq. ft. per hr.	Fuel Consumption Per B.H.P. Hr. Gross[2]	Fuel Consumption Per B.H.P. Hr. Net[3]	Fuel Consumption Per Kw. Hr.[4] Gross	Fuel Consumption Per Kw. Hr.[4] Net
A	5003	151.6	160.8	15.2	2.3	2.3	3.33	3.33
B	60740	194	228.5	19.4	1.59	....	2.31	
C	53925	181.1	207	18.1	1.14	....	1.66	
D	27157	169.2	222	16.9	1.39	1.19	2.01	1.73
E	35647	184.6	202	18.5	1.24	1.08	1.8	1.56
I	4849	104.2		10.4	1.37	....	2.0	....
J	6065	126.4		12.6	1.25	....	1.81	....
K	6403	133.4		13.3	1.06	....	1.54	....
L	10699	149.1		14.9	.983	....	1.42	....
M	16964	233.1		23.3	1.82	....	2.64	....
N	11503	272.1		27.2	1.80	....	2.61	....
F	8813	14.9			This standby rate applies to previous tests			
G	1680	10.0			Standby rate reduced by reducing updraft on fire			

[1] Based on area at level of green fuel.
[2] Including standby fuel.
[3] Standby fuel deducted.
[4] Based on 92.5 per cent generator efficiency.

TABLE 4 GAS DATA

Test	Cu. ft. uncorrected	Cu. ft. per hr. 60 deg. fahr. 30 in. mer.	Max. for 1 hr.	Cu. ft. per lb. fuel	Cu. ft per b.h.p. hr.	B.t.u.* per cu. ft.† total	B.t.u. effective
A	238,900	7245	8500	47.7	102	113.4	
B	3,873,000	12400	16700	63.8	101.4	118.2	
C	4,812,200	16210	23300	89.3	102.1	113.8	
D	2,074,500	15660	21500	88.2	101.9	110.7	
E	2,797,470†	16600	17853	90.2		111	
H		17815	21511	(9 hours)		111.4	
K						119	109.7
L						120.6	112.2
M	891,300	11933		51.3	93.1	118.7	107.8
N	569,300	13092	15500	48.05	86.75	129.7	118.75

* Total heat values. Water determination not made during some tests.
† Corrected to 30 in. mer. and 60 deg. fahr.

take is exceeded (about 1000 deg.), vitiation of the gas ensues from excessive combustion. The condition of the fuel bed may be readily watched by means of a pyrometer (in the discharge) and with proper adjustment of vapor and draught, temperatures may readily be held below this limit, especially with lignites.

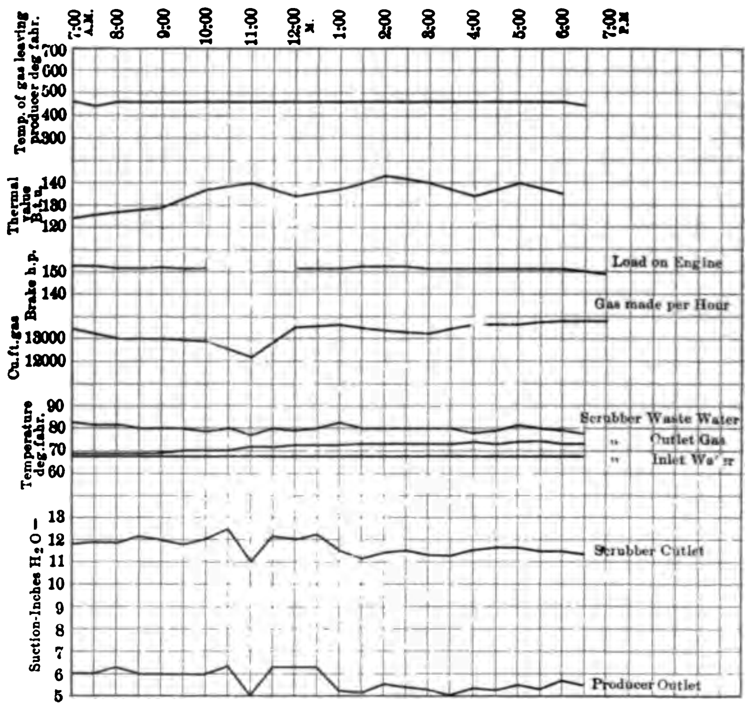


FIG. 5 TYPICAL LOG TEXAS LIGNITE, BEGINNING OF TEST

12 In this connection a comparison of these heat values with similar values from other plants is interesting, revealing the extreme range permissible with engines of modern design (Table 10). Both are fair operating plants, but deliver gas at a considerable variation from specified value (125 B.t.u.), without occasioning any disturbance in the operation of the engine. Results from the double-zone producer show that present engine ratings are well suited to the gas, a higher compression is permissible, and that a high hydrogen content —as high as 20 per cent—does not necessarily interfere with operation.

13 Perhaps the most important result is tar-free gas. The impurities normally consist of dust and lampblack. By the filter paper method, Fig. 7, it is possible to detect the least trace of tar, which quickly discolors through to the second layer of paper. Fig. 7 shows the maximum deposit from a run on Pittsburgh coal. Note that there is no discoloration of the second paper.

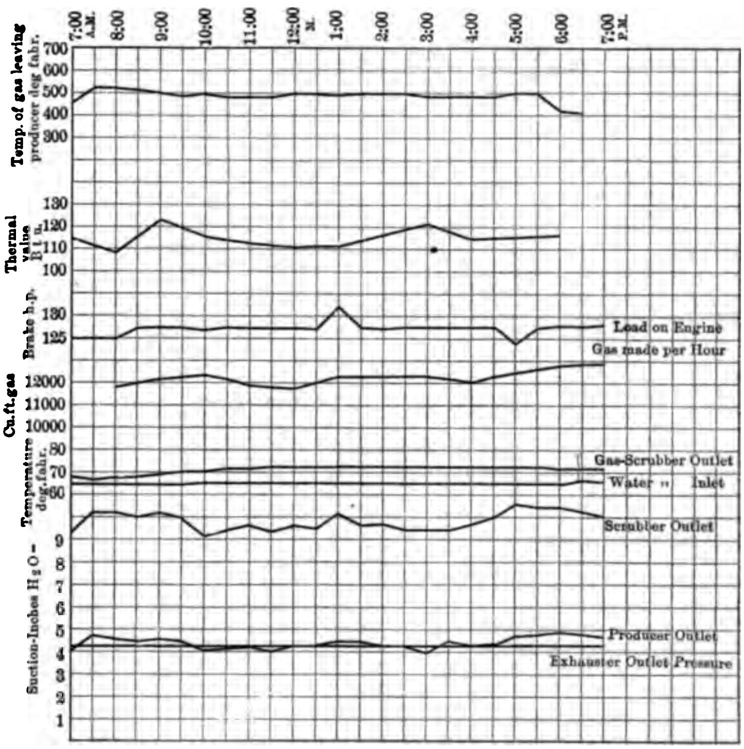


FIG. 6 LOG TEXAS LIGNITE, END OF TEST, 114 HR.

14 Test H, (Table 6), 25 determinations of Pittsburgh run-of-mine, shows well under 0.02 gr. per cu. ft., which is below the usual guarantee. In Tests M and N determinations on Texas lignite averaged 0.0193 gr. per cu. ft. These results are borne out by results in the field. Seventy-three determinations at Denver[1] averaged 0.022 gr. per cu. ft. In these tests all of the gas determinations represent average gas drawn continuously throughout the day's run. In no case are

[1] Western Chemical Co.

TABLE 5 PRODUCER EFFICIENCY

Test	Producer Efficiency Based on		Plant Efficiency Based on	
	Total	Effective[1]	Gross Fuel	Net Fuel
A................	76.3	....	.....	.....
B................	78.8	....	16.75	.....
C................	76.5	....	16.8	.....
D................	77.2	....	13.9	16.05
F................	75.5	....	15.45	17.6
V................	.....	....	13.2	.....
J................	.....	....	14.7	.....
K................	.....	....	17.5	
L................	.....	....	17.7	.....
M................	76.8	69.8	17.5	.....
N................	76.95	70.2	17.6	.....

[1] Efficiency on effective basis from 7 to 8 per cent lower than on total.

TABLE 6 DUST DETERMINATION

TEST H PITTSBURGH RUN-OF-MINE

Load b.p.	Gas per hr. cu. ft.	Impurities in Gas, Gr. per Cu. Ft.		
		Average of Five Determinations	Max.	Min.
142	14910	0.02079	0.0432	0.0129
137	14310	0.02087	0.0398	0.0100
170	17740	0.01712	0.0318	0.0062
184	19260	0.01611	0.0287	0.0034
183	19160	0.01718	0.0459	0.0063
204	21511	....	......	......

TEST M TEXAS LIGNITE

Duration of Tests, hr........	114
Average Load, b.h.p........	140
Lignite Fired, lb........	28,467
Gas Made, cu. ft........	1,400,000
Average Impurities, gr. per cu. ft........	0.0193
Maximum, gr. per cu. ft........	0.0770
Minimum, gr. per cu. ft........	0.0010

TABLE 7 TYPICAL GAS ANALYSES

Test	Fuel	H_2	CO	CH_4	CO_2	N_2
A	So. American Lignite	17.4	10.4	2.4	12.4	56.6
B	Colorado Lignite	17.6	11.6	2.6	13.2	54.4
C	Pittsburg Coal	14.1	15.2	1.8	9.6	58.9

TABLE 8 CHARACTERISTICS OF LIGNITE GAS

WESTERN CHEMICAL CO., DENVER

Date	Heat Value (Total)				Impurities			
	No. of Determinations	Max.	Min.	Average	No. of Determination	Max.	Min.	Average
4- 8-09	3	129.6	112.4	123.0	4	0.0290	0.01567	0.0213
4- 9-09	4	121.5	113.7	118.3	.	......	......	......
4-10-09	4	123.9	106.1	113.1	.	......	......	......
4-11-09	.	.....	.....	.....	.	......	......	......
4-12-09	1	.....	.....	122.0	7	0.0617	0.0144	0.0284
4-13-09	2	115.4	111.0	113.2	6	0.0667	0.0081	0.0231
4-14-09	3	124.0	121.0	122.0	7	0.0264	0.0115	0.0184
4-15-09	2	117.3	113.7	115.5	3	0.0204	0.0051	0.0114
4-16-09	3	128.0	114.7	119.5	4	0.0198	0.0048	0.0124
4-17-09	1	.....	.....	133.3	3	0.0260	0.0209	0.0247
4-19-09	4	128.6	108.5	121.2	2	0.0194	0.0058	0.0126
4-20-09	4	128.0	108.5	117.8	3	0.0046	0.0018	0.0033
4-21-09	3	117.8	110.0	114.7	3	0.0095	0.0051	0.0051
4-22-09	3	130.0	111.0	121.0	2	0.0288	0.0113	0.02005
4-23-09	3	129.0	123.4	126.0	3	0.0048	0.0018	0.0033
4-24-09	2	126.0	106.4	115.7	3	0.0064	0.0036	0.0053
4-26-09	1	.....	.....	125.0	.	......	......	......
4-27-09	1	.....	.....	120.4	3	0.0429	0.0040	0.0179
4-28-09	3	141.6	127.0	133.2	3	0.0089	0.0026	0.0056
4-29-09	2	131.0	136.0	128.5	3	0.0099	0.0059	0.0073
4-30-09	.	.....	.....	.....	3	0.0172	0.0066	0.0123
5- 1-09	.	.....	.....	.....	3	0.0356	0.0216	0.0304
5- 2-09	.	.....	.....	.....	2	0.0587	0.0340	0.0463
5- 3-09	.	.....	.....	.....	3	0.0328	0.0067	0.0205
5- 4-09	.	.....	.....	.....	3	0.2325	0.0735	0.1381
Average				121.2				0.02227

TABLE 9 SCREEN TESTS—FUEL AND ASH—NORTHERN COLORADO LIGNITE

WESTERN CHEMICAL Co., DENVER

	FUEL	ASH
Over ½ inch	.58 per cent	23.5 per cent
Over $\frac{1}{16}$ inch	23½ per cent	33.0 per cent
Through $\frac{1}{16}$-inch	18½ per cent	43.5 per cent
	100 per cent	100 per cent

Samples represent about one bushel of material quartered down from stock pile.

TABLE 10 TYPICAL PRODUCER OPERATION

	PLANT A			PLANT B		
	TEXAS LIGNITE			POCAHONTAS COAL		
	Max.	Min.	Avg.	Max.	Min.	Avg.
Date of Observation	May 20–29, '08.			Apr. 1–9, '08.		
No. of Determination		20			24	
CO_2	11.6	8.0	10.0	11.6	8.6	9.9
O_2	0.9	4.0	0.63	.....	.....	.....
CO	14.8	12.0	13.7	19.4	11.4	15.7
H_2	19.3	11.84	15.3	12.4	4.7	9.09
CH_4	2.52	1.5	1.88	8.8	5.1	67.6
N	62.4	5.7	55.2	65.1	53.9	5.93
Heat Value Total	109.9	89.2	102.7	166.4	110.4	133.1
Heat Value Effective	.....	....	.....	157.8	101.0	124.5
Fluctuation B.t.u	.....	....	20.7	.....	.....	56.0
Fluctuation Per Cent = Avg. Value..	....	....	10.1	.....	.....	21.0

snap samples used. The latter method of testing should be rigorously avoided except for some special purposes, as it affords no indication whatever of average conditions.

15 In former papers[1] the writer has described the method of obtaining producer efficiency from isolated tests. Fig. 3 shows the close agreement of this theory with the fact based upon these several different kinds of coal tested. The interesting point is illustrated, *that the gas producer varied only 10 per cent in efficiency throughout its normal range of load.* This type will give approximately 70 per cent

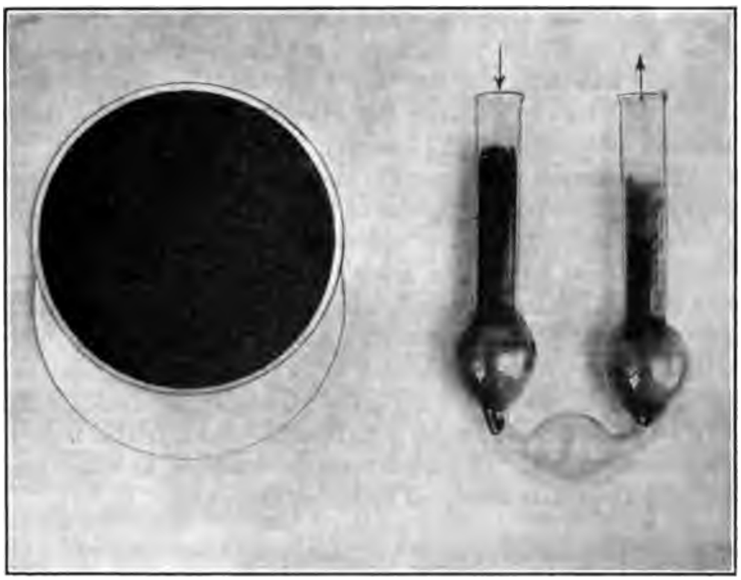


FIG. 7 COMPARATIVE METHODS OF TESTING FOR IMPURITIES

5 CU. FT. EACH, 30 MIN.
0.0258 GR. PER CU. FT. FILTER PAPER, TWO LAYERS
0.0216 GR. PER CU. FT. COTTON BATTING, TWO LAYERS

efficiency (on an effective heat basis), or 77½ per cent (total heat basis), at full load. This is manifestly reasonable by inspection of Table 5. That all fuels should fall so closely on the heat input lines at various loads, is a remarkable agreement, and closer than anticipated.

16 However, in a paper by the writer[1] the same agreement was found in plotting the results of tests on another type of plant at Rich-

[1] Norton Test, Vol. 29, Transactions A.S.M.E. 1907; Transactions A.I.E.E., page 1128, vol. 27, 1908

mond, Va. The standby, and three-load determinations, followed almost on a straight line of heat input to producer.

OPERATING RESULTS

17 **After the first year's period of tests, this plant was dismantled for examination. The gasification of 182,472 lb. of fuel showed no perceptible effect upon the condition of the producer, the walls being practically intact. This is due to the complete absence of clinkers and high temperatures. Experience plainly shows the latter to be the cause of clinker troubles. The fuel bed normally grades from small ash at the bottom through pure coke to green coal at the top. With proper handling, clinkers may be entirely avoided. For example, Table 9 shows a screen test of coal and ash at the Denver installation—43½ per cent ash through a $\frac{1}{16}$-in. screen.**

18 **An examination of a long gas main and the engine valves after the year's run, showed no deposits of tar either near or distant from the producer, indicating the complete fixation of the volatiles. All condensibles are removed in a static washer, the cells of which seem to clear themselves automatically of the deposit. For example the pressure drop or resistance through a month's run increased slightly more than ½ in. The principal skill in handling this producer is required in studying the characteristics of various kinds of fuels. Each must be handled differently for best results. With friable fuels the "let well enough alone" rule is particularly desirable, as resistance of the bed may be greatly increased by too much poking.**

GENERAL CONCLUSIONS

19 The writer's good fortune in obtaining results directly or in analyzing results from several types of producer plants, has led him to convictions on producer practice in general along certain broad lines. Knowing the facts regarding daily performance from both shop and field, the statements herein contained are believed to be conservative. It is not meant to convey the idea that successful working has been confined to any one particular type of plant or equipment. But those undertakings that have been backed up by experience gained in tests on a commercial scale are certainly most sure of success and worthy of support. There is much activity resembling plagiarism in the power gas field—guarantees based solely on productions from competitive results, portions of a foreign design

incorporated in an incoherent whole, etc. Such mis-matching inevitably leads to failure without thorough analysis of cause and effect. In short, the great desideratum is a campaign of investigation at the factory which shall not be of the shiftless pounds-and-kilowatt order, but sufficiently comprehensive to facilitate accurate interpretation. Many failures that now lie to the discredit of the internal combustion system would thus be avoided.

20 It is also patent that personal prejudice seemingly plays a far too important part in dominating the selection of plants, steam as well as gas, and its effect appears not only in the selection but also in the operation of the plant, making it often impossible to secure results which under proper conditions would be easily within reach. It is therefore as desirable to analyze reports of poor results obtained from a given plant, as of results which seem so good that they arouse suspicions of inaccuracy.

21 Finally, a present need in producer work is some reasonably accurate means of control indicating and compensating for "low gas," which may result from poor condition of fuel bed. Automatic production without the holder storage has now become an accomplished fact with the simplest apparatus, but no means are available for keeping under observation the heat value, except the cumbersome and delicate Junker calorimeters. Were it possible to alter in inverse proportion the ratio of air to gas at the engine accordingly, maximum efficiency could be maintained. But this variable factor has received practically no attention, and as a consequence producer operators are working entirely in the dark.

22 The principle of tar-free gas production has its demonstration in the type of producer under discussion, and it is believed the end justified the means, even at the slight expense of heat value. Due to unknown and complex reactions, tar laden or "green" gas possesses somewhat higher heat value than tar-free gas, due possibly to the preservation of more unstable hydrocarbons. This might then become a factor in rating, with engines designed with little or no margin. But good practice today recognizes no difference between gas of 110 and 125 B.t.u. The margin necessary is present for other reasons, and there is therefore no practical obstacle in the way of development of the tar-free type of gas producer in any form.

23 Further investigations are urgently needed as to the possibility of larger units. If the gas engine is considered behind the times in the matter of development of large units as compared with steam practice, the producer is hopelessly so. For rapid future devel-

opment, large units are an obvious necessity. We have steam turbine units capable of sustaining continuous loads of 30,000 h.p., boiler units of 2000 to 3000 h.p., gas producers of only a few hundred, maximum. The very multiplicity of units thereby required in the design of a large station, very seriously militates against the selection of the producer gas system of motive power.

DISCUSSION

G. M. S. TAIT. The results reported in this paper are entirely in accord with what we have found, namely, that the gas of the lesser British thermal units is much more satisfactory for engine practice. In other words, the efficiency of a gas of 90 B.t.u. is proportionately double that of a gas containing 600 B.t.u. per cu. ft., the gases in question being respectively blast-furnace gas and gasolene vapor.

2 I would like an expression of opinion as to the reason for this great discrepancy in efficiency between the two gases, my own opinion being that the excessive normal losses are due to the sudden high temperature developed in the gas of high B.t.u., which is greater than can be handled by normal piston speeds.

3 The tar washer used in this test appears to be a succession of water seals and I would like to know what would be the total frictional effect of these seals under normal conditions and on full load.

4 In all producers properly designed the thermal efficiency appears to remain constant between 20 per cent and 100 per cent load. I can confirm Mr. Bibbins' experience as to the action of this particular class of fuel and its desirability for producer work.

PROF. R. H. FERNALD. Mr. Bibbins places as his first essential requirement "continuous operation 365 days per year," and states that any departure from this condition means reserve equipment. He also states that the condition for producer operation must parallel steam boiler practice.

2 It is undoubtedly true that a producer which will operate continuously 365 days a year would prove a splendid commercial proposition, but it seems to me that in the requirements outlined the conditions imposed are much higher than those of any steam boiler plant and are beyond practical requirements. Every plant of any size must necessarily have one or more reserve units, as no plant can operate continuously 24 hours a day 365 days a year. If the producer described by Mr. Bibbins can approach this operating condition, it

will certainly revolutionize our present day power-plant practice. It would seem advisable, in the light of the present development of gas producers, to impose conditions which are less severe.

3 Relating to the adaptability of a single producer to all classes of fuel, it is well to bear in mind that the government testing station has practically demonstrated the fact that almost any variety and grade of our recognized fuels can be handled with more or less success in a given producer installation without change of details of design. It is questionable, however, whether such practice lends itself to the efficient use of a wide range of fuels. It is probable that better results can be obtained by utilizing a producer type to cover a certain range or variety of fuels and another plant of somewhat modified design for another range.

4 Mr. Bibbins refers to the excessive labor required by most producers. At the present time the labor requirements are excessive for the majority of the plants utilizing bituminous coal. This labor, however, even under bad conditions of operation, such as those involved when the fuel is one that clinkers badly, probably does not exceed that of the average steam installation, although the labor is of a somewhat different character. During the regular operating period of the plant this labor may amount to very little; but at the close of a week, two weeks, or any length of operating period, in the commercial plants now in operation, cleaning may be an exceedingly dirty, hot and tedious operation. With the steam boiler plant the labor is more uniformly distributed. In spite of the more erratic and more violent labor required at times by the producer installation, the total cost for cleaning, ash removal, etc., is probably within the limits of the average steam installation.

5 Experience with a large variety of fuels leads one to question whether the treatment accorded one fuel in order to prevent clinkering will produce the same results with a fuel possessing totally different characteristics. The impression from the tests carried on at the Geological Survey testing station is that fuels varying greatly in composition and in characteristics require widely different treatment. This impression has been obtained from tests on a large variety of fuels, but the number of tests on each of the different fuels was not sufficiently large to warrant positive conclusions regarding this point. European practice, however, seems to confirm this opinion, as practically every producer manufacturer finds it imperative to specify coals of certain characteristics for use in his type of producer and does not guarantee the plant on fuels outside of this class.

6 In the discussion of the results the point is brought out that with Texas lignite the rate of combustion in this producer can be so increased as to permit the same rating of the producer as when opering on a high-grade fuel. Note is made of the fact that a charging rate of 27.2 lb. per sq. ft. per hr. was obtained with this lignite. An installation in Texas, which I visited a year ago, consisted at that time of three producer units of 1100 h.p. rating each, or a total of 3300 h.p.

7 Owing to the character and high percentage of the ash, together with the excessive demands upon the plant each unit was cleaned every third day, or, what amounts to the same thing, one unit was cut out of operation during a part of each 24-hr. day. It required eight hours to cut out the gas from a given unit, to clean thoroughly, rekindle fires and cut in the new gas. During each 24-hr. day, then, the full plant capacity, rated at 3300 producer h.p. was in operation 16 hr., while only 2200 producer h.p. were in operation the remaining 8 hours. During the entire 24-hr. period, however, according to the operating records, the engines were developing 2800 h.p. The operating records also showed that the fuel consumption per square foot of fuel bed area per hour amounted to 33 lb. during the 16-hr. period and 48 lb. during the 8-hr. period.

8 The statement is made that the economy of less than 1 lb. per b.h.p.-hr. is probably below previous results in bituminous producers. It is assumed that this statement is not intended to cover the tests at the government testing station, which has reported a number of instances in which the consumption varied between 0.8 lb. and 1 lb. per b.h.p.-hr.

9 Mr. Bibbins states that perhaps the most important result is tar-free gas. It is undoubtedly true that tar-free gas is eagerly sought in all cases in which the gas is to be used in engines. In my own mind, however, it is somewhat questionable whether tar-free gas, as reported in this paper, means that the gas from any and all fuels used in this plant would necessarily be free from tar. Experience with a aroducer of somewhat different design shows tar-free gas with the m jority of fuels, but in the case of certain fuels the results are quite threeverse. If the producer under discussion can produce tar-free gas from any and all varieties of fuel, it is certainly a development in the right direction.

10 In the closing paragraph of Mr. Bibbins' paper the impr ssion is conveyed that the steam boiler units of 2000 and 3000 h.p. are found not infrequently, and that producer units are small in comparison with the usual boiler unit. In my opinion the condition at

TABLE 1 CAPACITIES OF PRODUCER-GAS POWER PLANTS

	No. of plants	Horsepower				Per Cent of Total	
		Total	Average	Minimum	Maximum	No.	h. p.
Anthracite Coal:							
Over 500 h.p.	8	7,550	950	600	1500		
500 h.p. or less	407	40,550	100	15	500		
Total	415	48,100	116	15	1500	88	43
Bituminous Coal:							
Over 500 h.p.	20	49,000	2,450	750	6000		
500 h.p. or less	17	5,150	300	35	500		
Total	37	54,150	1,460	35	6000	8	49
Lignite:							
Over 500 h.p.	3	7,275	2430	525	3750		
500 h. p. or less	19	1,725	90	25	250		
Total	22	9,000	410	25	3750	4	8
All Plants	474	111,250	235	15	6000	100	100

the present time is quite the reverse of this. In European practice it is not uncommon to find producer units of 1250 and 2500 h.p., and in the United States units of considerable size are in commercial operation, as shown by the accompanying summary of the producer-gas power plants operating in June 1909. There are undoubtedly over 500 plants in operation, as the list includes 474 (Table 1).

11 It is true that many of these larger plants are made up of several units, but an inspection of the original data shows the following single units of 500 h.p. or more:

h.p.	No.	h.p.	No.
500	4	1000	10
625	6	1500	1
750	3	2000	7

One single unit of 3,000 h.p. and one of 4,500 h. p. are reported, but these figures have not been verified.

12 It is interesting to observe that about 88 per cent of the total number of installations in the country are operating on anthracite coal (a few using charcoal or coke) and that bituminous coal and lignite are used in the remaining 12 per cent. It is not strange, therefore, that the majority of plants are at present made up of

relatively small units, although the number of large units is rapidly increasing as bituminous plants are becoming more common. In point of size the bituminous plants at present average 12½ times the size of the anthracite plants. Of the total horsepower approximately 57 per cent is derived from bituminous coal and lignite, and 43 per cent from anthracite coal, charcoal and coke.

13 Although in large central stations there are many operating advantages in relatively small units, yet it is believed that in the near future central station development will demand equipment of much larger capacity. A consideration of the fuel resources of the country indicates that in order to keep the price of power developed from fuel down to a consistent figure

a Grades of fuel which warrant transportation, or which may be defined as "marketable," should be used with the greatest practical economy.

b The very large percentage of coal of so-called low grade which today is left at or in the mine must be utilized.

c Advantage must be taken of the large deposits of lignite and peat which are found in many sections of the country.

It is undoubtedly true that in general, under conditions which do not require the use of steam for other than power purposes, the producer-gas power plant meets the requirements of *a*. At present the only method of advantageously handling the fuels mentioned in *b* and *c* is in the gas producer, and the utilization of these lower grades of fuel on an extensive scale demands concentration of large power units within close proximity to the fuel supply.

W. B. Chapman. In Par. 3, among the different requirements for successful operation, is mentioned the prevention of clinkers. I think the formation of clinkers can be avoided by the prevention of blow-holes or chimneys which allow the air to blow up through the fire bed, making hot spots. The average temperature across the hot zone in a producer is seldom high enough to produce clinkers. It is only in the neighborhood of the blow-holes that a sufficient temperature is attained to form clinkers. If the excessively high temperature necessary to the formation of clinkers existed throughout the producer, a clinker a foot or so thick would form immediately across its entire width. When ashes are melted they tend to run together, forming a clinker. The way to prevent this is to agitate the fuel bed continually, just enough so that the molten ash running down cannot take a permanent set in large masses, but is constantly kept in small pieces.

2 The successful producer should keep the fuel bed at an even temperature and uniform density throughout any horizontal plane. If there is a lesser density in any particular spot, the air blast immediately makes for this spot, causing an uneven temperature. To obtain this uniform density and temperature I believe that it is necessary to use some sort of mechanical agitation by hand methods, as no man or group of men can maintain a fuel bed of uniform density and temperature throughout any given horizontal plane long enough to get satisfactory results from soft coal.

3 Another point is that the successful producer should be made in a variety of sizes. The principles used in the producer described do not seem to admit of such variety. If this producer is of large diameter, the draft will go down the walls rather than in the middle, and the upper zone will not get hot enough in the middle to drive the tar out of the coal. If the tar is not removed by high heat in the upper zone, it is sure to get to the engine.

4 A successful producer should not require a delicately balanced draft, for the "balance" is often difficult to maintain. Uniform density in the two zones is imperative in double-zone or balanced-draft operation, as otherwise the draft will vacillate from one zone to the other according to their varying density or resistance. The density is apt to change with the loads and with change of operators. The density will also change when the ashes are removed, as during this process a cavern is often formed which drops suddenly. In a producer of this type I have seen the vacuum vary from 2 in. to 18 in. in the lower or up-draft zone, and from 10 in. to 30 in. or more in the upper or down-draft zone.

5 In Par. 21, referring to the question of varying the air supply to the engine according to variations in the heat value of the gas, Mr. Bibbins says: "But this variable factor has received practically no attention and as a consequence producer operators are working entirely in the dark." To my mind the proper way of overcoming this difficulty would be to provide suitable mechanical means for maintaining uniform conditions in the organization of the fuel bed.

H. M. Latham. I think Mr. Bibbins has struck the keynote in regard to bituminous gas producers, when he says that the primary requisites are continuous operation and tar-free gas. There is no question in my mind that these are the most important considerations. Any producer which satisfactorily meets these requirements should have a large field of usefulness.

2 We have already seen from the figures presented by Professor Fernald, that the bituminous producer is at present the predominant type, and it seems probable that future development, especially in large units, will be along this line. In New England the high cost of anthracite coal suitable for use in producers of the strictly anthracite type, offers serious objections to its employment as a fuel.

3 As regards continuity of operation, while it goes without saying that a certain reserve power should be provided, yet it is frequently convenient and desirable in installations where power is required every day in the year, to be able to operate without calling upon the reserve, or in other words, to run absolutely without interruption.

H. H. Suplee. In regard to the question of continuous operation, I think Professor Fernald will remember that we have had a number of gas producers running continuously in this country and elsewhere, not for one year only, but for a number of years, but we did not call them gas producers; we called them blast furnaces. But I hardly think we care to run our producers continuously.

2 In regard to the prevention of clinkers by keeping the contents of the producer in motion, that solution was adopted in the Kitson producer ten or twelve years ago, by means of an inclined grate which was made to revolve slowly. As a result the contents of the producer were kept moving up and down, and at no time did any clinker form. The producer was discontinued, but for other reasons. The inventor of that apparatus based it, he said, on the idea that running water would not freeze, and that in the same way, any substance would be prevented from solidifying by keeping it in continual motion.

3 It must be remembered that in the operation of gas engines, the calorific power of the gas produced is not the essential thing, but rather the value of the charge actually delivered to the cylinder; and this can be made almost anything which may be desired, the proportion of air being regulated according to the richness of the gas so as to give a charge of practically constant heating value.

E. N. Trump. In making tar-free gas all of the valuable by-products are destroyed. If Mr. Bibbins proceeds to burn up the by-products from the gas in the centre of his producer, he will lose from 80 to 90 lb. of sulphate of ammonia per ton of coal, which would pay for a large part of the coal used in his producer, if it were recovered.

2 As to continuous operation, we have had one plant burning from 150 to 155 tons of coal per 24 hours, in continuous operation for the past ten years; the pressure has never been off that plant but once, and then for a period of two hours.

3 If the fuel bed in the producer is agitated, and plenty of steam provided, clinkering is almost entirely prevented. Agitation can be produced by continuously extracting the ashes at the bottom, thus uniformly loosening the bed. Even with a very deep bed almost no poking is required.

4 Our experience has been with Hocking Valley coal, which will not coke. With coking coals it is more difficult to prevent the clinkering, but the agitation by the special mechanism for removing the ashes prevents clinkering to a great extent.

H. F. Smith. While it is of advantage to run continuously, still in most plants it is desirable to start and stop the engines. The majority of manufacturing plants run from eight to ten hours a day, and it is of equal importance to be able to shut the producer down, and to start up again in the morning with a reasonably uniform condition of operation, within thirty minutes, say, of starting the plant. Whether or not the type of producer outlined here is adaptable to meet that condition is open to question.

George D. Conlee. I would like some information regarding the possibility of naphthalene formation by the gas producer. In coke-oven and coal-gas practice, if the heats are sufficiently low to prevent the formation of naphthalene, an excessive production of tar results. Either the one or the other will be present.

2 Regarding the possibility of removing sulphur from gas by reheating, in the manufacture of enriched water gas for illuminating purposes, the gas is passed through checker brick heated to about 1600 deg. fahr. The gas is then scrubbed with water, cooled and passed through iron oxide to remove the hydrogen sulphide. The passage of the gas through the checkers seems to have no effect on the hydrogen sulphide, though it may change some other sulphur compounds to the sulphide.

The Author. In presenting this paper I have had misgivings that it would be considered by some as unduly optimistic. But I hope that I have been absolved from that charge through the simple showing of as complete facts as were at my command.

2 The producer under discussion is more or less the culmination of experiments of many years on different types. It represents the work of a number of engineers who have all striven for the perfection of the bituminous type in one form or another, and I feel safe in saying that the results are such as to give us some encouragement that the problem of gasifying bituminous coal is not as hopeless as supposed.

3 First let me define what is meant by continuous operation. While I think no commercial plant should have to shut down every fifth day to clean out, yet 365 days for the plant does not necessarily mean 365 days for the producer. Taking conditions such as normally exist in an electric light plant using steam boilers, we should expect a producer unit to run at least as long without excessive labor charge for cleaning and recharging. A small percentage of reserve equipment is always essential, but 100 per cent is certainly not required.

4 If the producer is to stand by itself, there is no occasion for especial leniency, i. e., we should demand from the designers a grade of service equal to that rendered by present steam plants, and from present indications this can be obtained.

5 These high rates of combustion—30 to 50 lb. per sq. ft. grate area mentioned in the discussion—are interesting, but it must be borne in mind that sometimes the amount of coal fired includes the additional fuel for building new fires. It is apparent from the Norton test that a very considerable proportion of the total coal fed into the producer was withdrawn at the end of a normal run, and if the heat equivalent of this fuel be deducted the rate of combustion will be lowered considerably. So, in comparing intermittent and continuous types of producers, it is necessary to take this extra fuel into account, for in the case of very frequent recharging the net loss is high.

6 The size of producer mentioned by Professor Fernald is rather extraordinary. I think not many of us realize that 3000-h.p. producers are being built. If it was a two-shell producer (the two rated as a simple unit) it should hardly be compared with the single shell producer on the same basis.

7 The sensitiveness of the balanced draft method of control has, I think, been overestimated by Mr. Chapman. While it is stated in the paper that the two control valves should be permanently set, I presume it would be recognized that these valves are put there to correct any inequalities or deficiencies in the fuel bed. When the producer is properly operated the valves need little or no adjustment, otherwise they must be adjusted occasionally.

8 I do not quite agree with Mr. Chapman's statement that it is impossible to maintain uniformity of the fuel beds with hand firing. When the plant illustrated was visited I noted this point especially by the aid of a simple apparatus. This is a double poker, consisting of a section of pipe with a solid rod through the center. By shoving both down into the fire and pulling out the rod and covering the pipe with a glass at the top, the condition could be noted. It was interesting to see that when the top of the fuel bed appeared practically dead, just under the surface it was at the proper temperature. I did not find the irregular conditions of fuel bed which Mr. Chapman mentions and I do not think it was merely a coincidence. The tendency towards short circuiting which he fears in large producers is not as marked as might be expected, excepting with wet peat, possibly owing in part to conditions.

9 As to the sulphide which Professor Rautenstrauch mentions, I can only say that it has not to my knowledge caused trouble. I have seen engines running successfully for a time on by-product coke-oven gas where it was found that by keeping the rods as hot as possible the deposition of sulphur was avoided and the consequent corrosion of the rods. As far as I know napthalene has not created similar trouble. A napthalene formation is characteristic of the distillate process where the higher hydro-carbons form the greater percentage of the heat value.

10 It is encountered in by-product coke-oven gas to some extent. But the difficulties arising from deposition of napthalene seem to be confined to delicate measuring instruments rather than the engine valves or rods which seem to be at a temperature sufficient to dissipate the accumulation. In producers the heats are run well above the destructive point.

11 Mr. Smith seems uncertain as to the possibility of the producer under description retaining its condition over periods of daily shutdowns. Table 1 shows a period of 18 days—432 hours—during which the producer was entirely idle for 23-hour periods. After a night's shutdown 15 minutes usually suffices to bring the fire into normal temperature conditions.

12 The automatic variation in the proportion of air and gas to the engine according to the richness of gas delivered to it is a problem of engine design relating to regulation of mixture. Designers must face the possibility of variations in gas from the best producers, and I do not believe any mechanical agitation of the fuel bed will avoid this necessity. In a plant employing a 15,000-ft. mixing holder I

have observed a puff of rich gas (liberated just after charging) make its way clear through to the engine at regular intervals quite destroying the mixture for the moment.

13 Mr. Trump assumes that the breaking up of hydrocarbons occasions a serious loss of efficiency not encountered in the generators of tar-laden gas. Just what are the precise reactions seems to be unsolved, but in the last analysis only one factor is uppermost—the comparative efficiency of the two systems. I doubt that much over 70 per cent is obtained from either process and less when the power consumption of tar extracting auxiliaries is taken into account.

No. 1265

THE BUCYRUS LOCOMOTIVE PILE DRIVER

By Walter Ferris, South Milwaukee, Wis.
Member of the Society

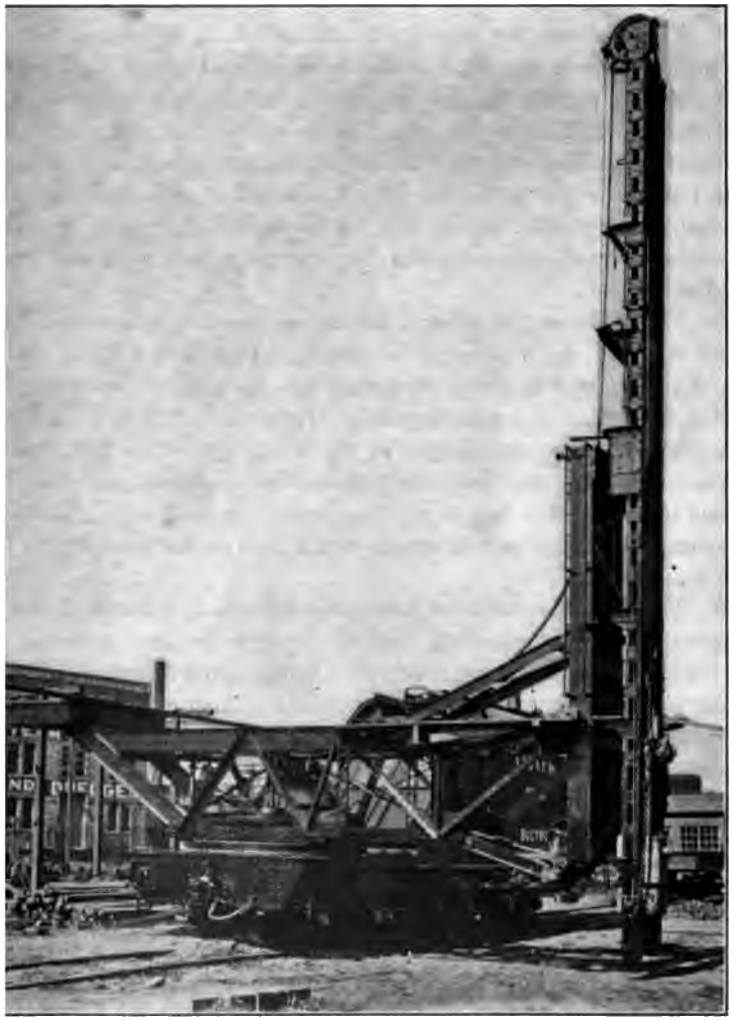
The machine described in this paper is of some engineering interest as the most substantial and complete railway pile driver yet produced. Its special claims to consideration as a new development in mechanical engineering, however, lie in the unusual arrangement and strength of the self-propelling mechanism, and in the self-contained hydraulic turntable, whereby the entire machine, including trucks, is quickly lifted clear of the rails and turned end for end. The propelling engines, mounted on the car body and delivering more than 250 h.p., are connected to the axles of ordinary bogie trucks without interfering with the movements of the trucks in turning curves, passing over frogs, and the like.

2 The machine was designed to meet the requirement of the Atchison, Topeka & Santa Fé Railway system, for a pile driver capable of climbing any grade on their line and hauling its own cars of piles, tools, etc. The self-propelling pile drivers built hitherto are capable of moving themselves for short distances while at work, but from lack of sufficient steam capacity as well as engine power must have a locomotive in constant attendance. The services of this locomotive are usually charged against the bridge department of a railway at the rate of from $20 to $30 per day. After having used several of the ordinary self-propelling machines, A. F. Robinson, bridge engineer of the Santa Fé system, prepared specifications calling for a pile driver of much higher propelling power. This resulted in the designing by the Bucyrus Company of the machine herein described, which has been in active service on the Santa Fé lines since January 1909.

3 The general appearance of the machine is shown in the illustrations. Fig. 1 shows the machine with leaders folded in shipping position. Fig. 2 shows the leaders up ready for driving, with the swinging frame turned across the track, and also shows how the coun-

Presented at the Annual Meeting, New York, (December 1909), of The American Society of Mechanical Engineers.

Fig. 1 The Bucyrus Locomotive Pile Driver with Leaders Folded in Shipping Position



THE LEADERS IN POSITION FOR DRIVING WITH THE SWINGING FRAME ACROSS THE TRACK terweight on the opposite side of the swinging frame balan weight of the leaders, keeping the machine always in a stable tion. In this position a pile can be driven 19 ft. from the cente track.

4 Fig. 4 shows the machine standing on its hydraulic tu with all wheels in the air. In this position and without any b the pile was picked up, put in place in the leaders and driv distance of 32 ft. from the center of the track. It was not d to drive this pile all the way in and the leaders were therefore down to clear the partially driven pile. The principal use hydraulic turntable, which will be described later on, is t the machine end for end when there is no railway turntable available.

5 Fig. 3 shows the general arrangement of machinery. ' is 40 ft. long, built entirely of structural steel and steel c On the front end is mounted the swinging frame, shown in Fi and 4, consisting of a pair of parallel trusses supporting the lea one end and a counterweight at the other end with the ne parts for raising and lowering the leaders and swinging the frame to the right or left at right angles to the car body. Thi is swung by a large worm wheel, which also serves to raise an the leaders.

6 The latter operations are accomplished by means of t worm-wheel hub projecting upward through the center pintl which the swinging frame revolves, a double-grooved sheave c being keyed to the upper end of the worm-wheel hub. This provided with a clutch by which it can be engaged with the ma plate of the revolving frame. When this clutch engages w swinging frame the latter moves with the worm wheel. Wl clutch is out of engagement, however, and a brake is applied b the car body and the swinging frame, the revolution of the wheel does not carry the swinging frame with it, but merely tu drum, which is keyed to the worm wheel.

7 The ropes leading from the drum to either end of the re frame are so arranged as to raise or lower the leaders. The of the worm wheel, drum, clutch, etc., are clearly shown in This figure also shows a large circular base plate on the car for s ing the weight of the revolving frame. The latter is provided w conical rollers which rest upon the finished upper surface of t plate.

8 From Fig. 3 it may be seen that the leaders are mount

FOLDER No. 3.

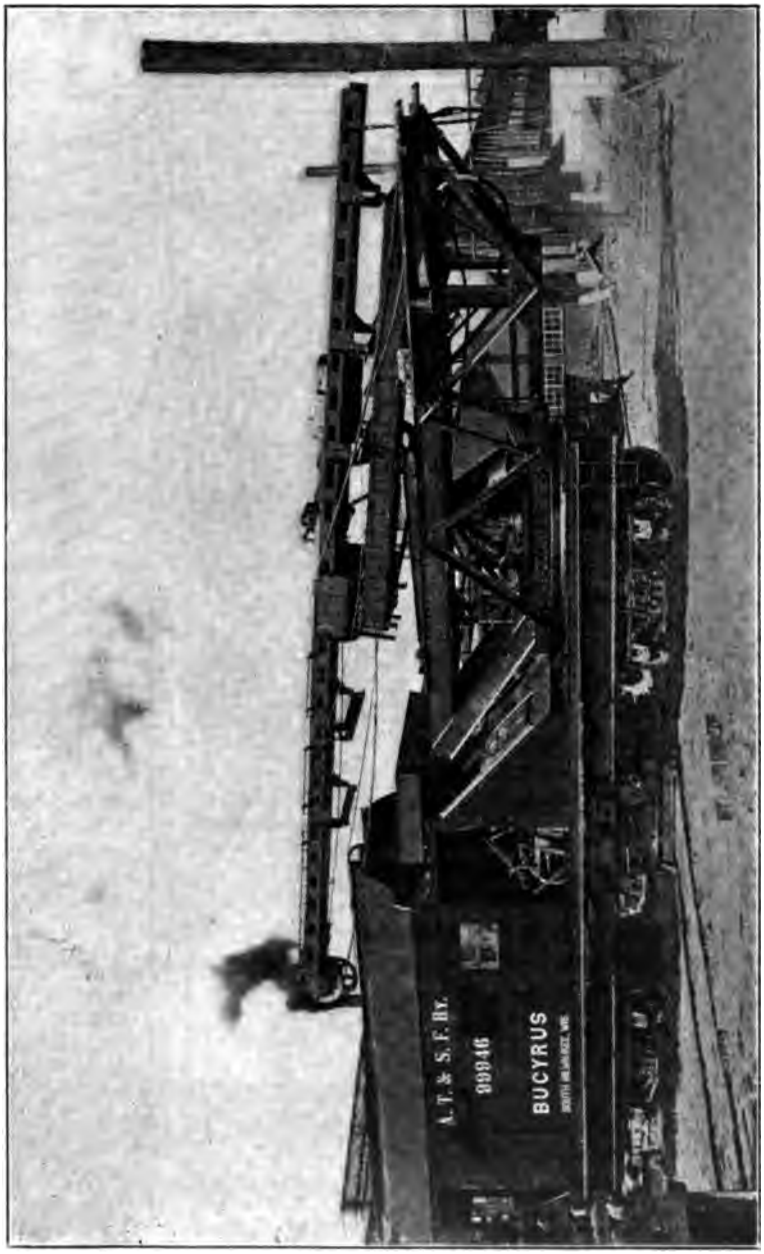
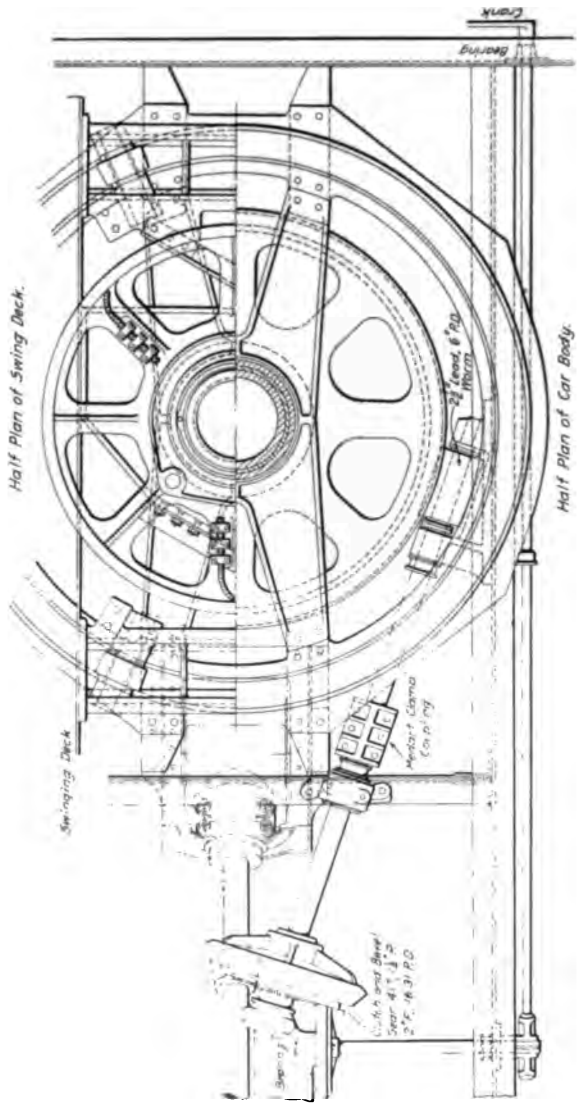


Fig. 4 The Pile Driver Standing on its Hydraulic Turntable with Both Trucks off the Ground


Crank
Bearing
Half Plan of Swing Deck.
Half Plan of Car Body.
Swinging Deck
Medart Clamp Coupling

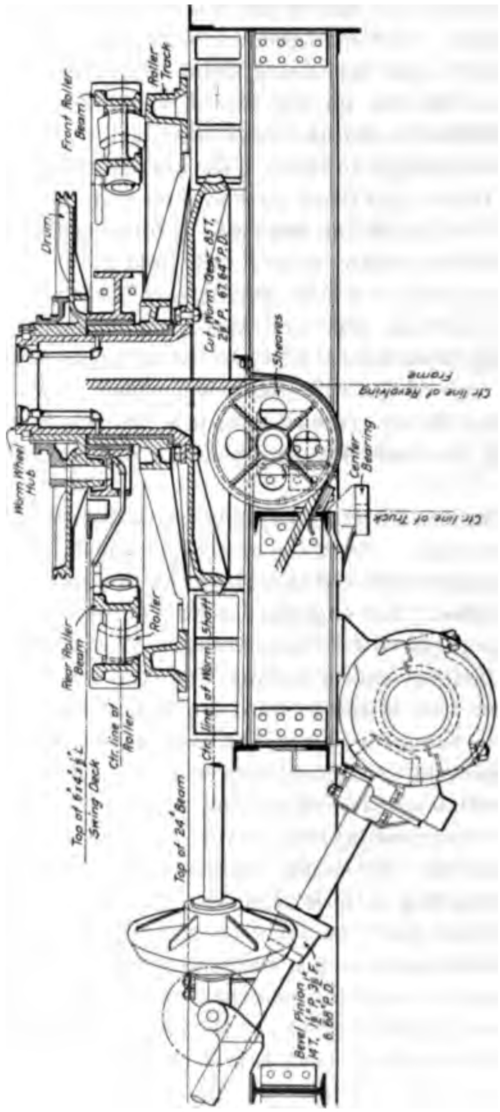


FIG. 5 PLAN AND SECTIONAL ELEVATION OF THE SWINGING AND PROPELLING MECHANISM OF THE LOCOMOTIVE PILE DRIVER

leader-raising frame by means of a pivot near the center of the leaders. A screw and nut device takes hold of the leaders some distance below the pivot and with this they can be inclined either to right or left so as to drive batter piles. The arrangement for raising and lowering the leaders acts directly upon the raising frame, which is carried by two rolling trucks *A* which roll on the top of the upper chords of the swinging frame, while the radius arm *B* takes hold of the lower end of the raising frame, causing it to move in the arc of a circle as indicated. The ropes *C* and *D* over the drum pass around suitable idler sheaves and are anchored to the sliding crosshead *E* forming a closed circuit. From this crosshead the raising arms *F* take hold of the raising frame, transmitting the movement of the crosshead to the latter. The hammer-hoist rope, pile-hoist rope and steam pipe (the last-named is not shown) run up from the car body to the swinging frame through the large hollow hub of the swinging worm wheel. The steam pipe is on the center and the ropes are so close on either side that they work equally well with the leaders in any position with regard to the car body.

9 The main engines are 11 in. by 12 in., with double cylinders and Stephenson link motion. From the crank shaft the two drums for the pile-hoist and hammer-hoist lines are geared in the usual manner with cone friction clutches. The engines, however, are much more powerful than would be required for these drums. The propelling gearing consists of two inclined shafts leading from the crank shaft of the engine to the rear axle of the forward truck and the forward axle of the rear truck. From Fig. 3 it will be seen that each of these shafts carries on its upper end two bevel gears, while the crank shaft carries a sliding sleeve with a small bevel gear on one end and a large one on the other end, the two meshing respectively with the two pairs on the inclined driving shafts. By sliding the sleeve to one end or the other a fast or slow propelling ratio is obtained.

10 With the fast gear, on level or moderate grades and with moderate loads, the machine can readily be driven at 25 miles per hour and has been driven at 30 miles per hour. With the slow gear the engines are powerful enough to slip the two driving axles and thus obtain all the tractive force that can be had with about 80,000-lb. weight on drivers. The machine can thus be used effectively as a switching engine and will readily haul its own weight with considerable additional load over grades of 1½ per cent or more. The acceptance-test of the first machine built was a run of 32 miles up a grade averaging 75 ft. to the mile, with a maximum of 97 ft. to the mile.

11 The lower ends of the inclined propelling shafts shown in Fig. 3 are provided with bevel pinions. These pinions mesh with bevel gears cast in one piece with large sleeves, as shown in Fig. 6. These sleeves surround the driving axles, a cored hole through the middle of the sleeves 10 in. in diameter providing about 2-in. clearance around the axles. The sleeves are supported by brackets rigidly attached to the car body with babbitted bearings. All this gearing is fastened to the car body only and remains in line without regard to the swiveling of the trucks.

12 The connection by which the driving torque is communicated from the propelling sleeves to the axles is also shown in Fig. 6. It consists of a modified type of universal joint so arranged that there is nothing to interfere with the axle passing through the middle. The propelling sleeve carries at one end a large flange with lugs supporting two pins *G*; these pins engaging with two bronze bushed lugs *H* formed on the inner side of the toggle casting *I*. On its outer side it carries another pair of lugs *J* on an axis at right angles to the axis of the pins *G* and these lugs *J* are connected to a U-shaped driving yoke *K*. The open end of this yoke is again pin-connected to a bracket *L* which is keyed to the axle.

13 Both pins, *G* and *M*, are made much longer than the lugs which engage them, to permit end play due to the displacements of the axle, as shown on the plan view in Fig. 6. As these two pin axes are at right angles to each other their combined slip will take care of any movement of translation, while the combined revolution of the parts around the pins *G*, *M* and *N* provides for any possible twisting. The wearing parts involved are six steel pins and six bronze bushings, all of the same size, and all parts are so made that the wearing surfaces can be replaced without taking the truck from under the machine. The pins are made hollow and are packed for continuous lubrication.

14 The method of detaching the driving gears when it is desirable to ship the pile driver in a freight train is slightly indicated in Fig. 3, at the rear axle of the front truck, where an operating lever is shown taking hold of the bearing which supports the bevel pinion at the lower end of the forward driving shaft. This bearing and the pinion are mounted in a sliding support, which enables the pinion to be drawn out of mesh with the bevel gear, permitting the propelling sleeves and gears shown in Fig. 6 to revolve freely with no gears in mesh. The same arrangement is provided on the rear truck.

15 In order to provide the necessary steam capacity for these propelling requirements, the boiler required is nearly three times the

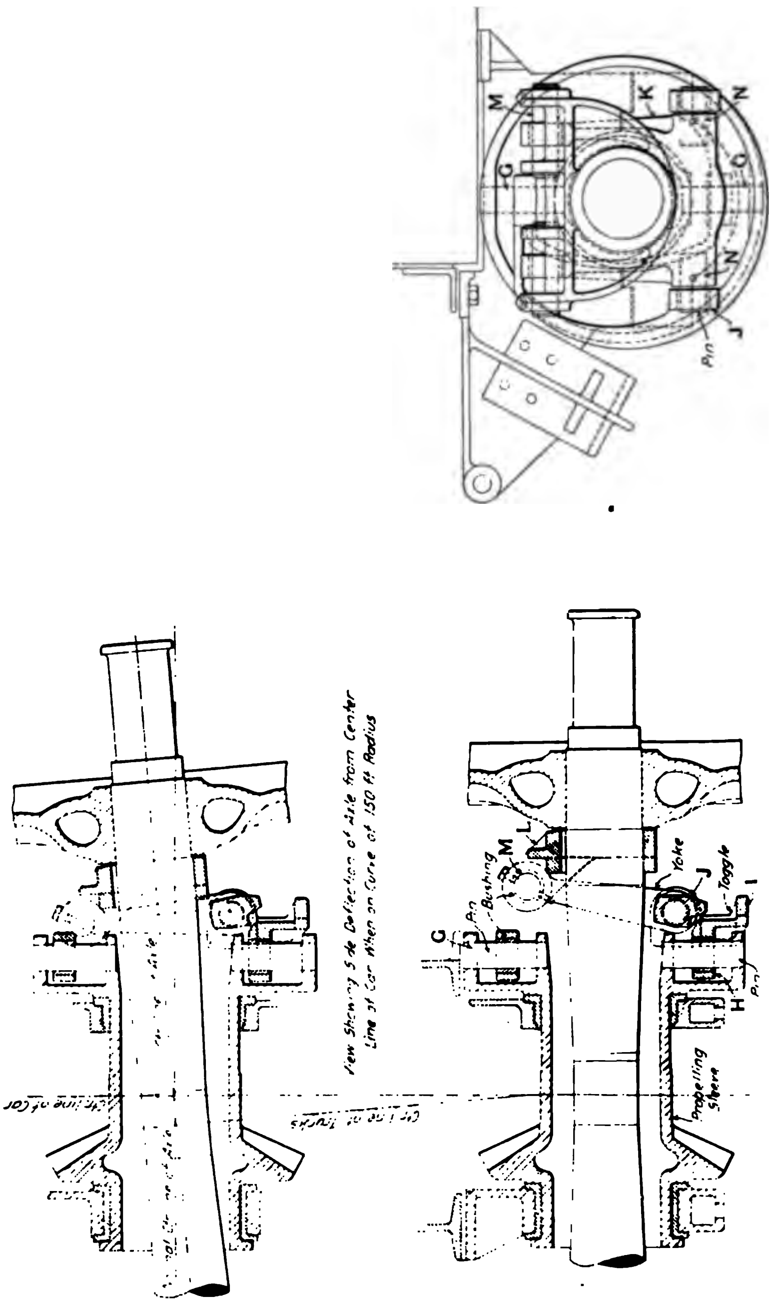


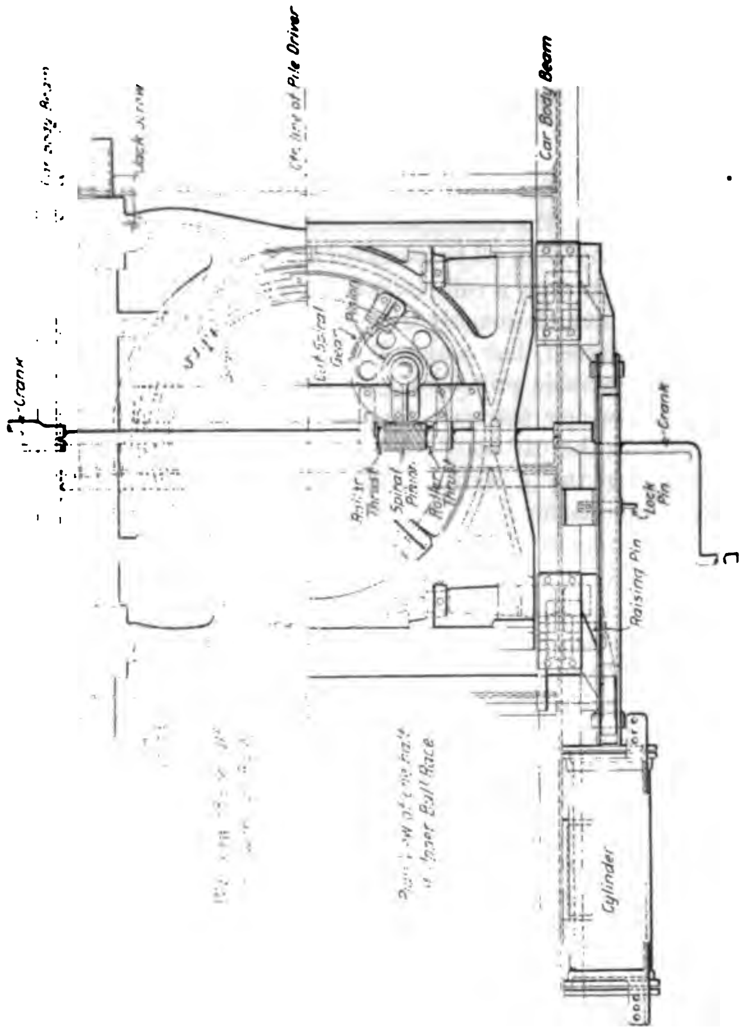
Fig. 6 Sections and End View of Toggle Joint

size of those ordinarily furnished for pile drivers. The boiler is of the locomotive type, 54 in. in diameter, 15 ft. 9 in. long, having about 800 sq. ft. of heating surface and designed for 175-lb. pressure. This pressure is required only for steam economy on propelling runs, as the engines are so large that all the ordinary movements of the machine can be made with 100-lb. pressure.

16 One of the striking features of the machine is the hydraulic turntable, which is shown in action in Fig. 4, and in shipping position in Fig. 1. It is frequently very important that a pile driver should be able to turn end for end or else to work at either end indifferently. The latter plan requires that the boiler and pile-driving machinery shall all be mounted upon a swinging deck, which can be turned through a full circle and reach either end of the car. This plan has been thoroughly tried and is satisfactory as far as pile driving is concerned, but makes it impossible to get a sufficiently powerful and reliable propelling gear between the engines and the trucks. In the new machine, therefore, the pile-driving apparatus is mounted on the car body where it can work at one end only, thus obtaining the powerful propelling drive already described. To reverse the machine the hydraulic lifting jack shown in Fig. 7 is attached underneath the car and under the center of gravity of the entire structure.

17 This jack consists of two ball-race castings having races about 5 ft. in diameter and is provided with 2-in. steel balls. The upper ball race is carried upon a set of four bell cranks or levers *O*, two on each side of the car, the bell cranks being pivoted upon brackets *P* attached to the main car beams. The upper ends of each pair of bell cranks are connected by a parallel rod, while the rear bell cranks on the two sides of the car are connected across by a heavy shaft *Q*. This arrangement compels all four bell cranks to act in unison, and when they are operated by the hydraulic cylinders the four pins from which the upper ball race is suspended move up and down the same distance, maintaining the turntable at all times parallel to the car, even though the center of gravity may be quite a distance away from the center of the turntable.

18 The system of bell cranks is operated by a pair of hydraulic cylinders 12 in. in diameter, having about 28-in. stroke. One cylinder is located on each side of the car. The cylinders have trunk pistons with sufficient area between the outside of the trunk and the bore of the cylinder to provide lifting force enough to raise the turntable away from the track and put it in shipping position. While lifting


Car Body Beam
Crank
Spiral Pinion
Roller Thrust
Lock Pin
Raising Pin
Cylinder

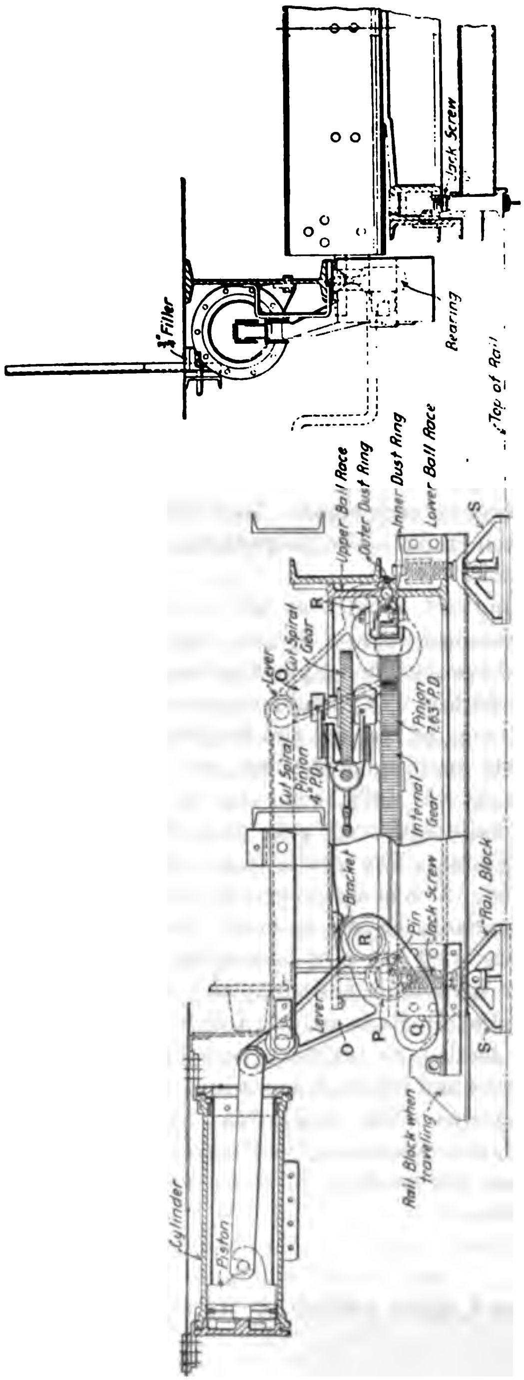


FIG. 7 PLAN AND SIDE VIEWS AND FRONT ELEVATION OF HYDRAULIC TURNTABLE OF LOCOMOTIVE PILE DRIVER

the car the pressure acts upon the full area of the 12-in. piston. The working pressure of about 200 lb. per sq. in. is provided by the boiler feed pump.

19 The lower ball race, which is suspended from the upper ball race by suitable clips, is also provided with a set of rail blocks s which rest on the rails and can readily be placed under the four jack screws, which are located in the four corners of the lower ball race. The lower ball race also carries a circular rack, while the upper ball race has a transverse shaft with a crank on each end and a double gear reduction to a swinging pinion which meshes with the rack on the lower ball race.

20 When the machine is to be turned it is necessary only to put the rail blocks under the jack screws and run the latter down until they touch the blocks. The entire car is then raised by pumping water into the hydraulic cylinders and is turned end for end by hand, two men working on each crank. In a high wind three men may be required on each crank. The entire turning operation occupies from 10 to 15 minutes.

21 An important incidental advantage of the turntable has already been touched upon in Par. 4. Fig. 2 and Fig. 4 show its use to enable the driver to reach a pile at a long distance from the center of the track. In this way, should occasion arise, any point within 32 ft. of the track may be reached and the pile driven.

22 The tests made since the first machine was put in operation indicate that it will fully come up to expectations. The first machine was built with slow gear only, having a maximum speed of 15 miles per hour. The results of its test on grades have already been mentioned. It has since been in constant use on the western divisions of the Santa Fé and on heavy grades. The fast propelling gear herein described has now been added and two machines thus equipped have been built and shipped. On one of these, built for the Canadian Pacific Railway, the following speed test was made. The machine hauled an ordinary passenger car from South Milwaukee to Racine and return, a distance of 12.6 miles each way. The run to Racine was made in 31 min., an average speed of 24.4 miles per hour, two miles being made at a speed of 30 miles per hour. The return run was made in 37 min., making an average speed of 20.5 miles per hour.

23 The shipping weight of the machine without the turntable, as shown in Fig. 3, is about 147,000 lb.; with the turntable, as shown in Figs. 1, 2 and 4, about 160,000 lb. It is equipped with either a No.

2 steam hammer or a 3500-lb. drop hammer, or both. The leaders are so made that either hammer can be used without change. The reach for driving piles is 18 ft. ahead of the center of the forward wheel, or 19 ft. on each side, as already mentioned; while with the turntable, 32 ft. on either side can be reached. The leaders are 40 ft. long. The construction is entirely of metal, except the house.

DISCUSSION

A. F. ROBINSON.[1] I feel very much pleased with the behavior of this driver as far as we have gone. I am especially pleased with the last three drivers, which are equipped with the extra high-speed gear. Our men find in handling this driver that it saves a good deal of time over the locomotive, especially in the short moves required in spotting the pile for driving and also the short run back to the end of a bridge to obtain piles.

2 As soon as this machine is thoroughly understood a great many will be used. This will especially be the case when we use reinforced-concrete piling more extensively.

L. J. HOTCHKISS.[2] There are in use many antiquated pile drivers which are slow and difficult to handle. In some cases the leaders must be raised by means of a set of blocks attached to the track ahead of the driver, the fall line being carried to a spool on the engine. With such a machine ten minutes may be required to raise or lower the leaders. Where the work is not too far from the station, and there are no overhead obstructions, it may not be necessary to lower the leaders when running to the station. In many places, however, the leaders must be lowered every time the pile driver goes in, and raised again on coming out. On a busy single-track railroad this may cause much loss of time in the course of the day.

2 The time loss may not be merely that directly caused by slow handling of the machine. In many locations the movement of trains is such that there are several periods during the day when with a quickly operated driver there is just time between trains to run out, drive one or two piles and get in the clear again. With a driver operated as previously described this cannot be done, as so much time is required to handle the leaders that there is none left for driving piles. There are, however, drivers which do not have this objection but which

[1] Bridge Engineer, Atchison, Topeka and Santa Fé Ry.

[2] Asst. Bridge Engineer, Chicago, Burlington & Quincy R. R., Chicago, Ill.

must be handled by a locomotive. This is expensive in two ways. There is charged to the work of pile-driving the cost of engine service, and the locomotive is kept out of regular train service. In times of heavy business the latter item is in itself one of considerable importance

3 The self-propelling feature of the machine described by Mr. Ferris, its large boiler capacity and the arrangement for turning it, are its most prominent features. As stated by Mr. Ferris, the usual charge for a locomotive and crew is from $20 to $30 per day, $25 being assumed as a fair average charge. The locomotive will furnish steam for the driver, making a fireman on the latter unnecessary. In the case of the self-propelling driver it is necessary to have a fireman, and as the machine is somewhat complicated, better men must be employed both as engineer and as fireman than would be needed ordinarily. For this reason the net saving by cutting out engine service probably will not exceed $20 per day. It is not unusual to have from 600 to 800 piles to drive on one division in a single season. If we estimate that 20 piles a day are driven, and this number is well above the average, 30 days will be required to drive 600 piles. For this period the charge for engine service would amount to $600, which is 5 per cent on an investment of $12,000. It will thus be seen that the elimination of engine service in pile-driving work is a matter of no small importance.

4 A machine such as Mr. Ferris describes has sufficient power and steaming capacity to handle its own train a considerable distance. Where a long haul is to be made the propelling mechanism is quickly thrown out of gear and the whole outfit put in a regular train. One of these pile drivers recently handled a train consisting of four bunk cars, a locomotive tender fully loaded with coal and water, one car containing 40 tons of coal, and a way car. This train was taken up a 1.4 per cent grade more than a mile long. A few days later this driver hauled 140 tons in addition to its own weight up the same hill at about 7 miles per hour. The steam gage showed 175-lb. pressure when the top of the hill was reached

5 The conditions of railroad operation today require that all possible economies be made both in operation and construction. The locomotive pile driver of large capacity is a recent development and one which must still be regarded, to a certain extent, as an experiment. Experience so far, however, indicates that it is an economical machine, in that it dispenses with locomotive service and is quickly handled on all classes of work.

THE AUTHOR. The railway pile driver is used for two general classes of work, construction and maintenance For construction work, in most cases, almost any track machine which is capable of driving piles will answer the purpose fairly well, because in such work the machine, if fairly well fixed, is able to stand for considerable periods of time at one place, and efficiency as a pile driver is the leading object.

2 In maintenance work, however, which generally consists in repairs, such as strengthening the abutment of a bridge which is showing some signs of washing down, or especially in repairs after a washout, the mobility of the machine is the leading feature. To illustrate this point, I may say that the first machine of this design which we built was tried out at a bridge in California which was a mile and a half from the nearest railroad siding. I happened to be with that machine at the time, and during the forenoon we ran it out from the siding to the bridge we were repairing, and back into the siding again, seven times, to dodge passing trains. During this time twelve piles were driven, one or two at each trip.

3 The base price of this machine is $11,650 without the turntable and the steam hammer. As the turntable and steam hammer, and electric light plant and other extras are added, the total price may run to something about $14,000. This represents an increase of cost to the railroad, above what they have been accustomed to pay for a pile driver, of $3,000 to $4,000 for each machine. The experiment in the case of this machine was quite as much in the line of commercial engineering as of mechanical engineering. When we built the first machine we were a good many thousand dollars behind, and it was somewhat doubtful if we would get it back. It looks now as if the machine would take very well. The operating department of the Southern Pacific, to which we recently furnished a machine, had previously charged the bridge department $45 a day for the use of a locomotive, which was dispensed with by the use of a machine capable of doing its own propelling work.

No. 1266

LINE-SHAFT EFFICIENCY, MECHANICAL AND ECONOMIC

By Henry Hess, Philadelphia

Member of the Society

The efficiency to be treated in this paper is that of the line shaft considered as an element for the transmission of power.

2 The complete power transmission system is made up of the shaft and pulleys; the belts, ropes or other equivalents; and the journals supporting all of these.

3 The difference between the power delivered to the system and that delivered by it is consumed in the work of bending and slipping the belts and overcoming the friction of the journals. There may be another loss due to the bending of badly aligned shafting; but as misalignment should not occur and as the remedy is obvious, it will not be considered further.

4 The power lost in the bending of the belts and in their slipping or creeping is but a small fraction of the total loss, and one, moreover, that cannot be materially lessened; assuming, of course, that belts are kept properly pliable and not allowed to dry out, become caked with dust or stiffened with adhesive dressings, all causes of loss of belt efficiency that no good shopman will allow to exist.

5 There remains the journal friction. In the average plant this accounts for nine-tenths or even more of the entire line-shaft losses. Included in the journal friction are the losses at the loose-pulley bearings and the countershafts.

6 The coefficient of friction of plain babbitted or of cast-iron bearings ranges all the way from ½ of 1 per cent to 8 per cent. This range covers all of the many methods of lubrication in general use. The better value is rarely realized outside of the laboratory; the poorer value is by no means as rarely found as it should be. A showing of 3 per cent friction coefficient is one that the manager may well pride

Presented at the Annual Meeting, New York, (December 1909), of The American Society of Mechanical Engineers.

himself on; while a coefficient of 5 per cent is much more general, but need not be taken, under existing conditions, as reflecting adversely on attention to details.

7 The remedy obviously lies in the substitution of roller bearings for plain bearings. In other fields than line shafting this remedy finds considerable employment; in some the plain bearing has indeed been superseded almost entirely. This is particularly the case where the power efficiency is of great importance, as for instance, in the automobile.

8 While some shopmen still doubt the reliability of the ball bearing, those who have followed the development of modern machinery know that hundreds of thousands of ball bearings are carrying loads varying from a few ounces to many tons, day in and day out, at speeds ranging from a few turns per minute to 10,000 or more. They realize that it is not a question of reliability *per se*, but one of selection of sizes suitable for the loads to be dealt with.

9 In line shafting the economic question is to the fore. The first cost of a ball-bearing installation is greater than a plain bearing equipment. Will it pay for itself by the savings effected and if so at what rate? What return on the difference in investment can be realized? That there is a saving is generally known, but accurate figures are wanted by which a manager can justify his recommendation to those who control the purse strings and are responsible for dividends.

10 When only the idle running of the line shafts is considered, answers to these questions can be easily obtained, now that electric motors are so generally applied directly to line shafts and it is so simple a matter to take readings of the power delivered to them. The difference in readings for the same shafts with plain and with ball bearings represents fairly accurately the saving for the idle run.

11 But line shafts are not put up to run idly; they drive machines and these machines are sometimes heavily loaded, sometimes lightly loaded and sometimes idle. While comparative current readings taken under these conditions may be fully satisfactory to those immediately concerned, this rather crude method cannot lay claim to that accuracy which is being demanded more and more by the engineering world.

PLAN OF TESTS[1]

12 To supply definite information the author decided that a series of comparative tests should be made, involving no variables other than the bearings themselves. In order further to eliminate possible personal bias in favor of the ball bearings the author called on Messrs. Dodge & Day to make these tests, giving them carte blanche as to methods, with instructions confined to a demand for definite and reliable figures. This investigation was the first undertaken, so far as the

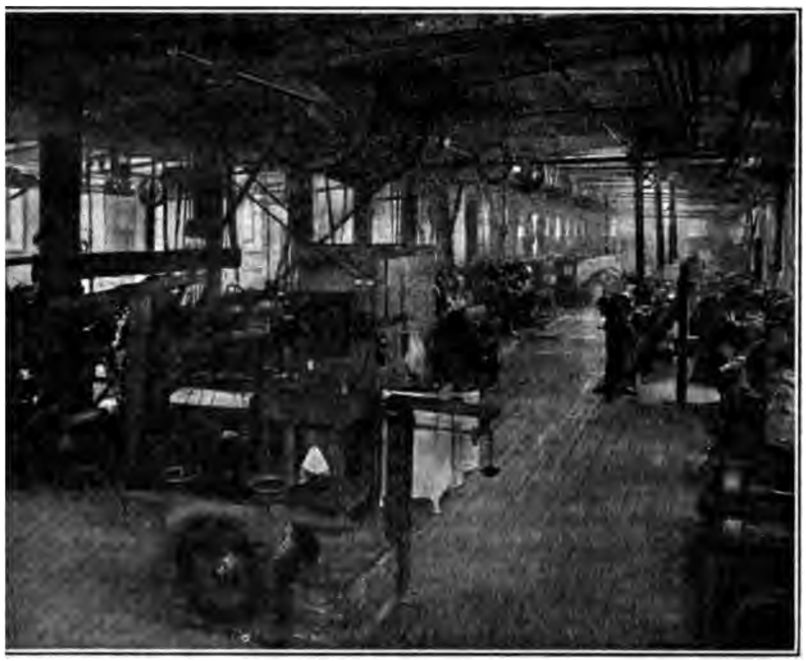


FIG. 1 VIEW SHOWING LINE SHAFTING TESTED

author knows, under conditions practically those of the workshop, the sole difference being the substitution of constant loads for the variables of ordinary working.

13 Besides the change in load due to the operation of the various

[1]The hangers employed in the tests were made by the Geo. V. Cresson Company, Philadelphia, and the plain bearings used were of the regular babbitt-lined ring-oiled type, made by the same company.

machines driven from a line shaft, already referred to, there is the change in load due to variation in belt stress. A preliminary test quickly demonstrated that reliance could not be placed on the use of tension weighing clamps in putting on the belts. The tension was found to differ from that determined by the clamp scales. This error could have been minimized by the use of the admirable methods and apparatus worked up by the engineers under our Past-President, Fred.

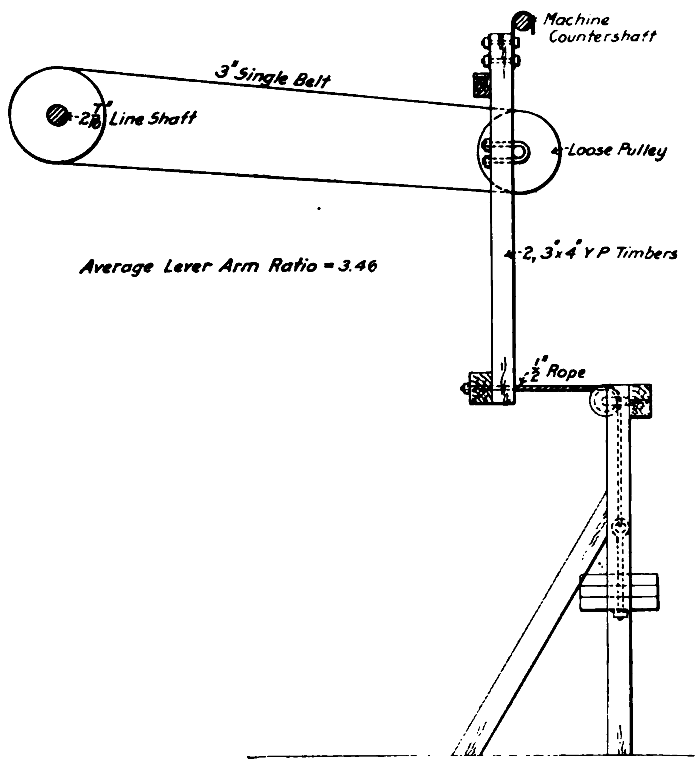


Fig. 2 Arrangement of Countershaft Frames Used During Comparative Tests of Ball and Ring-Oiling Bearings

W. Taylor, and this plan was given serious consideration until it was found that the influence of varying humidity and temperature in the shop was such as greatly to change the tension of the belts even after they were in place.

14 The complete plan finally decided on and carried through was as follows: A line shaft of $2\frac{7}{16}$-in. diameter and 72-ft. length used

to operate a series of heavy turret lathes was set aside for the test. This was alternately equipped with plain ring-oiling babbitted boxes and Hess-Bright ball bearings. In order to facilitate the exchange

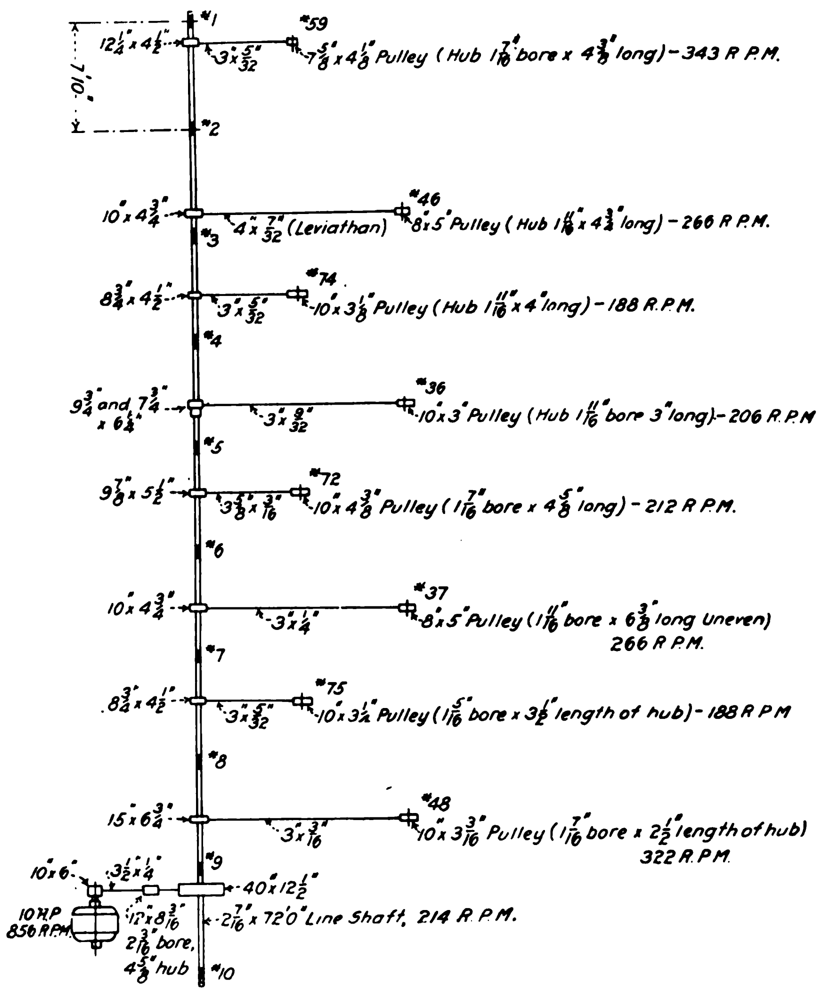


FIG. 3 ARRANGEMENT OF LINE-SHAFT AND BELT DRIVES

of bearings, the ball bearing boxes were placed on the shaft close to the hangers, making it necessary only to slip the plain bearings out of the hanger and slip the ball bearings in. Both types were held by the same supporting screws usual with modern hangers.

15 It is perhaps needless to mention that great care was exercised in seeing that the shaft was correctly aligned at the beginning of each test. It was supported by ten hangers, with an average spacing of 8 ft. See Fig. 1 to Fig. 3, the last of which shows the arrangement adopted to secure constant load.

16 The belts from the line-shaft drive pulleys mounted on swings hung at their upper ends, loaded by ropes attached to their lower ends. and leading over guide pulleys to weights. The tension in the belt is thus definitely determined by the weight and is independent of slight variations of belt length from whatever cause. The load on the journals is therefore constant and definitely known. The only possible variable is in the friction of the loose pulleys on the swings. These loose pulleys were ordinary tight pulleys picked up around the shop and arranged for oiling through the set screw holes and by the addition of channels. The friction was kept as nearly constant as possible by oiling at the beginning of every test. This answered fairly well except for two tests in which the rather small dimensions of the hubs gave rise to heating under the abnormally high belt tensions used.

17 Of such swings eight were employed. All were mounted on the same side of the shaft to avoid any uncertainty in load conditions that might have resulted from a possible balancing of pull from opposite sides.

18 Speeds and dimensions of shafts, pulleys, loose pulleys and loose-pulley hubs, belts and belt material are marked in Fig. 2. The drive was supplied by a 10-h.p. motor from the floor. The tension of the main belt was kept constant by a weighted idler pulley bearing on its driving side.

19 Constancy of line shaft speed was assured by a rheostat inserted in the field of the 110-volt, direct-current, shunt-wound motor and the use of a Warner tachometer connected to the motor. The electrical measurements taken were the voltage across the motor terminals and ammeter readings of the armature and field currents. The electrical and mechanical losses of the motor were determined from electrical resistance and no load tests. These motor losses were deducted from the total electrical output, the balance being the power consumed by the line-shaft system in journal friction, in loose-pulley hub friction, in belt bending and creep, in the motor belt tension idler and in windage. All instruments used were calibrated before and after the tests.

METHOD OF TESTING

20 The tests were divided into two duplicate series, the first, *A*, with plain bearings on the line shaft; the second, *B*, with ball bearings on the line shaft. The sole variable was therefore that of the line shaft as affected by the change from plain to ball bearings.

21 In each series the effect of varying loads was determined by changing the belt tension by approximately equal increments from 20 to 90 lb. per inch width of single belt. This was supplemented by a test with all the belts removed except the driving belt from the motor, leaving only the weight of shaft and pulleys for journal loads.

22 Each test lasted forty minutes of running time; a reading of the various instruments was taken every two minutes, making a total of 10 hr. 40 min. with a total of 960 recorded readings.

23 The loads on the line-shaft journals ranged from 126 to 662 lb. per journal; for the $2\frac{7}{16}$-in. by 10-in. ring-oiling babbitted bearings this gives loads ranging from 5.2 lb. to 27.3 lb. per sq. in. of projected area. For the ball bearings used, each of which had 12 balls of $\frac{9}{16}$-in. diameter, the load per ball was from 10.5 lb. to 55.2 lb.

24 It is pertinent to mention that these bearings have so far a record of nearly five years constant service under loads corresponding to about three-fourths of the maximum cited and show no evidence of wear. In that period they have been lubricated but three times, once when put up, once for the test and once incidental to a shop moving.

RESULTS SHOWN BY THE TABLES

25 Details of the loading of the line-shaft bearings and of the swing or countershaft idlers are given in Tables 1 and 2. Table 3 gives the averaged electrical readings for each test. Table 4 gives electrical readings with the power in kilowatts delivered to the belt and the percentage of saving due to the ball bearings. In the supplement to Table 4 is explained the derivations of the columns in the table in reference to the deduction of motor losses from total input.

26 Table 4, last column, shows that the saving due to changing ten $2\frac{7}{16}$-in. plain ring-oiling babbitted bearings running at 214 r.p.m. to the ball bearings increases with increasing belt tensions from 14 per cent to 36 per cent. With the more usual belt tensions of good practice ranging from 44 lb. to 57 lb. per inch width of single belt (tests 3 and 4), the saving amounts to 36 per cent and 35 per cent.

TABLE 1 RESULTANT PRESSURES ON LINE-SHAFT BEARINGS

Bearing No.	Line Shaft Dead Loads at Bearings Lbs.	Components of Belt Loads at Bearings (Lbs.)						Resultant Loads at Bearings (Lbs.)						Belt Number Corresponding to Bearing Loads
		Test Number						Test Number						
		1	2	3	4	5	6	1	2	3	4	5	6	
1	80.33	87.5	153.6	219.1	284.8	350.4	416.5	126	181.5	241.5	303.3	367.5	433	59
2	132.63	23.7 41.8	41.6 73.5	59.4 104.7	77.2 136.0	95.1 167.8	113.0 199.0	150.5	183.5	219	258	304.5	347.5	59 46
3	180.17	39.4 48.6	121.7 85.3	173.8 121.6	226.0 158.0	277.7 195.0	330.5 231.0	222.5	282.5	358	436	520	604	46 74
4	220.57	62.6 65.3	109.9 114.6	156.9 163.7	204.0 212.8	250.5 261.5	298.5 315.5	261	324.5	402	480	567	662	74 36
5	182.16	45.9 63.1	80.6 110.7	114.8 158.2	149.2 205.5	184.0 252.8	214.0 301.0	218	273.5	340	406	488	567	36 72
6	190.09	48.1 52.7	84.5 92.5	120.3 132.0	156.5 171.7	192.7 211.2	228.5 250.5	221.5	268	324.5	390	456	523	72 37
7	201.83	58.5 63.7	102.7 111.8	146.5 159.7	190.5 207.5	234.3 255.0	279.0 303.7	245	305.5	380.5	460	545	635.5	37 75
8	183.96	47.5 52.7	83.4 92.5	118.8 132.0	154.5 171.5	190.5 211.0	225.8 250.5	215	262	321	385	452	520	75 48
9	317.14	58.5 260	102.7 260	146.5 260	190.5 260	234.5 260	279.0 260	505	496	490	489	492	500	48 Motor Belt
10	138.71	60	60	60	60	60	60	185	185	185	185	185	185	Motor Belt
Total	1828.00	1210	1882	2548	3216	3884	4556	2350	2762	3262	3793	4377	4977	

TABLE 2 RESULTANT PRESSURES ON COUNTERSHAFT IDLERS

Pulley No.	Weight of Pulley and Half of Belt Lbs.	Belt Pulls on Idler Pulleys						Resultant Pressures on Idlers						R.P.M. of Idlers	Angle of Inclination of Belt from Horizontal
		Test Number						Test Number							
		1	2	3	4	5	6	1	2	3	4	5	6		
59	9.6	111.2	195.2	278.5	362.0	445.5	529.5	111.0	194.5	277.0	360.8	444.2	528.0	343	5° 38′
46	19.0	"	"	"	"	"	"	111.7	195.5	278.0	361.6	445.0	528.5	266	1° 16′
74	11.6	"	"	"	"	"	"	110.8	194.2	277.3	360.5	444.5	527.5	188	6° 41′
36	13.8	"	"	"	"	"	"	111.7	195.5	278.0	361.8	445.0	528.5	206	1° 55′
72	14.9	"	"	"	"	"	"	110.7	194.0	277.0	360.5	444.0	527.0	212	6° 51′
37	12.2	"	"	"	"	"	"	111.7	195.5	278.1	361.8	445.0	528.5	266	2° 0′
75	11.6	"	"	"	"	"	"	110.8	194.2	277.3	360.5	444.5	527.5	188	6° 30′
48	13.2	"	"	"	"	"	"	111.8	195.6	278.2	361.7	445.2	529.0	322	1° 16′
Average								111.3	194.9	277.6	361.2	444.7	528.1		
Motor															136° 0′0″

TABLE 3 AVERAGE ELECTRICAL READINGS FOR EACH TEST

Test No.	Date 1908	Total Resultant Loads on Lineshaft Bearings Lbs.	Average Resultant Pressures on Idlers Lbs.	Total Kw. Taken from Line	Supply Pressure Volts	Average Armature Current Amperes	Average Field Current Amperes
1*A*	2/4	2349.5	111.3	1.375	111.0	10.43	1.96
1*B*	2/6			1.302	109.6	10.1	1.79
2*A*	2/4	2762	194.9	1.560	109.5	12.3	1.92
2*B*	2/6			1.372	109.5	10.75	1.78
3*A*	2/4	3261.5	277.6	1.842	109.6	14.9	1.95
3*B*	2/6			1.516	110.0	12.0	1.82
4*A*	2/4	3792.5	361.2	2.051	108.6	17.1	1.82
4*B*	2/6			1.653	110.0	13.25	1.81
5*A*	2/4	4377	444.7	2.098	110.5	17.1	1.91
5*B*	2/7			1.802	109.4	14.7	1.81
6*A*	2/4	4977	528.1	2.238	109.9	18.4	1.92
6*B*	2/7			1.933	110.0	15.8	1.78
7*A*	2/1	2000	0	0.479	110.0	4.00	0.366
7*B*	2/6			0.381	103.0	3.33	0.381
8*A*	2/4	2070	0	1.107	111.0	8.00	1.98
8*B*	2/7			0.955	110.0	6.75	1.93

Tests 7*A* and 7*B* made with 1-h.p. motor. All other tests made with 10-h.p. motor. Motor, 856 r.p.m. Line shaft, 214 r.p.m.

TABLE 4 ELECTRICAL READINGS, NEW POWER IN KILOWATTS, AND PERCENTAGE OF SAVING DUE TO BALL BEARINGS

Test No.	Total Kw. Taken from Line	Ave. Line Volts	Ave. Field Current	Ave. Field Watts	Watts Delivered to Armature	Armature and Brush Resistance Loss	Armature and Brush Drop	Load + Iron Loss at Apx. Speed	E.M.F. at 856 R.P.M. / E.M.F. in Experiment	Load and Motor Iron Loss (856 R.P.M.)	Iron Loss + Pulley Bearing Loss	Kw. Delivered to Belt at 856 R.P.M.	% Saving
1A	1.375	111.0	×1.96	=217	1158	11	1.0	1147	$\times\frac{106.6}{110}$	=1112	586+60	0.466	14
1B	1.302	109.6	×1.79	=196	1106	10	1.0	1096	$\times\frac{100.7}{108.6}$	=1016	554+60	0.402	
2A	1.560	109.5	×1.92	=210	1350	15	1.2	1335	$\times\frac{105.2}{108.3}$	=1297	579+60	0.660	29
2B	1.372	109.5	×1.78	=195	1177	12	1.2	1165	$\times\frac{100.4}{108.4}$	=1079	552+60	0.467	
3A	1.842	109.6	×1.95	=214	1628	22	1.5	1606	$\times\frac{106.3}{108.1}$	=1579	585+60	0.934	36
3B	1.516	110	×1.82	=200	1316	14	1.2	1302	$\times\frac{101.8}{108.8}$	=1218	560+60	0.598	
4A	2.051	108.6	×1.82	=198	1853	29	1.7	1824	$\times\frac{101.8}{106.9}$	=1737	560+60	1.117	35
4B	1.653	110	×1.81	=199	1454	18	1.3	1436	$\times\frac{101.4}{108.7}$	=1340	558+60	0.722	
5A	2.098	110.5	×1.91	=211	1887	29	1.7	1858	$\times\frac{104.9}{108.8}$	=1791	557+60	1.154	25
5B	1.802	109.4	×1.81	=198	1604	22	1.5	1582	$\times\frac{101.4}{107.9}$	=1487	558+60	0.869	
6A	2.238	109.9	×1.92	=211	2027	34	1.8	1993	$\times\frac{105.2}{108.1}$	=1940	579+60	1.301	25
6B	1.933	110	×1.78	=196	1737	25	1.6	1712	$\times\frac{100.4}{108.4}$	=1586	552+60	0.974	
7A	0.479	110	×0.366	= 40	439	48	12.0	391	$\times\frac{92.5}{98}$	= 369	76+25	0.268	21
7B	0.381	103	×0.381	= 39	343	34	10.1	309	$\times\frac{95.4}{92.9}$	=317	81+25	0.211	
8A	1.107	111	×1.98	=220	887	6	0.8	881	$\times\frac{107.3}{110.2}$	=858	590+60	0.208	65
8B	0.955	110	×1.93	212	743	5	0.7	738	$\times\frac{105.6}{109.3}$	=713	581+60	0.072	

SUPPLEMENT TO TABLE 4

For 10-h.p. motor : armature resistance = 0.04 ohms, and brush contact resistance = 0.05 ohms to 0.06 ohms (2.5 sq. in.) for current densities not greater than 7.5 amperes per sq. in.; hence drop in armature and brush contact = 0.1 ohms × armature current, also loss in armature resistance and brush contact resistance = 0.1 ohms × (armature current)^2.

Iron loss = 5.5 amperes × e.m.f.

E.m.f. with 1.6 amperes in field = $(94 - 0.5)\frac{856}{850} = 94.2$; with 2.0 field amperes = $(108 - 0.5)\frac{856}{852} = 108$.

E.m.f. with any field current at 856 r.p.m. = $94.2 + 13.8 \times \frac{\text{field current} - 1.6}{0.4}$.

For 1-h.p. motor (tests 7A and 7B): armature resistance = 2.5 ohms, and brush contact resistance = 0.85 ohm $\left(\frac{5}{16}\text{ sq. in.} \times \frac{1}{2}\right)$, only half of brush in contact; drop in brush contact for 4 amperes (7A) = 2 volts, and for 3.33 amperes (7B) = 1.8 volts.

Iron loss = $\left(0.80 + 0.1 \times \frac{\text{field current} - 0.35}{0.7}\right)$ e.m.f.

E.m.f. with 0.35 amperes in field = $(95 - 0.8 \times 3.35)\frac{856}{884} = 89.4$ volts; with 0.42 field current = $(109 - 3.1)\frac{856}{880} = 103$ volts.

E.m.f. with any field current = $89.4 + 13.6 \times \frac{\text{field current} - 0.35}{0.7}$.

DISCUSSION OF RESULTS

27. Tests 5 and 6 with belt tensions of 70 lb. and 83 lb. per inch width of single belt show lower savings of only 25 per cent. This falling off is due to the fact that the pressures were too high for the loose-pulley hub-bearing surfaces, causing excessive heating and losses. This reduction to 25 per cent does not indicate a smaller actual saving due to the ball bearings, but simply that the decrease was due to improper excessive friction in the loose pulley hubs, particularly during the "B" runs. The pressure on the smallest countershaft pulley bearing surface during these tests, Nos. 5 and 6, rose to 124 lb. and 148 lb. per sq. in. of projected area, respectively, which are excessive values.

28. Tests 7 and 8 were with all the belts off and the line-shaft journals consequently sustaining only the weight of the shaft and pulleys and the pull of the one driving belt. The great discrepancy between a saving of 21 per cent and 65 per cent for apparently similar conditions needs explanation. Test 7 was made with a small 1-h.p motor; for test 8 the same 10-h.p. motor used for the other tests was

employed. On subsequent examination it was found that the small motor bearings were badly in need of oil and were quite hot. A no-load reading of this motor showed 250 watts, which dropped to 100 watts after oiling, a difference of 0.15 kw. Deducting this from the readings of 0.268 and 0.211 gives 0.118 and 0.061, the latter representing a saving of 52 per cent which compares reasonably well with test 8.

DERIVATION OF CONSTANTS FOR USE IN ESTIMATING LOSSES

29 While the conditions of loading in this series of tests certainly include those of general practice and it may thus be safely inferred that the savings here shown may be generally realized, it is still desirable to derive constants that may be applied to any set of conditions.

30 The losses incurred are: Line-shaft journal friction; countershaft journal friction; belt slip and resistance to bending; belt and pulley windage. The last two may be safely neglected as not being a serious percentage of the total power losses under the average shop conditions although they may become a serious percentage under very light loads.

31 For good ball bearings the coefficient of friction is known to be close to 0.0015. For plain bearings the coefficient of friction may be taken at an average value of 0.03 under good conditions. For plain countershaft bearings the coefficient of friction may also be taken at an average value of 0.03 under good conditions.

32 Under the conditions of this test the countershaft bearings were replaced by the hubs of loose pulleys on the swings. With the very primitive oiling conditions and the rather high pressures the coefficient of friction here may be safely taken as high as 0.08.

Let L = load in pounds.
d = shaft diameter in inches.
S = shaft speed in r.p.m.
$\mu_p = 0.03$ = coefficient of friction for plain ring-oiling bearings.
$\mu_l = 0.08$ = coefficient of friction for loose-pulley bearings.
$\mu_b = 0.0015$ coefficient of friction for ball bearings.
kw. = power consumed in kilowatts.

$$\text{kw.} = \frac{0.746\,\pi\, d L S \mu}{12 \times 33000}$$
$$= 0.000{,}0059\, L d s \mu$$

and for $d = 2\frac{7}{16}$ in., $S = 214$; kw. $= 0.00308\ L\mu$.

33 This works out for the various total loads (Table 4) of the six tests:

LINE-SHAFT LOSSES IN KILOWATTS

Load in pounds	2350	2762	3262	3793	4377	4977
Plain Bearings, kw	0.217	0.255	0.301	0.350	0.405	0.460
Ball Bearings, kw	0.011	0.013	0.015	0.018	0.020	0.023

34 The loose pulleys on the countershafts had various diameters and speeds, given in Fig. 2. The sum of the products of these corresponding diameters and speeds is 3060. For this, kw. = 0.018 $L\mu$. This works out for the various average loads (Table 5) of the six tests:

COUNTERSHAFT LOSSES IN KILOWATTS

Loads in pounds	111.3	194.9	277.6	361.2	444.7	528.1
Countershaft, kw	0.160	0.280	0.400	0.520	0.640	0.760

35 Adding these countershaft losses to the plain and then to the ball bearing losses above gives

TOTAL LOSSES

Plain Bearings, kw	0.377	0.535	0.701	0.870	1.045	1.220
Ball Bearings, kw	0.171	0.293	0.415	0.538	0.660	0.783

COMPARISON OF ACTUAL AND CALCULATED LOSSES

36 In order to make convenient comparisons of these calculated losses with those found by measurement they are tabulated and compared as follows:

TOTAL LINE AND COUNTERSHAFT POWER SAVINGS COMPARED

Tests	1	2	3	4	5	6
Plain bearings, calculated, kw	0.377	0.535	0.701	0.870	1.045	1.220
Ball " " "	0.171	0.293	0.415	0.538	0.660	0.783
Calculated savings due to ball bearings, kw	0.206	0.242	0.286	0.332	0.385	0.437
Savings expressed as per cent	55	45	41	38	37	36
Plain bearings, measured kw.	0.466	0.660	0.934	1.117	1.154	1.301
Ball " " "	0.402	0.467	0.598	0.722	0.869	0.974
Measured savings due to ball bearings kw	0.064	0.193	0.336	0.395	0.285	0.327
Savings expressed as per cent	14	29	36	35	25	25

37 A comparison of the calculated per cent of saving with the measured per cent of saving as given in the preceding table shows a

fair correspondence in tests 3 and 4 but a considerable divergence for tests 1, 2. 5, 6. Now 1 and 2 are for very light loads and the difference may probably be accounted for as due to the neglected belt resistances and windage. As these are constant and probably independent of the load, they are a large factor for light loads and less so for heavier loads.

38 Tests 5 and 6 showed abnormal losses in the countershafts, accounted for by serious overheating of the loose-pulley hubs.

CONCLUSIONS FROM TESTS MADE UNDER NORMAL BELT CONDITIONS

39 Fortunately tests 3 and 4 were made under conditions of normal belt tensions of 44 and 57 lb. per inch width of single belt and so indicate the following:

a Savings due to the substitution of ball bearings for plain bearings on line shafts may be safely calculated by using 0.0015 as the coefficient of ball-bearing friction, 0.03 as the coefficient of line-shaft friction, and 0.08 as the coefficient of countershaft friction.

b When the belts from line shaft to countershaft pull all in one direction and nearly horizontally the saving due to the substitution of ball bearings for plain bearings on the line shaft may be safely taken as 35 per cent of the bearing friction.

c When ball bearings are used also on the countershafts the savings will be correspondingly greater and may amount to 70 per cent or more of the bearing friction.

d These percentages of savings are percentages of the friction work lost in the plain bearings; they are not percentages of the total power transmitted. The latter percentage will depend upon the ratio of the total power transmitted to that absorbed in the line and countershafts.

e The power consumed in the plain line and countershafts varies, as is well known, from 10 to 60 per cent in different industries and shops. The substitution of ball bearings for plain bearings on the line shaft only, under conditions of paragraph *a* will thus result in savings of total power of $35 \times 0.10 = 3.5$ per cent to $35 \times 0.60 = 21$ per cent. By using ball bearings on the countershafts also, the saving of total power will be from $70 \times 0.10 = 7$ per cent to $70 \times 0.60 = 42$ per cent.

EXPENDITURE REQUIRED TO EFFECT POWER SAVING

40 While power saving is of interest and desirable the man responsible for the earning of dividends will want to know what it costs to bring about such power saving and what the investment involved will pay.

41 A reference to the bearing cost of this test will give the answer.

Ten $2\frac{7}{16}$-in. by 16-in. drop ball-bearing hangers, complete cost........		$212.60
Ten $2\frac{7}{16}$-in. by 16-in drop ring-oiling hangers, complete cost..........		53.60
Extra investment		$159.00
Value of saving of 0.395 kw. at 3 cents per kw-hr. for 3000 hr. per year..........		$35.50
(Conditions of test No. 4 representing average)		
This saving represents on the extra investment................22 per cent		
A closer calculation, taking into account all of the elements, shows a still better result:		
First cost, plain bearing installation, $53.60		
Depreciation at 20 years..........................	$2.68	
Maintenance Oil; ½ pt. per day at 20 cents per gal............	3.75	
Labor, 2 hr. per week at 20 cents..........................	20.80	
Total		$25.23
First cost ball bearing installation, $212.60		
Depreciation at 20 years..........................	10.13	
4 per cent interest on first cost difference...................	5.15	
Maintenance:		
Oil, 1 gal. per year..................................	.20	
Labor, 5 hr. once per year............................	1.00	
Total		16.48
Difference		$ 8.75
Value of power saving of 0.395 kw. at 3 cents per kw-hr. for 3000 hr....		35.50
Annual saving total		$44.25
Annual saving as return on extra investment of $159.00 = 28 per cent		

DISCUSSION

T. F. Saltfr. It has long been conceded that appreciable power economies were to be secured through the use of ball or or roller bearings in place of plain bearings. The following cases show the economy obtained by the use of roller bearings.

2 A Pennsylvania shoe manufacturer, with an electrically-driven shop, found himself compelled to add considerable new equipment in departments where the motors used were already overloaded. He concluded that new and large motors were necessary, but before taking action, he consulted engineers who after investigation recommended that roller-bearing hanger boxes be purchased and the old motor equipment retained. One department required 68 h.p., with babbitted boxes. The application of steel roller-bearing hanger boxes reduced the power consumption to 50 h.p., a saving of 18 h.p., or nearly 24.5 per cent, and enabled the old motors to drive the new equipment, with a small reserve for additional equipment.

3 A Baltimore belting company had a 4 $\frac{7}{16}$-in. bearing which gave a great deal of trouble through overheating. Oil bath and water jackets were tried with more or less success. A roller bearing was tried, proved successful, and forty additional bearings of various sizes were installed.

4 A wire company of Worcester, Mass., equipped their entire plant with roller bearings and have reported a 65 per cent reduction of friction load.

5 A friction disc transmission was designed by a New Jersey corporation, the requirements being that the driven shaft revolve at a constant speed. The driving shaft was subject to slight variations in speed which were to be compensated for by automatically moving the friction wheel across the face of the friction disc. The driven shaft was thus required to move laterally about 1½ in., and to rotate at 500 r.p.m. Plain bearings with sight-feed lubrication could not be used because of their resistance to lateral motion. A special ball bearing was designed to permit a free radial and lateral movement of the shaft, resulting in an extremely sensitive and satisfactory device.

6 Roller thrust bearings are widely used wherever a thrust load or pressure parallel to the axis of a shaft is to be carried. Practically any combination of load and speed can be provided for. Nearly three years ago a bearing of this type was built for a Pittsburg steel company to operate under a pressure of 1,500,000 lb. at 100 r.p.m. As a matter of fact it carried 1,477,650 lb., applied by hydraulic pressure of 1200 lb. per sq. in. on a 32-in. piston. There was recently delivered to the same company a set of bearings the specifications of which required that they be capable of carrying 2,000,000 lb. or 1000 tons at 100 r.p.m.

7 These bearings have been applied with signal success on apparatus such as vertical hydro-electric generators, synchronous converters,

frequency changers, etc., and for this work are rapidly displacing the high-pressure oil thrusts. The advantages of roller bearings are practical indestructibility, and economy of floor space (doing away with pressure pumps, accumulator, and a mass of piping required with pressure thrust); they require little attention.

8 On an installation such as a hydroelectric generating unit, it is difficult to carry on tests which would indicate by electrical instrument reading the efficiency of thrust bearings. This is due to a number of losses, the values of which it is almost impossible to determine; for instance, the loss in guide bearings, windage, variation in load on thrust bearing occasioned by fluctuations of gate openings, etc. Laboratory tests have enabled the manufacturer to be reasonably sure of the possible efficiencies which could be secured. Data obtained in this way are not as acceptable to engineers in general, however, as results obtained through actual practice.

9 Believing that calculations could be made which would closely indicate the efficiency of this type of bearing, tests were made in which the rate of flow of the oil, the temperature of the oil, and the revolutions per minute of the bearing, were carefully recorded. The load was estimated and might have varied, thus affecting results. Two machines were tested, each test lasting about a week. Readings were taken at intervals of ten minutes.

10 The bearings tested carried an estimated load of 140,000 lb., at a speed of 250 r.p.m.; the temperature rise was 50 deg. fahr.; the flow of oil was 11¼ quarts (18.8 lb.) per min. From the data obtained the coefficient of friction was calculated to be 0.0016 or 0.16 of 1 per cent.

11 In the tests referred to, the heat loss, due to radiation from the oil casing of the bearing, was calculated to be 2 per cent of the total heat generated. Another test was made later with the oil casing jacketed with asbestos and the results showed a difference of 2.74 per cent.

12 These figures may be somewhat low; laboratory tests indicate that they are. I believe, however, that with a bearing of this type designed to meet the conditions of load and speed under which it is to operate, a coefficient of friction of less than 0.0025 can be obtained readily.

C. A. Graves. In tests made on something over two hundred different line shafts in various industries, I have found that a unit termed "watts per bearing" is best suited to making comparisons. This unit was obtained as follows:

2 Tests were made, stopping all the machines connected to the

shafting and measuring the power required to run the motor and shafting. The main motor belt was then taken off and the power required to run the motor free was found. The hanger bearings were counted and also the loose pulleys over which belts were passing. The difference in power, measured in watts between the shafting running free and the motor running free, was divided by the number of hanger and loose-pulley bearings.

3 It developed that, on the average, loose pulleys and the hanger bearing of about the same size took approximately the same amount of power, so that the sum of the loose pulleys and hanger bearings was called the "bearings." These tests were tabulated, first, by class of industry or business, and then according to the size of the shaft. For instance, in fifty tests in machine shops, with speeds ranging from 150 to 300 r.p.m., the average power absorbed by the shaft is 49 watts per bearing. Other tests gave results shown in the table.

No. of Tests Made	Size Shaft In.	r. p. m.	Power Consumed Average Watts per Bearing
43	1	400	27.1
21	$1\frac{1}{2}$	320–400	66.8
38	2	190–400	99.1
4	$2\frac{1}{2}$	200–250	108

One-inch shaft means $\frac{7}{8}$ in. or $1\frac{1}{16}$ in.

4 We were fortunate in having eight different shafts equipped with roller bearings and loose pulleys. It was found that with the shafts running from 108 to 300 r.p.m., 22 watts per bearing were required, with roller bearings on a 2-in. shaft. Taking the author's figures of tests, 3 *A* would give 5.25 watts per bearing, while 4 *A* would give 62.0 watts per bearing.

5 The author might have mentioned an additional saving obtained by using ball bearings, as smaller motors may be used to drive the shaft, thus reducing the fixed charges.

C. J. H. Woodbury. Without questioning the general conclusions of the author, I wish to inquire if the three per cent coefficient of friction referred to in Par. 31 was derived from his experiments or from other sources. The friction of a lubricated bearing varies according to the temperature of the bearing and the pressure upon it. Different oils also give different results. With light pressures, the viscosity of the oil plays a large part, so much so that if the film of oil is thick, the internal resistance from the fluid friction among the particles of this oil constitutes a large element.

2 Under heavy pressures the film of oil becomes thinner, the resistance due to its internal viscosity becomes diminished and the frictional resistance of the whole bearing approaches a direct ratio of the pressure upon it. In other words, the coefficient of friction becomes very nearly constant and slightly diminishes with increased pressure as long as the lubrication is sufficient to prevent the material of the two surfaces from coming into contact with each other, after which the frictional coefficient increases, although it may not reach the conditions of a hot bearing.

WALTER FERRIS. The coefficient of friction of railway journals is extremely low. Without being sure of the accuracy of the statement, I believe it is nearly always below one-half of one per cent, and approaches one-quarter of one per cent. Under these circumstances, granting for the moment the correctness of the statement, the saving of friction due to the ball and roller bearings would have to be balanced carefully against additional complication, first cost, and delay in making repairs.

FRED J. MILLER. The author has given no description or drawings of the bearings. The language of the paper will apply quite generally to ball bearings, whereas I understand that the test was made with specific ball bearings which had been in use for five years. I think we should have all the specific information about these bearings—including drawings—that the author is inclined to give, and a statement of the degree of refinement necessary in the making of the bearings in order to get these results.[1]

ARTHUR C. JACKSON. An advantage of ball bearings over plain bearings is that the speed of the shaft can be decidedly increased, permitting a reduction in the weight of the shaft and the driving pulleys, and reducing windage and other losses. The smaller driving pulleys will give an increased arc of contact for the belt on the driven pulley. In my experience in driving high-speed machinery, increasing the speed of the line shaft, which can be accomplished by the use of ball bearings, has a distinct advantage.

[1]This information is given by the author in his closure and in a paper in The Journal of the Society for May 1910.—EDITOR.

Chas. D. Parker. The value of the ball bearing or roller bearing seems to be conceded in a general way, but its application immediately brings up the question of excessive cost, so that it is hardly considered in many cases. Data of the sort given in the paper should be highly valuable as giving confidence to engineers in recommending the use of ball bearings on a large scale, even though the cost may be high. The question cannot be decided by a single experiment. Several experiments, including tests on a shaft 400 or 500 ft. in length, would be even more valuable, especially if made on bearings that have had a few years' service under ordinary care.

2 It might be of interest to know whether the apparently high cost of ball and roller bearings is due to the high cost of manufacture or to large selling expense, which we may expect to be reduced with a more general demand for the goods.

3 With the general introduction of electric-motor drive, the belt drive from line shafts has become somewhat old-fashioned. However, as the motors have large factors of inefficiency, if the efficiency of the line-shaft belt drive can be greatly improved by the use of ball bearings, it would be of interest to know to what extent this can be done. It would probably be shown that the older method is still the more economical method in a great many instances.

Oliver B. Zimmerman. I would like to ask Mr. Hess if he has considered the application of ball bearings to countershafts which do not run the same proportion of time as the line shaft. What would be the relative return on the investment in that case, as compared with the line shaft itself? Furthermore, would it be advisable to lengthen the line shaft when the ball bearings are used; for instance, in group driving, would it be an advantage to use a line shaft 90 ft. or 100 ft. in length, as compared with a group of machines driven from 60 ft. of line shafting?

W. F. Parish, Jr. Mr. Hess's paper brings out an important point usually overlooked in comparative tests requiring great accuracy, namely, the influence of temperature and relative humidity on the power delivered, by causing variations in belt tension.

2 For comparative tests made under workshop conditions it is advisable to have the belts made up half of cotton and half of leather, thereby eliminating the effect of humidity, which may cause variations of 12 per cent in the power delivered.

3 An English firm five years ago purchased a cotton belt to drive a

dynamo, but this belt was not equal to the speed and power required of it, so a leather belt was substituted. It was decided to use the cotton belt on one of the main mill drives, but it was found to be much too short. So a piece of leather belt was spliced in, the whole being, when finished, half leather and half cotton. A casing was built under it, as it was low down and in a dangerous position. The manager was annoyed to find that this casing had been built too close to the belt, no allowance being made for sagging.

4 The dampness greatly affected the leather belt, as the drive was in a low part of the mill, but the casing under the patched belt was never altered. The length of this belt never varies whether the weather is damp or dry and it is the best belt drive in the mill for steady work. Moisture has an opposite effect on leather and cotton, leather lengthening and cotton contracting with an increase of humidity, so that in the half-cotton and half-leather belt the weather effect is practically compensated for.

5 In tests 3 and 4, the average saving of power by using ball bearings instead of ring-oiling bearings is 36 and 35 per cent, respectively, which is unusually good. It would be interesting to know what oil was used in the ring-oiling bearings during these tests and if the oil was new or old. With a very poor oil in the ring-oiling bearings the saving in power may be only partially caused by the change to ball bearings.

6 Oil and lubrication play a very important part in the economical distribution of power. Many power tests which I have made show that when very poor and cheap oil is used, a saving as high as 40 per cent can be obtained simply by using a better oil. Forty-two comparative power tests, made in small workshops or sections of large shops, show an average saving in power of 19 per cent, due to the use of a good and suitable oil. By using a good oil there will be but little increase in cost, as it can be used sparingly, so that the yearly cost for the better oil may be even less than for the poor oil. One test on a machine gear-driven by an electric motor showed a power saving of 55.5 per cent by using a good oil instead of a poor oil and grease.

Geo. N. Van Derhoef. In the results of the tests summarized in Par. 41 of Mr. Hess's paper, the quantity of oil required to maintain ten $2\frac{15}{16}$-in. bearings is given as $\frac{1}{2}$ pint a day, or 150 pints per year, which is equal to $18\frac{3}{4}$ gal. There is probably no make of self-oiling hanger on the market today that requires anything like this quantity of oil to maintain it. Three gallons a year for ten hangers would be ample allowance for even the poorest make.

2 The item of labor charged is two hours a week, which is also excessive even if the enormous quantity of oil specified were used. As a matter of fact, three or four hours a year should be ample time to devote to the care and attention of ten $2\frac{7}{16}$-in. hanger boxes.

3 The allowance of twenty years for depreciation would seem fair for babbitted bearings, as probably all of us know of bearings running in daily service for a longer period. I would like to know if Mr. Hess has any figures showing ball bearings on line-shaft service for anything like this period. As I look at the matter—and I think others will agree with me—it is not so much a matter of a lower coefficient of friction as it is of the "staying properties" under practical conditions, as distinguished from a test experiment extending over a brief interval of time.

The Author. Taking up the various points raised and the questions asked during the discussion, the author wishes to reply as follows:

2 *Percentage of Saving and Actual Saving.* A saving of power cannot be intelligently considered as a percentage of the entire driving power without full knowledge of the entire conditions. A given actual saving may be one per cent or ninety-nine per cent of a total. The saving in line-shaft journals when referred to the line-shaft loss is one ratio, and when referred to the total power consumption, is quite another ratio. So far as I am aware the literature on the subject quite generally refers to the line-shaft friction as a percentage of the total power consumption. That is misleading, since the percentage ranges from only sixteen or so in some textile mills to seventy or more in some of the rougher machine industries. In all probability the actual friction loss, bearing for bearing, does not vary in anything like so great a degree as sixteen to seventy per cent. The thing that is of real importance is not the ratio of the saving to a given whole, but the actual value of the actual saving.

3 *Estimating Power Losses and Savings.* Mr. Graves suggested that the power consumption of a bearing might be stated from experience in "watts per bearing." Such an expression would be convenient if it could be correctly applied; but the watt loss depends upon the coefficient of friction, the load and the surface speed. The coefficient of friction for a given type of bearing may be said to be fairly well known, or at least not to vary between very wide limits. That may also be said of the load; but the surface speed is made up of the shaft diameter, or rather the circumference, and the angular speed, both varying between very wide limits. So general an expression is therefore hardly possible, nor is it necessary.

4 For any given installation, the shaft diameter and speeds are known; the loads are due to the definitely determinable weight of the shaft, pulleys and belts, and to the belt pull, the last-named of which should not be allowed to exceed 60 lb. per inch width of single belt, while it certainly will rarely fall below 40 lb. The coefficient of friction for plain bearings may range from 2 to 8 per cent, with 3 per cent a very fair and general value, and $\frac{1}{8}$ per cent for ball bearings. A rise to $\frac{1}{4}$ per cent for ball bearings would indicate a poor quality of bearing.

5 An actual calculation, using the known constants of the installation in question, will always give closer results than the use of any general expression, necessarily much less accurate, such as "watts per bearing." In Par. 32 the expression for kilowatts is given as

$$Kw = 0.000,0059\ L\ d\ s\ \mu$$

or

$$\text{watts} = w = 0.000,000,0059\ L\ d\ s\ \mu$$

which may readily be converted to the convenient form

$$K w_y = \text{watts per bearing for year of 3000 hours}$$
$$K w_y = 0.000,001\ L\ d\ s\ \mu$$

6 Mr. Graves has found the "watts per bearing" to range from 27.1 to 108 in 106 tests of plain bearings. The measured losses of the test cited in the paper are under average conditions of belt pull. For the usual belt load, tests 3 and 4 show for the ten plain bearings (see table in Par. 33) losses in kilowatts of 0.350 and 0.405, and for the ball bearings 0.018 and 0.020, or in watts per plain bearing 30 and 35, and for ball bearings 15 and 18.

7 Mr. Graves' four tests of a $2\frac{7}{16}$-in. line-shaft at 200 to 250 r.p.m. may be fairly compared with the author's tests of a $2\frac{7}{16}$-in. line-shaft at 214 r.p.m; Mr. Graves' result of 108 watts per bearing, as against the author's of 30 to 35, shows how unsafe a general wattage figure is. Changing the coefficient of friction from the 3 per cent found to be approximately correct for the test cited, to 10 per cent, would raise the 30 watts per bearing to Mr. Graves' 108 watts per bearing. In reality the tests cited by Mr. Graves are confirmatory of the author's, since the former range from 27 to 108, proving that the author's values of 30 to 35 for correct belt loads and 22 to 46 for extremely light and extremely heavy loads, represent an average of good practice.

8 *Indirect Economies.* Mr. Graves has suggested that the mounting of line shafts on ball bearings will reduce the sizes of the

motors required to drive the shafts. While that is obvious, the consequent economy is greater than is at first apparent. A motor must always be selected of sufficient size to perform its work safely. As the frictional resistance of a plain-bearing line-shaft is apt to vary between very wide limits—27 to 108 watts per bearing, according to Mr. Graves' tests—the motor must necessarily be selected to cover nearly the maximum safely. That means a rather large motor compared with the average useful plus friction load. Not only is there thus an unnecessary increase of first cost of the motor but, more seriously, the operating cost is unduly enhanced, as it is well known that a motor operating much below its rated capacity has low efficiency and is wasteful of current. When, on the other hand, the line shaft is mounted on ball bearings, the friction load is greatly reduced, its amount is more definitely determinable beforehand, and the initially smaller motor is used nearer to its point of maximum efficiency.

9 Mr. Jackson refers to a possible increase in shaft speed due to mounting the shaft on ball bearings, resulting in decreased weight of shaft pulleys and belts and more favorable belt contacts. All of these elements in time make for decreased bearing loads and consequently still further increases in economy. Mr. Jackson has had under his continual observation during several years a number of plain and ball-bearing line shafts of medium and high speeds, and so speaks not merely from theoretical reasoning, but from actual practice and observation.

10 *Reduced Importance of Improper Belt or Rope Tension.* The great variations in belt tensions that may be brought about by weather and temperature conditions, moist and dry atmosphere, etc., have been referred to by Mr. Parish. Both leather and cotton belts, as well as fibre ropes, are subject to considerable variations from these conditions. Possibly fully as important a factor is the average millwright or mechanic. The properly stressed belt is the exception. Most belts are tightened almost to the breaking point. The work thus lost in friction in plain bearings is directly proportional to a coefficient of friction ranging from 3 to 10 per cent for those conditions; but with the low coefficient of friction of $\frac{1}{8}$ to $\frac{1}{4}$ per cent for ball bearings a relatively enormous over-stressing of the belt has comparatively little influence in increasing the journal friction losses.

11 The ball bearing is a most important factor in belt economy, since the absence of the plain bearing friction load permits the use of slack belts and makes for greatly increased belt life. Mr. Fred. W.

Taylor showed the consequent economy most conclusively in his paper, Notes on Belting.[1]

12 *Relative Efficiency of Direct Motor Drive and Ball-Bearing Line Shafts.* Mr. Parker refers to the large factor of inefficiency of motors and inquires concerning the possible improvement in line-shaft belt drives due to the use of ball bearings. While in the early days of the introduction of direct-driven tools much was expected from the saving due to cutting out the line-shaft friction, it soon developed that the need for using motors equal to the maximum demand of a tool brought in greater power losses because of such motors working on an average at points of low efficiency.

13 Unless the direct application of the motor results in greater convenience of handling the machine to produce a greater output, the direct drive is not justified. In that case, the mounting of countershaft, loose pulleys and line shaft on ball bearings will result in very considerable power savings. The tests made for the author by Messrs. Dodge & Day on line shafts showed savings of 35 per cent under average conditions; extended to the countershaft and loose pulleys the savings will readily amount to more than half of the total power consumption.

14 In line with this general question Mr. Zimmerman asks whether it would be advantageous to lengthen a group-drive line shaft to 60 ft. to take a larger group involving a shaft length of 100 ft. Unquestionably that will be economical so long as other considerations than those of line-shaft and line-shaft-motor losses do not govern. As to the relative losses in countershaft and line shaft, it may be said in general that they will be fairly equal. It is true that the countershaft does not run as continuously as does the line shaft, but that simply involves a transfer of the loss from the countershaft hanger to the loose pulleys; only when the belt is actually thrown off does this loss cease; if the loose-pulley diameter is decreased, as it should be to decrease the belt tension, the loss is lessened.

15 *Ball vs. Roller Bearing.* Mr. Graves makes inquiry concerning the relative values of ball and roller bearings and their coefficients of friction. The coefficient of friction for good ball bearings has already been given as close to $\frac{1}{8}$ per cent; for roller bearings the friction is about double, assuming always that the rollers are kept in alignment and that hard and true rollers rolling on true and hardened surfaces are used. The real advantage of the ball bearing is not

[1]Trans. Vol. 15, p. 204.[2]

the difference in friction, but its endurance and the consequent permanence of the power saving. As the correct ball bearing employs only a single row of balls it has no length; that at once cuts out all disturbances, due to deflections of shafts or housings, that seriously affect rollers. The readiness with which the ball bearing is housed to retain its lubricant and to keep out injurious grit, as well as the small space occupied, are also advantages peculiar to it alone. The coefficients of friction cited have been determined by oft-repeated tests. They are referred to the shaft diameter so that the values are directly comparable with those of plain journals.

16 *Reasons for Ball Bearing Cost.* Mr. Parker wishes to know whether the apparently high cost of ball bearings is due to the high cost of manufacture or to large selling expense. Concerning the latter it may be said that the expense of selling ball bearings is not at all high; it is, in fact, lower than in many other lines of high-grade precision machine elements. The cost resides in the absolute necessity for precision, and the character of manufacture. Ball bearings can fitly be compared only with high-grade tools of high-grade steels. The material is a special alloy steel, relatively high in carbon, manganese, chrome and silicon; this is a combination that is very refractory under the cutting tool. After hardening, rough and finish grinding cannot be forced, as that spoils the integrity of the rolling surface. Accuracy of a high degree is essential; the unit of measurement is the ten-thousandth part of an inch. Interchangeability of a high order is not to be secured cheaply.

17 The data showing the saving in power consumption, not in percentage, but in actual consumption, that Mr. Parker asks for, are given in the body of the paper in the table in Par. 36, on lines marked "Plain Bearings measured kw." and "Ball Bearings measured kw."

18 *Ball Bearings on Railways.* This use of ball bearings is outside of the subject matter of the paper, but as inquiry has been made by both Mr. Ferris and Mr. Graves it may be noted that ball bearings of the same type are in regular use for main-line railways and electric railways, on the axles in the former and for both axles and motors in the latter. On the Prussian-Hessian state railways the first of these bearings are still in use, and as the result of somewhat over 400,000 kilometers' run (250,000 miles) under standard passenger coaches, show no evidences of wear.

19 In Europe, as well as in the United States, careful comparative measurements, extending over many weeks of 2-min. observations, have shown savings in electric railway power consumption of over ten

per cent, with incidental decrease in motor temperature. For main-line and electric railway service the direct power saving is of less importance than the ability to take advantage of coasting; this saving may frequently rise to 37 per cent. The chief economy lies not in power saving, but in saving of lubricant, attendance, cost of renewals and, in electric railway operation, the keeping of the equipment more in service, and less in the repair shop for renewing bearing linings and rewinding armatures that worn plain bearings have allowed to sag onto the polepieces.

20 *Type of Bearing Under Discussion.* The author purposely confined the paper to a report of results of tests made for him

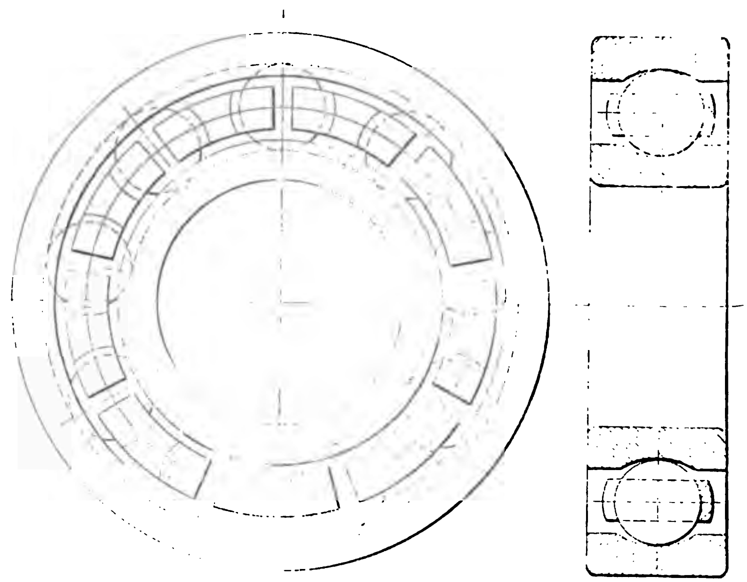


Fig. 1 Elevation and Cross Section of the Hess-Bright Ball Bearin

by Messrs. Dodge & Day, preferring to bring out the engineerin value and economic value to be expected of correctly made, correctl selected and properly mounted ball bearings. As Mr. Miller has aske for information concerning the specific ball bearing involved in the test it is proper to say that it is known in the United States as the Hess-Bright or DWF, and in Europe generally as the DWF.

21 Fig. 1 illustrates the ball bearing proper, in cross section and in elevation. It will be seen to consist of an inner race, an outer race, a

series of balls, all of special steels hardened throughout, and a cage or separator for the balls. The ball tracks have curvatures approximating the ball outline, the inner track very closely, the outer track slightly less so. The contact between balls and tracks is on a plane at right angles to the axis of the shaft, thus providing only one point of contact of the ball with each track. The sides of the races are continuous, with no interruption at any point for filling in the balls; that ensures absolutely smooth rolling of the balls and the absence of any possible contact with the edges of filling openings. In lieu of side interruptions or filling openings for the balls, assembly is by eccentric displacement of the two races, filling in balls through the wider space at one side, bringing the races into concentric relation, spreading the balls evenly and retaining them in proper position by the separator.

22 As to the refinement necessary in the making of these bearings, to which Mr. Miller kindly refers from his own observations, I would say that balls must be true to shape and to size within a limit of 0.0001 in. The bearing bore is held within a tolerance of 0.0002 in. +, and 0.0004 in. −. The outside diameter is held within 0.0006 in. +, and 0.0012 in. −, according to size. The width is held within 0.02 in. −. Each finished bearing is gaged for trueness of rotation with reference to the bore, and for trueness of the outer race on the ball circle. Each race is tested for uniformity of hardness, referred to a standard, at four points on each side, or eight per race; the sclerescope is used for this purpose, and that in turn is occasionally checked by the Brinnell, as well as the Turner and the Howe hardness test apparatus.

23 Lest it may appear that these refinements are not necessary, it may be well to say that the knowledge of their necessity has been acquired at great cost; also that only to the most painstaking care in material, treatment and workmanship is the success of the ball bearing due as an every-day reliable element of mechanism. A knowledge of proper proportions for various conditions of load, speed, shock, etc., is, of course, also essential.

No. 1266

PUMP VALVES AND VALVE AREAS

By A. F. Nagle, South Bethlehem, Pa.

Member of the Society

There has grown up a custom of requiring in waterworks pumping engine specifications that the area through the valve shall exceed the area of the plunger by a certain amount, varying from 25 to 125 per cent. The probable intent of this clause is to obtain a low velocity through the valve and consequent low loss of head, and it is my purpose to demonstrate that this condition is not realized in practice.

2 The above form of expression, namely, proportioning the valve area to the plunger area, is defective because (*a*) it fails to distinguish between the valve-seat area and the circumferential area of the valve at an assumed or specified lift; (*b*) it leads to an absurdity unless coupled with the length of stroke and the number thereof.

3 To the first criticism it may be replied that the engine builder interprets the clause to mean the net area through the valve seats, but the city's engineer occasionally requires the circumferential area. To this the builder will not seriously object, for he simply increases the *possible* lift, knowing very well that "it will never go there."

4 The second criticism can be best illustrated by the following example: Compare two pumps, each making 25 r.p.m., one having a plunger 6½ in. in diameter by 60-in. stroke, and the other a plunger of 13-in. diameter by 15-in. stroke. Precisely the same volume of water passes through the two pumps, yet the rule laid down in the specification would require for one pump four times the valve area of the other.

5 What is the real purpose in specifying anything at all about valve area? Evidently the same that is sought in limiting the plunger travel per minute, and founded upon the law that in a pumping engine low velocities of water are conducive to low cost of operation but proportionately great cost of construction, and conversely,

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high velocities imply high cost of operation but lower cost of con struction; hence, very properly, the buyer should specify the maximum velocity he will accept.

6 Briefly, it may be said that city waterworks engines are now quite generally limited to about 250 ft. of plunger travel per min., although frequent attempts are made with special designs to increase this travel to nearly twice this amount. This plunger travel with 5–to–1 connecting rod entails a maximum plunger velocity of 6.67 ft. per sec., and the *head due to this velocity* is 0.30 lb. per sq. in. It is desirable to speak of the maximum plunger velocity rather than the mean, because that governs the maximum valve area to be provided.

7 The fluid losses within a pump may be divided as follows:

- *a* Velocity head due to plunger velocity, varying from zero to the maximum above cited. This loss may be ignored however, since with well-rounded plunger ends and rounded water passages, the accelerating head of the fluid column during the first half of the stroke is conserved by its retarding force during the second half.
- *b* Friction head due to surface contact. As the main parts of a pump are comparatively large, the velocities are low; and the lengths of contact being short, this friction-head is equal to a velocity-head for only about 50 diameter lengths, and becomes so small as to be negligible.
- *c* Velocity head through the valves. This, whatever its amount, is a total loss because the energy of the issuing streams is destroyed in eddies as it enters the large valve or pump chambers. To keep this head low is the purpose of the specification that the valve area shall exceed the plunger area by a certain amount.

8 Let us assume that valve area means valve-seat area, and pass on to the study of the valve. A pump valve consists essentially of three elements: (*a*) a fixed seat, (*b*) a movable valve, (*c*) a spring. The most important of these is the spring, and yet on this point all specifications are silent. Is this because the writer of the specification knows nothing about the subject? In a general way, it is obvious that a spring may be so stiff that on the suction stroke, where only atmospheric pressure is available, the valve will not open at all; or it may be so light that it will nearly float in its place and will close only with the return stroke of the plunger. Between these two extremes, is there not an ascertainable strength of spring which will

allow the valve to close promptly without shock and yet require for lifting force but a small percentage of the total pressure in the pump? So far as I am aware, this problem has not been stated and solved in any publication, but is left for each pump builder and user.

9 The pressure of the spring per square inch of the inside seat area seems to me to be the force that causes the rate of flow of the water through the valve. In my experiments of 1875 (see Vol. 10 of Transactions) with a Cornish double-beat valve, this hypothesis did not hold, that is, the velocity through the valve was from 60 to 90 per cent greater than that due to the pressure of the valve: in other words, the valve did not rise as high as theory would demand,

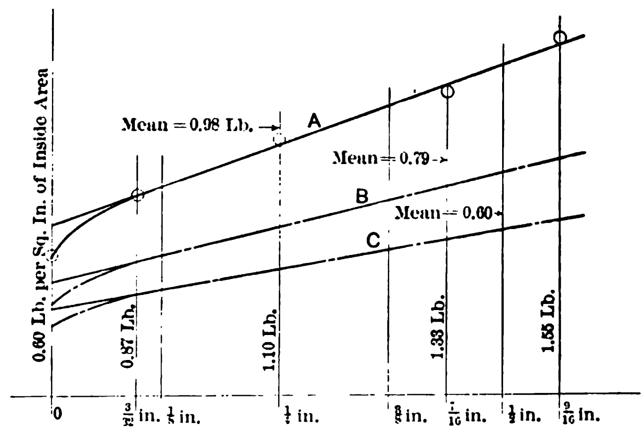

FIG. 1 DIAGRAM SHOWING VARIATIONS OF TENSION

LINE A IS THE ACTUAL TENSION OF A SPRING AT VARIOUS POINTS OF LIFT. LINES B AND C SHOW ESTIMATED TENSIONS AT DIFFERENT LIFTS WITH AN INITIAL TENSION OF 0.4 LB. AND 0.3 LB.

but I think the deviation may be attributed to the large curvature given the upper passage. In extensive experiments recently made by the Bethlehem Steel Company with a large flat-hinge, or flap, valve to be used in the Baltimore sewage pumps, the hypothesis held very well at the beginning of the lift and fell off only about 10 per cent at full lift. These experiments also confirmed the Providence experiments in that the varying lift of the valve follows closely the varying velocities of the plunger, except as it is modified by increased weight or spring tension.

10 I shall therefore assume that in a flat rubber pump valve held down by a spring: (*a*) the velocity of the water is that due to pressure

per square inch of the inside valve area, (*b*) the area of discharge is the net circumference of the inside of the seat multiplied by the lift.

11 The well-known formula for the velocity of flow in feet per second is

$$V = 8.025 \sqrt{h} \qquad [1]$$

(*a*) where *h* is the head of water in feet. As 2.31 ft. of water 1 sq. in. in area weighs 1 lb., the formula can be changed to

$$V = 12.23 \sqrt{p} \qquad [2]$$

(*b*) where *p* is the pressure of spring per square inch of inside area

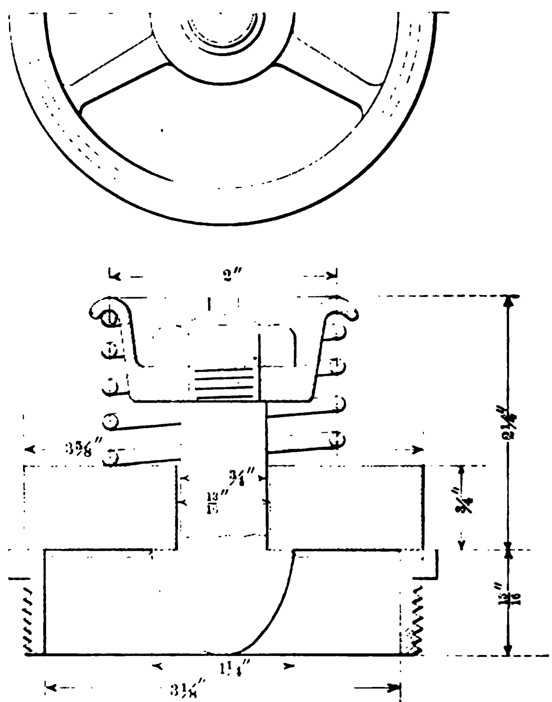

Fig. 2 Section of a Standard Make of Pump Valve

FREE LENGTH OF SPRING IS 1$\frac{9}{16}$ IN., NO. 12 B. W. G. SPRING BRASS

of valve. Tables 1 and 2, computed from Formula 2, may be convenient in studying this subject.

12 *Springs.* The springs in common use vary from 0.40 to 0.60 lb. per sq. in. of inside valve area at the beginning of the lift, and as they are comparatively short (about 1⅝ in. closed), they tighten up

TABLE 1 CONVERSION OF VELOCITY INTO PRESSURE

Velocity Ft. per Sec.	Pressure Pounds per Sq. In.	Velocity Ft. per Sec.	Pressure Pounds per Sq. In.	Velocity Ft. per Sec.	Pressure Pounds per Sq. In.
4	0.107	8	0.430	12	0.967
5	0.168	9	0.544	13	1.135
6	0.242	10	0.672	14	1.317
7	0.329	11	0.813	15	1.512

TABLE 2 CONVERSION OF PRESSURE INTO VELOCITY

Pounds Pressure	Velocity Ft. per Sec.	Pounds Pressure	Velocity Ft. per Sec.	Pounds Pressure	Velocity Ft. per Sec.	Pounds Pressure	Velocity Ft. per Sec.
0.15	4.74	0.50	8.65	0.85	11.27	1.40	14.47
0.20	5.47	0.55	9.07	0.90	11.60	1.50	14.98
0.25	6.11	0.60	9.47	0.95	11.92	1.60	15.47
0.30	6.70	0.65	9.86	1.00	12.23	1.70	15.94
0.35	7.23	0.70	10.23	1.10	12.82	1.80	16.40
0.40	7.73	0.75	10.59	1.20	13.39	1.90	16.85
0.45	8.20	0.80	10.94	1.30	13.94	2.00	17.29

TABLE 3 RATIO OF PRESSURES, VELOCITIES, AND LIFT OF VALVE, LINE A, FIG. 1

Lift Inches	Tension Pounds per Sq. In.	Velocity Ft. per Sec.
At Start	0.60	9.47
3/32	0.87	11.40
1/8	1.10	12.82
5/32	1.33	14.10
3/16	1.55	15.23

TABLE 4 RATIO OF PRESSURES, VELOCITIES AND LIFT OF VALVE, LINE B, FIG. 1

Lift Inches	Tension Pounds per Sq. In.	Velocity Ft. per Sec.
At Start	0.40	7.74
3/32	0.58	9.31
1/8	0.73	10.47
5/32	0.88	11.51
3/16	1.03	12.41

TABLE 5 RATIO OF PRESSURES, VELOCITIES AND LIFT OF VALVE, LINE C, FIG. 1

Lift Inches	Tension Pounds per Sq. In.	Velocity Ft. per Sec.
At Start	0.30	6.70
3/32	0.44	8.07
1/8	0.55	9.07
5/32	0.66	9.97
3/16	0.77	10.83

rapidly as the valve rises. Fig.1, Line A, illustrates this rate of increase taken from a new spring. The apparently needlessly stiff springs are used (*a*) to provide against the relaxation sure to occur with all bronze springs; (*b*) to allow for the lengthening of the spring as the rubber valve wears away.

13 Fig. 2 shows a standard pump valve used by a prominent builder. These valves run from 2½ in. to 3¾ in. inside diameter. Table 3 is made up from Fig. 1, with the velocities computed by Formula [2]. A larger lift than $\frac{5}{16}$ in. is not generally allowed for, as the valve is not expected to rise higher or even as high as this, and considering the increased tension of the spring one would not expect it.

14 *Other Spring Tensions.* In Fig. 1, Line A represents the actual tension of a spring at various points of lift. If the same type of spring were made of smaller wire, its varying tensions at different lifts would be proportional to the initial tension. Lines B and C show these estimated tensions at different lifts with an initial tension of 0.40 lb. and 0.30 lb., respectively; Tables 4 and 5 give the tension at these lifts, together with the velocities corresponding thereto.

15 *Valve Lift.* If it is true that the spring tension governs the velocity of water through the valve, we can readily find the lift of a valve under specified conditions. The inside net circumference multiplied by its lift and the velocity must equal the volume of water passing through the valve per second. By formula, $P \times V_m = C \times L \times v_m \times N$, where

P = plunger area in square inches.
V_m = maximum velocity of plunger in feet per second: which is 1.60 $\times$ the mean velocity.
C = net circumference of valve seat, inches.
L = lift of valve, inches.
v_m = maximum velocity of water at Lift L, found by aid of the diagram, Fig. 1.
N = number of valves.

or

$$L = \frac{P\,V_m}{C\,N\,v_m} \qquad [3]$$

Contractions at sharp corners and angular turns make this calculation inexact, but the method will be found exact enough for the comparisons in this paper, and is the only practical method in the present state of knowledge on this subject.

16 *Comparisons of Valve Areas and Springs.* To illustrate my views, let us take the case of a vertical triple-expansion crank and flywheel pumping engine, having each plunger 34 in. in diameter by 60-in. stroke, making 25 r.p.m.—practically a 25,000,000-gal. engine. Plunger travel = 4.167 ft. per sec., and maximum plunger velocity = 6.67 ft. per sec. Assume the pump valve to be 3¾ in. inside diameter with 5 ribs, leaving a net area of 8 sq. in., and a net circumference of 10.53 in. The theoretical lift of this valve, to give the same area on the circumference as through the ribs, would be 0.76 in. Let us assume a valve-seat area equal to 150 per cent of the plunger area and an initial spring tension of 0.60 lb., and ascertain the number of valves, their lift, velocities at various points and loss of efficiencies. Then let us make the same calculations for a valve-seat area equal to the plunger area, with an initial spring tension of (*a*) 0.40 lb. per sq. in.; (*b*) 0.30 lb. per sq. in. The results are given in Table 6.

TABLE 6 LOSS OF EFFICIENCIES, ETC.

Initial Spring Pressure Pounds	Valve Seat Area Per Cent	Number of Valves	Lift of Valves Inches	Velocities in Feet per Second			Loss of Efficiency Per Cent
				Plunger	Valve Seat	Valve	
1	2	3	4	5	6	7	8
0.60	150	170	¼	0–6.67	0–4.44	9.47–12.82	2.45
0.40	100	114	7/16	0–6.67	0–6.57	7.74–11.51	1.97
0.30	100	114	½	0–6.67	0–6.57	6.70–10.37	1.50

17 *Explanations.* Column 4. The plunger area (908 sq.in.) multiplied by its maximum velocity (6.67 ft. per sec.) must equal number of valves (170 or 114) multiplied by the net circumference (10.53 in.), its lift L, and the maximum velocity at its highest lift. This is a trial process, but easily found after one or two trials. Taking the first case, we would have

$$908 \times 6.67 = 170 \times 10.53 \times L \times 12.82, \text{ or } L = 0.264 \text{ in.}$$

Second case:

$$908 \times 6.67 = 114 \times 10.53 \times L \times 11.51, \text{ or } L = 0.438 \text{ in.}$$

Third case:

$$908 \times 6.67 = 114 \times 10.53 \times L \times 10.37, \text{ or } L = 0.50 \text{ in.}$$

18 Column 5. While the crank velocity may ordinarily be taken as the maximum plunger velocity with a connecting rod five times the crank, its maximum velocity is 1.019 times that of the crank or, maximum velocity = $5 \times 3.1416 \times 25 \times 1.019 \div 60 =$ 6.67 ft. per sec. We can also take the mean travel of the plunger and multiply it by 1.60 to find the maximum velocity.

19 Column 6 is self-evident.

20 Column 7. These velocities are obtained from Tables 3 to 5 and Fig. 1, and were computed in the manner already described.

21 Column 8. To get an expression for the effect of strong *vs.* light springs upon the economical working of a pump, I have assumed a pump working under a total head of 80 lb. per sq. in. and computed the *mean* pressure required to operate the valve, calling this ratio of pressures its loss of efficiency. A careful examination of the diagram of spring compression, Fig. 1, shows that at the beginning of the lift the spring did not assume its full uniform resistance. It took nearly $\frac{3}{32}$ in. of motion to tighten it uniformly. I think this is due to the fact that these single coil-wound springs are always a little stiffer on one side than the other, thus canting the valve to an oblique position to conform itself to the center line of resistance of the spring. I have taken as mean pressures of the springs slightly more than the mean of the two extreme positions, because the times during which the different pressures prevail are not equal. Mean pressures were taken as follows:

Case A, maximum tension = 1.10 lb. at $\frac{1}{4}$-in. lift, mean pressure = 0.98 lb.

Case B, maximum tension = 0.94 lb. at $\frac{7}{16}$-in. lift, mean pressure = 0.79 lb.

Case C, maximum tension = 0.72 lb. at $\frac{1}{2}$-in. lift, mean pressure = 0.60 lb.

As this mean pressure exists during both strokes, it must be multiplied by two to find its ratio to the effective head of 80 lb., operating only during one stroke. Thus the values given in Column 8 are found.

22 *Discussion and Recommendation.* A study of the figures given in Table 6 shows that the proper place to look for "loss of head" in a pump is in the spring tension, and not in the valve-seat area. As long as the maximum velocity through the seat does not exceed that through the valve, it does not add to the total loss of head. The only

reason for having a large number of pump valves and a large inside diameter is to keep the lift down, basing judgment on the number of reversals per minute.

23 We have seen that a $3\frac{3}{4}$-in. valve, having a net area of 8 sq. in., needs 0.76 in. lift to give the same area at the circumference as through the seat. If there were no rib obstruction, 25 per cent of the diameter of a circle gives the height to which a valve must lift to give a circumferential opening equal to its area. Because of the ribs, we need but 20 per cent of the diameter for the lift; and with the lightest spring C, a maximum lift of $\frac{1}{2}$ in. or 13 per cent of the diameter was sufficient to discharge the required volume of water. It will be good construction to limit the lift of a pump valve of this type to, say, 15 per cent of its internal seat diameter. The spring will not allow it to rise to that height, but it is a safe limit for a stop.

24 The place to begin the study of the proportions of a pump is at the *spring* of the valve. *Make* a sample spring of such diameter and length and strength as you may think desirable, and by *experiment* construct a diagram of its rate of compression, as in Fig. 1. Now you can find the maximum velocity at an assumed lift and proceed in the manner already pointed out. The spring would be improved, that is, it would not tighten up so rapidly when compressed, if it could be made somewhat longer than present practice, but this is not practicable, as it would enlarge the valve chamber, where the valve-cage design is used.

25 The largest and weakest castings within a pumping engine are the valve chambers and anything that can be done to reduce them to the minimum size permissible is good engineering. I think the line of study I have pursued will indicate that the total valve-seat area in this type of engine need not be more than the plunger area. That rule, if adopted, would reduce the diameter of the valve chambers an appreciable extent, probably 10 per cent, and this is well worth saving.

26 The number of valves saved by the construction recommended (about 33 per cent) is also worth while. No loss whatever would be entailed and a part of the money saved could be expended in making a better spring. I would make the spring of steel, if possible oil-tempered, and protected against corrosion by copper electroplating. Then I would have all springs tested and brought to a like tension under a rigid specification. With these improvements I believe that a little better pumping engine than we now have could be obtained at a little smaller cost.

DISCUSSION

Charles A. Hague. The practice referred to by the author, of specifying that the area through the pump valves of waterworks engines shall bear a certain relation expressed in percentage of plunger area, is becoming less frequent, and it is to be hoped that it will finally be disregarded altogether. The relation between the plunger areas is merely incidental, because the valve area is a function of the quantity of water to be handled, the important matter being the velocity of the water through the valve seats to fill the plunger chamber as nearly complete as possible under the conditions.

2 The total valve area, or total area of valve-seat opening, ought to depend upon the velocity needed to pass the required quantity of water in a given time. Some authorities advocate a velocity not to exceed 3 ft. per sec., others 4 ft. per sec. and some as low as 2½ ft. per sec. Two factors are to be considered, as follows:

3 First, as to the lift of the valves. The lower the pressure, and the lower the speed of the engine, the higher the valve may lift; on the contrary, the higher the pressure, and the higher the speed of the engine, the less the valve may lift, if a smooth, easy running, economical engine is desired.

4 Second, regarding the circumferential area of the valve space, or the area of the space around the edge of a disc valve, when it is open or off its seat. This is a factor that need not be very seriously considered, because the water, having succeeded in getting easily through the grating formed by the seat, will meet with very little resistance in moving out from under the valve. Valves free to lift to an unchecked height will often get so far away from their seats that slamming will take place at the reversal of the plunger. A pumping engine will work best when provided with sufficient valve-seat area to keep the mean velocity of the water down to about 3 ft. per sec., the lift of the valves being so restricted that they will return to the seats when the plunger approaches the end of its stroke.

5 With reference to plunger travel in conjunction with pump valve area, mentioned or implied in the paper, the vital question is, How shall we obtain any certain plunger travel per minute: by a short stroke at many revolutions per minute, or, by a long stroke at few revolutions per minute?

6 After the water is well started through the pump valves, a larger increase in speed would be permissible than is found in practice, if it were not for the reversals at the end of the strokes. The

250-ft. per min. plunger travel mentioned in the paper would be permissible with a 60-in. stroke at 25 r.p.m., or better with a 72-in. stroke at 21 r.p.m. The pump valves would work in a very satisfactory manner, the pumps would give very good hydraulic efficiency and the engines would run smoothly. But if we should attempt to obtain 250 ft. per min. with a 30-in. stroke at 50 r.p.m. there would be a great reduction in economy, smoothness of running and general efficiency.

7 The items in Par. 7 are all within the scope of mechanical efficiency, and will be reasonably well taken care of, if the valve factor is properly attended to. The most effective method for dealing with the question of valve area is to establish a certain satisfactory area per unit of pumpage, at some definite minimum rate of revolution as a standard. Then, for every revolution per minute above the standard rate, add a certain per cent to the standard valve area. This will give an engine of more revolutions, a greater proportionate valve area than a slower machine, thus in fast engines keeping the valves nearer to their seats than in slow ones.

8 In Par. 25, the author makes a statement, with which one feels compelled to take issue, that, "the total valve area in this type of engine need not be more than the plunger area." As already pointed out, there is no necessary relation between the valve and the plunger area at all. The relation is only incidental, or whatever it happens to be after the proper proportions are established. A certain area of plunger, with a certain stroke, at a given number of revolutions per minute, sets up a certain velocity in the water through the valve seats. A plunger of half the area, with the same stroke and at twice the revolutions per minute, will set up the same velocity of displacement, and consequently the same mathematical velocity will be required through the valve seats; although the increased frequency of opening and closing will introduce another element for consideration, which will call for a greater proportionate valve area, for the greater number of revolutions per minute. In other words, a larger plunger running slowly will require the same valve area as a smaller plunger running faster, so far as the calculated displacement and velocity are concerned. The valve area in both cases depends upon the quantity of water and the selected velocity through the valve seats, regardless of the size and speed of the plungers.

9 The spring diagram and expressions are very nicely worked out, but the differentiation is too fine for real work, and could be for the most part avoided by keeping the valves closer to their seats and

avoiding refinement in springs. The idea is to get away from the laboratory engine, determine the conditions to be found in a pumping station, and then meet those conditions as they really exist, rather than try to adjust the working conditions to some real although impracticable refinement in some particular factor.

10 In many pumping engines now at work, some of the details worked out very nicely on the drawing board but failed to meet the actual requirements. There are waterworks engines of the cage valve construction, in which the ends of the valve stems, with valves exactly like those shown in the paper, have been sawed off, the valves being kept in place by means of wooden wedges, just because someone who never saw the inside of a pump after it left the shop did not understand the requirements involved in the care and maintenance of the machine. In one or two such cases, the cages were difficult to remove and there was not room enough to remove the valves, with the cages in the pump chamber, by the regular method of taking off the spring guard.

IRVING H. REYNOLDS. Mr. Nagle calls attention to two very common errors which purchasers of pumping machinery fall into when preparing specifications:

a The absurdity of specifying the ratio between plunger and valve area without other limiting clauses.

b Specifying an unnecessarily large amount of valve area.

Mr. Nagle suggests as a remedy for the first, specifying *velocity* through the valves rather than a *percentage* of plunger area, and for the second, the use of lighter springs, thus enabling the valves to rise to their full lift and thereby reduce the number of valves required.

2 In regard to the first, there is an increasing tendency among engineers to specify a maximum velocity of flow through the valves rather than their area relative to the plunger.

3 Quietness of operation rather than cost is the first consideration in the design of pump valves, and the present excessive valve areas have grown from this idea. Time is also an important element in determining pump valve action; therefore, the number of reversals or valve seatings, rather than the piston speed, is the important factor, and consequently valves of small diameter and therefore of relatively low lift, have displaced the large diameters in common use a few years ago.

4 To decrease further the lift of the valves and, therefore, permit them to close quickly and quietly at high speeds, valve areas have been increased to a point where in actual operation the valves lift only a fraction of the theoretical height to which they should lift to give a full opening; in other words, large valve area is provided for the purpose of not using it.

5 If on a high-speed (high-revolution) pump the valves were fitted with light springs, permitting them to lift to their full height as suggested by Mr. Nagle, it is probable that the pump would be exceedingly noisy, as the valves would be so far from their seats at the time of plunger reversal that they would not seat until the flow through them had reversed, and this slowness in seating would be still further aggravated by the light springs employed. There is no doubt, however, that in many cases the springs used are unnecessarily stiff and on slow-speed engines the lighter springs would be found satisfactory.

6 In earlier practice, particularly with direct-acting pumps, the valve area was small in proportion to the plunger and the valves were obliged to lift nearly to their full height. In this type of pump, as the plunger speed was relatively high to nearly the end of the stroke, the valves became noisy if the pumps were operated at high speed.

7 With the general introduction of the crank and flywheel pump came higher rotative speeds and the necessity for larger valve area and smaller valve opening, i. e. lower lift, until present practice has crystallized at velocities of 3 ft. to 3½ ft. per second through the valves, and valves of between 3½ and 4 in. in diameter for ordinary waterworks service. In general, the best results would be obtained if engineers in drawing specifications would limit the mean velocity of water through the valves at about 3 ft. per second and the diameter of the valves to not over 4 in.

F. W. Salmon. I prefer to make these valves somewhat different from the one illustrated in the paper. I do not believe it is best to use the radial ribs of the valve seat to screw it in, but that it is better to cast small projections on the outside, as at *A*, Fig. 1 herewith. This part is of such a size that an ordinary black pipe will fit neatly when properly milled out at the end, thus making a good socket wrench at a minimum cost.

2 I prefer to put a brass plate on the top of the rubber valve, as shown at *B* (Fig. 1), and partially to punch out and turn up little pro-

jections from this plate as at C (Fig. 1 and Fig. 2.) The plate prevents the spring wearing into the top surface of the valve, and the projections keep the spring properly centered.

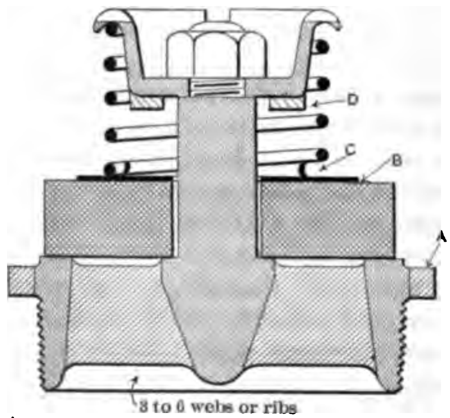

FIG. 1 CROSS SECTION OF PUMP VALVE, SHOWING IMPROVEMENTS SUGGESTED BY MR. SALMON

3 Small projections should be cast on the under side of the spring guard as shown at *D* (Fig. 1 and Fig. 3), the latter being the under side of the spring guard. If the valve is ever drawn so high as to come into

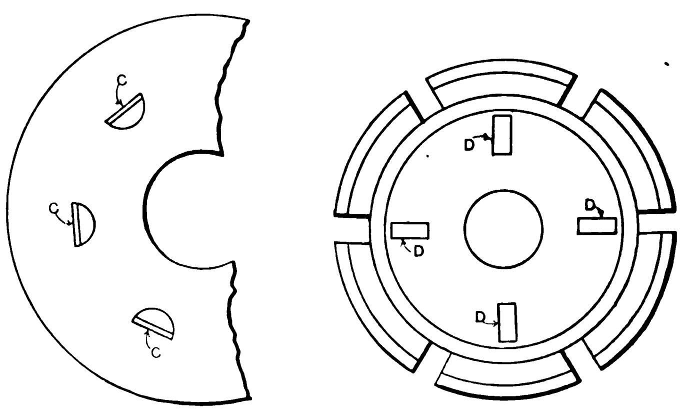

FIG. 2 AND FIG. 3 SHOWING PROJECTIONS ON BRASS VALVE PLATE AND ON SPRING GUARD

contact with these projections it will still descend freely, not being in the least hindered by the soft surface of the valve forming a close contact with the smooth under-surface of the spring guard, as it is sometimes made. I consider that this is useful in cases of fast running pumps, as in such machines it is particularly desirable to have the valves seat while the crank is passing the dead center, and so a quick closing action is required.

William Kent. I hope Mr. Nagle will supplement the paper by telling us what proportion of valves and valve springs he would use for certain conditions. The paper is now largely one of criticism, and I would like to have the author make it a constructive paper. Par. 24 reads "The place to begin the study of proportions of a pump is at the spring of the valve. Make a sample spring of such diameter and length and strength as you may think desirable, and by experiment construct a diagram of its rate of compression, as in Fig. 1." This is good advice for pump designers, but other mechanical engineers are called in to confer about these points, and if Mr. Nagle would tabulate the proportions of springs suitable for pumps, and give the lifts at certain velocities of water, his paper would be more useful to these engineers.

2 The author criticises the practice of specifying the percentages of area of the valve and the pump. I see nothing very wrong in that, provided the plunger area and the speed are also specified, as is usually done, otherwise some of the bidders will put in a small pump. In order to compel them to supply a pump large enough, we limit the velocity of the plunger; and having limited the velocity of the plunger and specified its size, we may as well say that the valve must be so many per cent of the plunger area, as to state what the velocity of the water must be. The specification is good enough, provided these additional items of plunger area and speed are also specified.

Prof. R. C. Carpenter. It is quite evident to any one familiar with hydraulics that the difficulties arising from the lessening of the valve area are largely inherent in the spring. If a spring could be obtained which would open uniformly with increase of pressure the troubles due to certain inertia effects which are mentioned, would disappear. This, however, merely points out the source of trouble and leaves the question open as to what shall be done. In substance, defects are merely pointed out without remedies. I would suggest that Mr. Nagle, if he can, give some of these remedies for the troubles which he has described.

E. H. Foster. Attention should be called to the fact that this paper refers to the valves of one type of pump. Many pumps of other types are built, particularly those without fly wheels, to which it is not absolutely necessary that these rules should apply. It is well known that a considerable pause at the end of the stroke of the duplex pump facilitates the closing of the valves, so that these empirical rules for lift and area must be quite different for that type of pump.

The Author. Some new matter which has come to the attention of the writer of this paper, is appended herewith. A careful study of this will answer most of the points raised in the discussion, especially the point made by Mr. Reynolds and Mr. Hague, to the effect that the maximum velocity through the valve should be limited to 3 or 4 ft. per sec. It can be assumed that Formula [2] of the paper and Table 1, quoted from experiments of Prof. Karl Bach of Dresden, governing the relation of spring pressure and velocity of flow to the coefficient of contraction, are correct.

2 Let us apply Formula [2] to the 3 ft. per sec. assumption. For a lift of, say, 0.20 × diameter,

$$p = 0.77 \times \frac{V^2}{100} \text{ or } = 0.07 \text{ lb. per sq. in.}$$

of inside seat area. Such small spring pressure is out of all proportion to what common practice has established, which is from 0.30 to 0.60 lb. at the initial and 0.75 to 1.50 lb. at the full lift. The formula for the resulting velocities is very simple. Suppose we solve for four spring pressures of, say, 1.50, 1.25, 1.00 and 0.85 lb. at full lift, and 40 per cent, or 0.60, 0.50, 0.40 and 0.34 lb. at the initial point at a lift of 0.20 × diameter, the formula would be

$$V = 1.14\sqrt{100 \times p}$$

and the velocities for

$$\begin{aligned} p = 1.50 \text{ lb.} \quad & v = 13.96 \text{ ft. per sec.} \\ 1.25 \text{ lb.} \quad & v = 12.74 \text{ ft. per sec.} \\ 1.00 \text{ lb.} \quad & v = 11.40 \text{ ft. per sec.} \\ 0.85 \text{ lb.} \quad & v = 10.51 \text{ ft. per sec.} \end{aligned}$$

The coefficient of contraction would be 53 per cent in each case.

ABLE 1 PROFESSOR BACH'S EXPERIMENTS WITH A FLAT VALVE AND A FLAT SEAT (SEE FIG. 1)

Inside diameter of valve seat d = 1.968 in. Outside diameter of valve d_1 = 2.362. Ratio of inside and outside areas, 1 to 1.44. Inside area, 3.04 sq. in.

		H = 1.27 – 1.29 ft.				H = 3.08 – 3.11 ft.	
1	M = lift of Valve, in..........	0.23	0.55	1.01	0.122	0.40	0.65
2	G = Weight of valve, lb......	2.028	2.218	2.304	4.610	5.073	5.238
3	Q = Volume of water, lb. per sec..........		6.548	8.554	3.086	6.768	11.20
4	H = Head of water, ft........	1.29	1.29	1.27	3.11	3.10	3.08
5	W = Velocity through seat, ft. per sec..........		4.97	6.46	2.33	6.17	8.46
	Calculations by Nagle						
6	$\frac{m}{d}$ = Ratio lift, to diameter.	0.12	0.28	0.51	0.06	0.20	0.33
7	G = p Weight per sq. in., lb. per sq. in...........	0.666	0.728	0.760	1.516	1.668	1.723
8	V_g = Velocity due to p, ft. per sec..............	9.55	9.47	9.07	14.70	14.72	14.35
9	V_h = Velocity due to H, ft. per sec................	9.12	9.12	8.92	14.14	14.10	14.07
10	Ratio of $\frac{V_g}{V_h}$................	1.04	1.04	1.02	1.04	1.04	1.02

Line 7 is obtained by dividing the weight G, given in line 2, by the area of d, or 3.04 sq. in.

Line 8 is obtained by the aid of Table 2, where opposite the value of $\frac{m}{d}$ is found the coefficient and formula. For example: taking the first case of a lift of 0.23 in., or a percentage of 0.12 of the diameter 1.968 in., we find by interpolation in Table 2, the formula, $V = 1.17\sqrt{100\,p}$, or $V = 1.17 \times 8.16 = 9.55$ ft. per sec.

Line 9 is obtained from the fundamental hydraulic formula $V = 8.025\sqrt{H}$, when H is the head in feet and V the velocity in feet per second. For example, in the first case cited we have, H = 1.27 ft. $V = 8.025\sqrt{1.27}$, or = 9.12 ft. per sec.

Line 10 is self-explanatory, and is introduced as a check upon the work and formulæ, as if correct, it should be unity. The slight deviations are due to the various decimals not being carried far enough, but they are carried far enough for all practical purposes.

TABLE 2 ORIGINAL DATA AND RESULTS OBTAINED

1	$\frac{m}{d}$ =	....	0.05	0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.5d
2	u =	....	0.65	0.60	0.56	0.53	0.50	0.47	0.44	0.41	0.37
5	p =	0.67	0.69	0.72	0.74	0.77	0.80	0.83	0.86	0.89	0.92 $\frac{V^2}{100}$
6	v =	1.22	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06	$1.04\sqrt{100p}$

Line 1 $\frac{m}{d}$ is the actual rise, or lift, of the valve, divided by its inside seat diameter.

Line 2 u is the coefficient of contraction at the point of discharge with a given lift.

Line 5 p is the pressure in pounds per square inch and is found by dividing the weight of the valve (in water), plus its spring pressure in pounds by its inside seat area in square inches.

Line 6 v is the velocity of the issuing stream at the point of discharge in feet per second.

3 It is plain, therefore, that we are far from realizing four feet per second with our present spring practice.

4 To Mr. Reynolds: The writer did not mean to lighten the springs abnormally, in fact, 0.45 lb. to 0.50 lb. initial is probably light enough, but if they could be made somewhat longer, so as not to tighten up too rapidly, it would seem to be desirable.

5 To Mr. Kent: The formulae given by Professor Bach are a very great addition to our knowledge of pump-valve action. Within the limits prescribed, we have now a safe guide for valve construction. What it should be for other numbers of revolutions and plunger veloc-

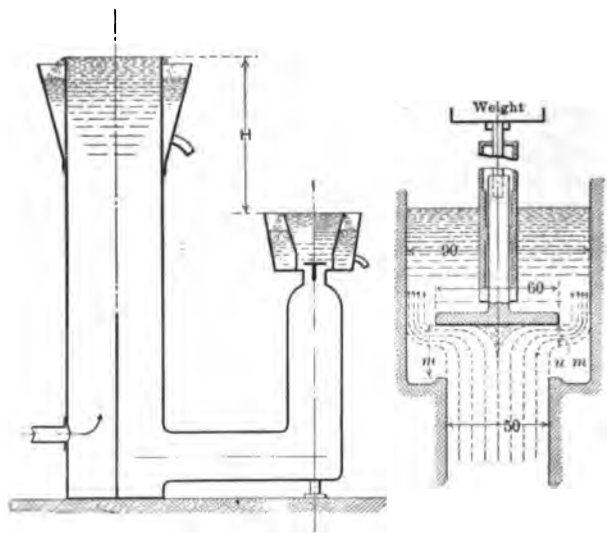

Fig. 1 Apparatus used by Professor Bach for Determining Coefficient of Contraction

ities, I am not able to formulate. Prof. Herman Haeder of Duisburg goes into that phase of the problem, but as his theory is not confirmed by extensive experience, I have not taken it up in this paper.

6 In Par. 15 of the paper is given a formula for ascertaining the lift of a pump valve, from which was omitted, as stated, the coefficient of contraction. Not knowing the value of this coefficient with certainty, the writer hoped the information would be supplied in the discussion. The omission was not referred to, however, and he is now able to supply it himself.

7 In a German book on pumps and pump valves[1] by Professor Haeder, the subject is treated in an exhaustive manner. The actual coefficients of contraction are given, with the results of impact upon the valve, based upon experiments by Professor Bach. In what follows reference is made only to that part which bears on the subjects of flat valves and flat seats, of which the inside and outside areas bear the ratio of 1.00 to 1.44. The notations were originally in French, but in what follows have been transformed into English units.[2]

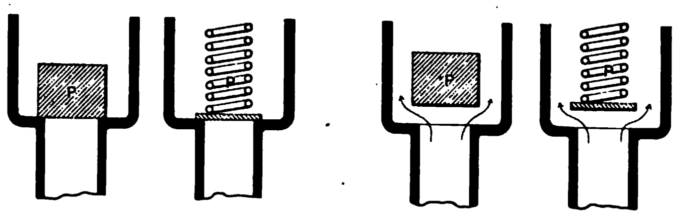

FIG. 2 AND FIG. 3 SHOWING, RESPECTIVELY, A VALVE OPEN AND A VALVE CLOSED. THE FORMULAE FOR THESE TWO POSITIONS ARE AS FOLLOWS:

V = VELOCITY IN FEET PER SECOND P = POUNDS PER SQUARE INCH

VALVE CLOSED	VALVE OPEN
$V = 1.22 \sqrt{100 P}$	$V = 1.04 \sqrt{100 P}$
$P = 0.67 \frac{V^2}{100}$	$P = 0.92 \frac{V^2}{100}$

8 Fig. 1 shows the apparatus used by Professor Bach. Table 2 (Haeder 261) gives the original data and results obtained and some calculations of my own, the better to elucidate the subject.

9 Fig. 2 and Fig. 3 (Haeder 110 and 110a) show a valve closed and one open, with the respective formulæ for the two positions of the valve, giving the values for velocity or pressure in the two extreme positions. "Open" signifies a lift of one-half the diameter, which, needless to say, is far beyond American waterworks practice.

10 Table 2 (Haeder 213) gives the values of v and p for the intermediate positions of the valve, and also the value of "u", the all-important coefficient of contraction at all positions. Use this table to ascertain the velocity v of the water through the valve opening and also the coefficient of contraction u at the same point.

[1]Pumpen und Kompressoren, Haeder, Duisburg.

[2] The original tables in French units can be referred to in the author's manuscript on file in the rooms of the Society.—EDITOR.

11 We can now say that we have a practically correct formula for ascertaining the volume of water discharged through a flat disc pump valve of a certain diameter, an assumed lift, and a certain tension of spring.

12 Throughout all the following calculations a maximum lift of valve of 0.15 d, is taken, leaving the reader to make for himself other assumptions of lift and the consequent calculations. Various tensions of springs will be taken, to illustrate the importance of giving more attention than heretofore to the strength and length of springs.

13 Take, for examples, the same dimensions of pump and valve as those used in Par. 16. Formula [3] (Par. 15) would now be better expressed in terms of N, the number of valves, than assuming the number of valves and solving for the lift L. The formula would then read,

$$N = \frac{P \times V_m}{C \times L \times u \times V_m} \qquad [4]$$

Applying this formula to the three different strengths of springs before used, we get the following results:

14 First, ascertaining the velocity through the valve by the aid of Table 2, the spring tensions were as follows;

Case 1: initial, 0.60 lb.; final 1.55 lb. per sq. in.
Case 2: initial, 0.40 lb.; final 1.03 lb. per sq. in.
Case 3: initial, 0.30 lb.; final 0.77 lb. per sq. in.

The formula for the velocities due to these final pressures at a lift of 0.15 d, are

Case 1: $V = 1.16\sqrt{100 \times 1.55}$, or $V = 14.41$ ft. per sec.

Case 2: $V = 1.16\sqrt{100 \times 1.03}$, or $V = 11.77$ ft. per sec.

Case 3: $V = 1.16\sqrt{100 \times 0.77}$, or $V = 10.18$ ft. per sec.

The coefficient of contraction in each case is $u = 0.56$. Substituting these values in Formula 4, we have,

$$\text{Case 1: } N = \frac{908 \times 6.67 = 6056}{10.53 \times 5.625 \times 0.56 \times 14.44} = 127$$

$$\text{Case 2: } N = \frac{6056}{3.317 \times 11.77} = 155$$

$$\text{Case 3: } N = \frac{6056}{3.317 \times 10.18} = 180$$

15 Let us make a similar calculation for springs of the same initial strength, but longer, so that they will tighten only one-half as much in their nine-sixteenths lift. Then the first spring final tension becomes 1.08 lb., the second spring 0.72 lb., and the third spring 0.53 lb.; and the velocities become

$$\text{Case 1: } V = 1.16\sqrt{100 \times 1.08} = 12.05 \text{ ft. per sec.}$$
$$\text{Case 2: } V = 1.16\sqrt{100 \times 0.72} = 9.84 \text{ ft. per sec.}$$
$$\text{Case 3: } V = 1.16\sqrt{100 \times 0.56} = 8.68 \text{ ft. per sec.}$$

and solving for N in formula 4, we have

$$\text{Case 1: } N = \frac{6056}{3.317 \times 12.05} = 152$$
$$\text{Case 2: } N = \frac{6056}{3.317 \times 9.84} = 186$$
$$\text{Case 3: } N = \frac{6056}{3.317 \times 8.68} = 210$$

16 To calculate the loss of efficiency for these different springs let us take the mean pressure on the springs to be the initial, plus two-thirds of the increase, and twice this for the two strokes; and this sum must be divided by the total pump head, say 80 lb., to obtain the loss of efficiency. We would then have

$$\text{Case 1: } [0.60 + (1.55 - 0.60)\tfrac{2}{3}] \times 2 \div 80 = 3.06 \text{ per cent}$$
$$\text{Case 2: } [0.40 + (1.03 - 0.40)\tfrac{2}{3}] \times 2 \div 80 = 2.05 \text{ per cent}$$
$$\text{Case 3: } [0.30 + (0.77 - 0.30)\tfrac{2}{3}] \times 2 \div 80 = 1.50 \text{ per cent}$$

With stronger springs, we would have

$$\text{Case 4: } [0.60 + (1.08 - 0.60)\tfrac{2}{3}] \times 2 \div 80 = 2.30 \text{ per cent}$$
$$\text{Case 5: } [0.40 + (0.72 - 0.40)\tfrac{2}{3}] \times 2 \div 80 = 1.50 \text{ per cent}$$
$$\text{Case 6: } [0.30 + (0.53 - 0.30)\tfrac{2}{3}] \times 2 \div 80 = 1.15 \text{ per cent}$$

Grouping these figures for better comparison, we have Table 3.

17 We have now, in Table 3, figures which enable us to study pump valve constructions in an intelligent manner. The formulæ given enable us to construct a similar table for any other assumed dimension of plunger and its velocity, height of lift of valve, or spring tension.

18 In conclusion the writer wishes to say that now, for the first time in the history of the modern high-duty pumping engine, we have a formula for designing a pump valve that is scientifically correct, and

one based upon hydraulic experiments carefully made by a competent authority. The subject seems important enough to bear repetition in grouping the previous instructions, as follows:

19 Find the area of the plunger in square inches, and the maximum speed of the plunger in feet per second. The latter is found by multiplying the stroke in feet by the maximum number of revolutions per minute, multiplying this result by 1.60, to reduce it to its maximum velocity (the crank velocity), and dividing by 60 to reduce it to feet per second. This product, algebraically expressed by $P \times V_m$ in Formula [4], becomes the numerator of the equation.

20 Determine the size of the pump valve-seat and its net area between the ribs, whether the valve bears on the ribs or not; that will be the inside area of the valve against which the impinging stream acts.

21 Decide what lift of valve you intend to have. American waterworks practice is from 0.10 to 0.20, the diameter of the inside of the outer seat. This lift is designated by L in Formula [4].

22 Decide what spring pressure you will have, both at the beginning and at the full lift. This spring pressure is expressed in pounds per square inch of inside valve area and usually runs from 0.30 to 0.60 lb. per square inch at the beginning of the lift, and it ought not to be quite double this amount when the valve is full open to its stop. It will be this final tension, plus the weight of the valve in water, designated by p in Table 2, that will be the determining factor for the velocity of the issuing stream. To illustrate, if the final pressure be 0.81 lb. per sq. in., with a lift of 0.15 d, the equation (see Table 2) for $V_m = 1.16\sqrt{81} = 10.44$ ft. per sec.

23 The discharging area is the net circumference of the inside valve diameter C, taking out the ribs whether they support the valve or not, multiplying this by the actual lift and this product by the coefficient of contraction u, as found also in Table 2, which, for the lift cited, is 0.56.

24 Algebraically expressed, these factors become the denominator in Formula [4],

$$N = \frac{P \times V_m}{C \times L \times u \times v_m}$$

TABLE 3 $\frac{9}{16}$-IN. LIFT

PLUNGER 34 IN. DIAMETER BY 5-FT. STROKE BY 25 R.P.M. MAXIMUM VELOCITY = 6.67 FT. PER. SEC. VALVES $3\frac{3}{4}$-IN. INSIDE DIAMETER. NUMBER OF VALVES, SPRING TENSIONS AND PUMP EFFICIENCIES

	Initial and Final Spring Tension Pounds	Valve Seat Area Per Cent	Number of Valves	Lift of Valves Inches	Maximum Velocities Feet per Second			Loss of Efficiency Per Cent
					Plunger	Valve Seat	Valve	
1	2	3	4	5	6	7	8	9
1...........	0.60 to 1.55	112	127	$\frac{9}{16}$	6.67	5.96	9.44 – 14.44	3.08
2...........	0.40 to 1.03	137	155	$\frac{9}{16}$	6.67	4.87	7.71 – 11.77	2.05
3...........	0.30 to 0.77	159	180	$\frac{9}{16}$	6.67	4.20	6.68 – 10.18	1.50
				Longer Springs				
4...........	0.60 to 1.08	134	152	$\frac{9}{16}$	6.67	4.98	9.44 – 12.05	2.30
5...........	0.40 to 0.72	164	186	$\frac{9}{16}$	6.67	4.07	7.71 – 9.84	1.52
6...........	0.30 to 0.53	185	210	$\frac{9}{16}$	6.67	3.60	6.68 – 8.68	1.15

Column 3 is obtained by multiplying the number of valves by the net area through the seat (8 sq. in.), and finding its ratio to the plunger area.

Column 5 is taken at $\frac{9}{16}$ in. = $(0.15 \times d)$ in all cases.

All the other data have been already explained.

No. 1268

A REPORT ON CAST-IRON TEST BARS

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Member of the Society

On machinery castings as well as on cast pipes, separate bars are cast, to be subjected to tensile or transverse stress to the breaking point, these results being used as evidence of compliance with the contract specifications. The writer has examined a large number of such test bars for castings used in the Baltimore sewage pumps and here reports the results of this examination and study. Perhaps the most important conclusion is that the test bar is not to be regarded with too much confidence as indicative of the exact strength of the casting.

2 All transverse bars were nominally 2 in. by 1 in. by 24 in. centers. They were cast from two patterns in one mold, made in the same kind of sand as the main casting. The flask was inclined about 30 deg. There was but one gate for the two bars, with suitable risers. The iron for the bars was poured from a small ladle of iron taken as near as might be from the middle of the pour of the main casting. The breaking loads were corrected for varying dimensions of the bars by the formula $W' = \frac{Wbd^2}{2}$, where b and d are the actual dimensions, W the actual breaking load, and W' the corrected load or weight. These results are used throughout this paper. The deflections were not corrected.

3 The tensile bars, 1¾ in. by 6 in., were cast upright in the same mold as the main casting,within three or four inches thereof, and connected by an upper and a lower gate. The tensile bars were turned to 1½-in. diameter and threaded, and the middle portion reduced to 1.129-in. diameter, which is equal to 1 sq. in. area. Table 1 gives the results of the chemical analysis of the several bars tested.

4 From August 5, 1907 to April 4, 1908 there were made 67 *single* tensile bars and the same number of pairs of transverse bars, and the

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TABLE 1 ANALYSIS OF CAST-IRON TEST BARS

BARS USED IN I. P. BED PLATE, AND I. P. FRAME, FOR BALTIMORE SEWAGE PUMPS

Date Cast	Total Carbon	Graphite Carbon	Combined Carbon	Manganese	Phosphorus	Sulphur	Silicon	Tensile Strain	Transverse Load	Deflection
November 21, 1907	3.580	2.830	0.75	0.79	0.485	0.081	1.59	24,900	2440	0.49
November 26, 1907	3.396	2.736	0.66	0.38	0.459	0.124	1.91	22,000	2075	0.40

average of the latter was used in this record. From April 4 to December 19, 1908 there were made 91 pairs of tensile bars and an equal number of pairs of transverse bars, and each piece of the pair is recorded, instead of the average.

5 Of these 249 tensile bars and their corresponding transverse bars, 32 sets—26 flat and 6 round—were rejected for defects due to blow-holes and four tensile bars were too hard to bear threading, but the companion piece was used in this record.

6 Of the 217 specimens here recorded, 42 are designated as abnormal, that is, the ratio between the tensile and the transverse bars was either considerably greater or smaller than the average.

7 By referring to Table 2, it will be seen that of the 175 specimens of cast iron running from 20,000 to 30,000 lb. tensile strength, the ratio of tensile to transverse loads is practically 10 to 1 and the deflection 0.45 in.

TABLE 2 COMPARISON OF CAST-IRON TEST BARS

Number of Specimens	Limit of Breaking Load of Transverse Bars	Breaking Loads Pounds		Deflection Inches	Ratio of Tensile to Transverse
		Transverse	Tensile		
29	2000 to 2200	2065	21,630	0.43	10.47 to 1
36	2200 to 2400	2289	22,940	0.45	10.02 to 1
51	2400 to 2600	2523	24,880	0.47	9.86 to 1
43	2600 to 2800	2756	26,500	0.49	9.61 to 1
16	2800 to 3000	2894	28,460	0.49	9.83 to 1
175	Averages...........	2383	23,732	0.45	9.96 to 1

NOTE.—Transverse bars, rough 2 in. by 1 in. by 24 in. centers; tensile bars, turned 1.129 in. diameter (1-sq. in. area). In this and the following tables the averages given are for the total number of specimens.

8 Table 3 gives 25 abnormal cases where this average ratio is as high as 12.56 to 1 with a deflection of 0.43 in.; also 17 abnormal cases where this average ratio is as low as 7.91 to 1 with a deflection of 0.44 in. And yet the average of both normal and abnormal bars was 10.07 to 1, again very nearly 10 to 1.

TABLE 3 COMPARISON OF CAST-IRON TEST BARS

ABNORMAL RESULTS

Number of Specimens	Limit of Breaking Load of Transverse Bars	Breaking Loads Pounds		Deflection Inches	Ratio of Tensile to Transverse
		Transverse	Tensile		
		Above 10 to 1 ratio			
10	2000 to 2200	2088	27,143	0.41	12.95 to 1
10	2200 to 2400	2294	28,530	0.43	12.44 to 1
4	2400 to 2600	2436	29,600	0.49	12.15 to 1
0	2600 to 2800				
1	2800 to 3000	2890	34,000	0.45	11.76 to 1
25	Averages...........	2258	28,365	0.43	12.56 to 1
		Below 10 to 1 ratio			
1	2000 to 2200	2105	17,600	0.50	8.36 to 1
4	2200 to 2400	2359	18,825	0.41	7.98 to 1
7	2400 to 2600	2487	18,814	0.43	7.57 to 1
3	2600 to 2800	2656	21,230	0.45	8.00 to 1
2	2800 to 3000	2969	24,500	0.47	8.25 to 1
17	Averages...........	2521	19,954	0.44	7.91 to 1

9 Breaking loads, presumably alike, varied in pairs of transverse bars, and also in pairs of tensile bars, as follows:

Out of 65 pairs of flat or transverse bars,
- 14, or 22 per cent, average variation 18 per cent.
- 17, or 26 per cent, average variation 5.4 per cent.
- 34, or 52 per cent, average variation less than 2 per cent.

Out of 65 pairs of round or tensile bars,
- 22, or 34 per cent, average variation 15 per cent.
- 20, or 31 per cent, average variation 5.5 per cent.
- 23, or 35 per cent, average variation less than 2 per cent.

61 other pairs of flat bars, which had only one companion tensile bar, varied in about the same ratios.

10 Two special flat bars and two special round bars, cast in one mold, one gate and at one pour, varied as follows:

2 flat bars, 12 per cent.
2 round bars, 7 per cent.

11 In order to get some more definite information on these variations, if possible, I had a pair of transverse and a pair of tensile bars made and cast in the same mold, and while the average ratio of tensile to transverse strength was again nearly 10 to 1, as shown in Table 4, the same type of bars again varied 12 per cent and 7 per cent respectively as shown in Par. 10.

12 I have no satisfactory explanation for the great variations in these test bars, and we can only accept the fact that mathematical uniformity in strength of cast-iron bars is not found in the present

TABLE 4 COMPARISON OF CAST-IRON TEST BARS

SPECIAL, TWO SETS CAST IN SAME MOLD AT SAME TIME

Number of Specimens	Limit of Breaking Load of Transverse Bars	Breaking Loads Pounds		Deflection Inches	Ratio of Tensile to Transverse
		Transverse	Tensile		
1		2350	23,000	0.50	9.79 to 1
1		2100	21,470	0.45	10.21 to 1
2	Average............	2225	22,235	0.47	10.00 to 1
217	All averages.........	2380	23,970	0.45	10.07 to 1

state of the art. To anyone questioning the results I can only say from my own knowledge of the circumstances that the personal equation did not enter into them.

13 Careful observation of broken bars did not show that the so-called "skin of the metal" was of any appreciable thickness, and the metal was remarkably homogeneous throughout. The tensile bars being turned, the skin, if there was any, of course disappeared. It is my opinion that the skin adds practically nothing to the strength in either transverse or tensile bars, other causes, though obscure, producing far greater deviations.

14 Although many castings were condemned for physical defects, such as blow-holes, shrink-holes, sand-washes, and shifting of cores, not a single case of cold-shut was discovered. This is in marked contrast with the writer's experience on similar work in other foun-

dries. Excepting a number of steam valves, which were of iron too soft for their purpose, but one large casting, a discharge air chamber weighing 16,000 lb., was condemned for being of unsatisfactory iron. In this case the iron was coarse-grained and brittle, and was required to stand at least 23,000 to 24,000 lb. To remove all doubt that the test bars were truly representative of the iron in the main casting, two tensile bars were cut out of a large flange, which had been at the bottom of the mold. These, from the most favored part of the casting, as will be seen, stood but about 17,350 lb., 90 per cent of that revealed by the test bars. In this case there was a remarkable agreement among these pairs of bars.

TABLE 5 TEST BARS FROM CONDEMNED CASTING

Breaking Loads Pounds		Deflection Inches
Transverse	Tensile	
1,968	19,800	0.35
2,019	19,000	0.50
	Cut out of flange	
	17,000	
	17,700	

15 It may be interesting to apply these results to the formula for the strength of cast-iron beams subjected to similar stress. The formula commonly used (Kent, page 268) is

$$R = \frac{3\,Pl}{2\,bd^2}$$

where R is called the modulus of rupture, or stress per square inch of extreme fiber.

P = load at center.
l = length in inches between supports.
b and d = breadth and depth, respectively, in inches.

Making the proper substitutions, we have $R = \frac{3 \times 2380 \times 24}{2 \times 2 \times 1 \times 1}$, or $R = 42{,}840$ lb. This is not the correct figure, however, for the extreme fiber stress, since we know this cannot exceed the tensile strength, which we have found to be 23,732 lb.

16 I think it is better to use D. K. Clark's formula, given on page 507 of his Engineers' Tables,[1] etc., $S = \frac{Wl}{1.155\, bd^2}$, where S = extreme fiber stress, or tensile strength. If we use the tensile strength found in these tests as 23,732 lb., the breaking load W would become $\frac{23{,}732 \times 1.155 \times 2 \times 1}{24} = 2284$ lb. As this is within 4.3 per cent of the average breaking load actually found in these tests, 2383 lb., this formula, using the tensile strength for the extreme fiber stress, seems to me to be the more intelligible and dispenses with the "coefficient of rupture."

17 "Mr. Barlow found by experiment that for 1-in. square bars of cast iron, the breaking weight in tons [2240 lb., I presume] was expressed by the formula $W = \frac{bd^2}{l} \times 13.6$, and Mr. Robert Stephenson arrived by experiment at exactly the same coefficient." (Clark, page 561).

TABLE 6 CIRCULAR TEST BARS CAST IN VERTICAL DRY-SAND MOLD

Bar Mark	Breaking Loads Pounds		Deflection Inches	Value of W by Formula Pounds	Original Diameter Inches
	Transverse	Tensile			
H	3344	23,070	0.15	2948	1.305
H	3344	23,754	0.15	3036	1.305
X	3026	24,670	0.12	3153	1.300
1	2	3	4	5	6

18 If we should substitute the value for W found in these tests we would have W, or $\frac{2383}{2240}$, = 1.064 tons = $\frac{2 \times 1}{24} \times$ a constant, or, constant = 12.77, which is within 7 per cent of the coefficients found by Barlow and Stephenson.

19 Since the foregoing was written I have had the opportunity to observe two circular test bars nominally 1¼ in. in diameter by 15 in. long, with 12-in. centers. These bars were cast from two patterns in one vertical dry-sand mold and poured from a small ladle, first one and then the other, with the results shown in Table 6.

[1] Rules, Tables and Data for Mechanical Engineers, by D. K. Clark: Blackie & Son, London.

20 The tensile bars were taken from the bottom end of the broken test bar, but I do not know whether H or X was poured first. The first tensile bar H had a small air hole, which being corrected for added 7 per cent to its tensile strength, and this is also given in the table. A second bar was then turned up from the immediate joining piece with the result recorded first in the table. The turned bars were 0.937 in. in diameter. Column 6 gives the original diameter. Column 2 was found by reducing the actual breaking loads in the ratio of the cubes of the diameters, and Column 3 was reduced to the square inch area. Why the transverse breaking loads should vary 10 per cent and the tensile bars 4 to 7 per cent the opposite way, a total variation of 14 to 17 per cent, I leave to the reflection of the reader. If we apply Clark's formula for the breaking weights for circular bars, $W = \frac{0.7854 \times d^3 \times S}{l}$, we find the values given in Column 5.

21 In this age of economic production, the cost of these turned tensile bars is frequently objected to by the manufacturer. While blowholes seem to be more frequent in flat transverse bars than in round attached tensile bars, the latter seem liable to a greater abnormal hardness, for which I have no explanation. Some indication of the toughness of cast iron may be seen in its deflection, which is not revealed in a direct tensile pull. I should, therefore, be satisfied with two or three transverse test bars 2 in. by 1 in. by 24 in. centers, and a deflection record poured, as near as may be, from the middle of the pour of the main casting, as giving a fair indication of the iron in the main casting, but mathematical exactness cannot be looked for as yet.

22 If we wish to know approximately the corresponding tensile strength of the iron, we can multiply the breaking load of the 2 in. by 1 in. by 24 in. flat bar by 10. If the test bar is of 1¼-in. diameter by 12-in. centers, its breaking load should be multiplied by 8, to obtain the approximate tensile strength. The general rule seems to be, that where both flat bars agree in breaking loads, the tensile strength is 10 to 1 of the breaking load, but where they differ, the 10 to 1 ratio does not hold. A better practice, therefore, might be to cast three round transverse bars and accept the two that agree, if each is round, as a fair sample of the iron, dispensing with the tensile bars. This concession to the manufacturer, I believe, would entail not only no loss to the customer's interests, but a positive gain.

DISCUSSION

Prof. W. B. Gregory. The writer has recently made a large number of tests of cast-iron specimens of 1-in. square cross section and with supports 12 in. apart, a few being also broken in tension. The results confirm the deductions of the author as to the relationship between breaking loads in tension and in cross bending. The 10–1 ratio holds in these tests as in those given by the author. Table 1 gives the results of the cross-bending tests, the load being applied at the center.

TABLE 1 TESTS IN CROSS BENDING

Specimens 1 in. by 1 in. 12 in., between Centers, Load Applied at Center

Number	Breaking Load Lb. per Sq. In.	Deflection In.
1................................	2250	0.10
2................................	2250	0.10
3................................	2680	0.09
4................................	2410	0.09
5................................	2250	0.08
6................................	2370	0.09
7................................	2240	0.09
8................................	2310	0.08
9................................	2250	0.09
10................................	2470	0.08
11................................	2180	0.10
Mean	2335	0.09

2 From the specimens broken in cross bending, six were selected from which were turned tension test pieces approximately ½ in. in diameter at the smallest section, their length over all being 5 in. The threads at the ends were ¾ in. outside diameter. The test pieces were made to fit loosely into the tension bars of the testing machine so

TABLE 2 TENSION TESTS

Number	Breaking Load Lb. per Sq. In.
1..	22,900
2..	23,300
3..	22,800
4..	23,550
5..	24,600
6..	22,050
Mean	23200

that side stresses were entirely eliminated, and the specimens were broken in pure tension. The results are given in Table 2. The ratio of tensile strength to load in cross bending is

$$\frac{23,200}{2335} = 9.94$$

This comparison can be made only on the basis of averages, as no record was kept of the numbers of the specimens broken in cross bending. The six tension specimens therefore represent six of the eleven specimens broken in cross bending. Specimen No. 9 of the cross-bending tests may be taken as fairly typical. A chemical analysis was made of this specimen with the following results:

Total carbon	4.04
Silicon	1.76
Phosphorus	0.562

3 The mean deflection as given by the author averaged 0.45 in. for two sets of specimens and 0.44 in. for another set. The highest value of deflection in any case was 0.50 in. Since the deflection varies as the cube of the length of specimens between supports, and inversely as the width of section, it follows that the deflection for specimens 2 in. by 1 in., tested with supports 24 in. apart, should be four times the deflection of specimens 1 in. by 1 in., for a length between supports of 12 in. On this basis the specimens tested by the writer should have

$$\frac{0.45}{4} = 0.112 \text{ in.}$$

deflection instead of 0.09 in. average as the tests showed.

4 This raises the question of what deflection ought to be specified for specimens 1 in. square, with 12 in. between supports. Some specifications have recently been brought to the attention of the writer, in which the minimum deflection was placed at 0.15 in. Is this commercial cast iron or does it call for a special mixture, expensive and hard to obtain?

5 The author has mentioned that the "skin of the metal" was of no appreciable thickness. I would like to ask if he has ever tried the effect of rattling on specimens. The process of rattling will remove the sand and the skin of the metal. In this connection the results in Table 3 may be of interest.

6 The tests given in Table 4 are on specimens of the same size as those in Table 3. The metal used was as nearly the same as the

TABLE 3 TESTS OF CAST IRON IN CROSS BENDING

SPECIMENS ROUND, 1¼ IN. IN DIAMETER, 12 IN. BETWEEN CENTERS; NOT RATTLED

No.	Breaking Load Lb.	Deflection In.	Remarks
1.................	2450	0.075	Cast in pairs on end
2.................	3010	0.08	" " " " "
3.................	2670	0.07	" " " " "
4.................	2580	0.14	" " " " "
5.................	2700	0.09	" " " " "
6.................	2580	0.14	" " " " "
7.................	2620	0.08	" " " " "
8.................	2450	0.075	Cast flat
9.................	3360	0.09	" "
10.................	2750	0.08	" "
11.................	2990	0.09	" "
12.................	3170	0.09	" "
13.................	2950	0.095	" "
14.................	2960	0.12	" "
15.................	3060	0.10	" "
16.................	2580	0.075	Cast on end
Mean........	2805	0.093	

foundry could make it and the specimens were placed in a rattler and the sand and "skin" removed by abrasion. From these figures it will be seen that rattling has increased the strength of the specimens by 23.85 per cent. This phenomenon has been noticed by other experimenters.

7 The statement that rattling increases the strength by about 25 per cent seems to be borne out by experiments. The increased strength is probably due to a removal of some of the internal stresses in the specimens and to the fact that the particles of iron, by the

TABLE 4 TESTS OF CAST IRON IN CROSS BENDING

SPECIMENS LIKE THOSE IN TABLE 3, BUT RATTLED

No.	Breaking Load Pounds	Deflection Inches
1................................	3750	0.09
2................................	3330	0.095
3................................	3400	0.08
4................................	3520	0.09
5................................	3640	0.09
6................................	3640	0.10
7................................	2760	0.075
8................................	3670	0.095
9................................	3060	0.09
10................................	4020	0.10
11................................	3440	0.09
Mean..............	3474	0.0904

process of tumbling the bars together, are allowed to arrange themselves so that they are better able to resist stresses.

8 Since the breaking load varies directly as the moment of inertia of the cross section of the specimen about the gravity axis, we have

I_g for the specimens $1\frac{1}{4}$ in. diameter $= \frac{1}{4}\pi r^4 = 0.7854 \times 0.625^4 = 0.1203$
I_g for the specimens 1 in. square $= \frac{1}{12}bh^3 = \frac{1}{12} = 0.0833$

Then

$$\frac{0.1203}{0.0833} = 1.44$$

Making the comparison between the unrattled round specimens and the square ones, we have

$$\frac{2805}{2335} = 1.2$$

Comparing the rattled round specimens with the square ones we have

$$\frac{3474}{2335} = 1.487$$

A. A. Cary. It is unfortunate that the value of the structural study of metals and alloys, by use of the pyrometer and microscope, is not more widely appreciated. I feel safe in saying that by such means all the variations noted in Mr. Nagle's paper can be most satisfactorily accounted for. The fact is now generally recognized that iron or steel identical in chemical composition may possess widely differing mechanical properties which are quickly recognized by microscopic examination.

2 Chemical analyses, as given in Table 1 of the paper, are undoubtedly of considerable value in the investigation of cast-iron; but without a physical examination our knowledge of the ability of the metal to withstand stresses and strains is very uncertain. Not only will investigations of this kind show us the cause of the variations noted in Mr. Nagle's paper, but they will give us the information needed to produce a metal of great uniformity.

Prof. T. M. Phetteplace. It would be interesting to know whether a thorough sand-blasting would have any effect, as different results seemed to be obtained by cleaning off the skin of the material.

THE AUTHOR. Since the paper was written I have had opportunity to examine some instructive records of eleven sets (of three each) of round test bars. The bars were 1¼ in. in diameter, rough, on 12-in. supports, the breaking loads being corrected for actual diameters. The deflections were not corrected.

BREAKING LOADS IN POUNDS. DEFLECTION FROM 0.12 IN. TO 0.15 IN.

1........	3276	3185	3044	4400	4005	2913	3276	3306	3382	3204	[illegible]
2........	3367	3276	3162	3100	3913	3003	3185	3204	2976	3204	3124
3........	3276	3534	3255	3500	3640	3115	3026	2937	3003	2913	[illegible]

2 The three bars in each set were cast in three separate molds, No. 1, or the upper line, being cast from the first pour of the ladle, No. 2 from the middle and No. 3 from the bottom. It will be observed that in eight of these eleven sets, the bar selected from the two nearest in agreement came from the middle of the pour, and that all of the extreme variations were found in either the first or last pour. If we have only two bars they would differ as much as 22 per cent, while if we took the two out of three nearest in agreement, those two would not vary more than 2 per cent or 3 per cent.

3 I have had no experience with bars 1 in. by 1 in. by 12 in., but I think that the deflection of 0.15 in., mentioned by Prof. Gregory would be difficult to realize in machinery castings.

No. 1269

SYMPOSIUM ON

THE EFFECT OF SUPERHEATED STEAM ON CAST IRON AND STEEL

Three papers: Cast Iron Fittings for Superheated Steam, by Prof. Ira N Hollis, Boston, Mass; The Effect of Superheated Steam on the Strength of Cast Iron, Gun Iron and Steel, by Prof. Edward F. Miller, Boston, Mass.; Cast Iron Valves and Fittings for Superheated Steam, by Arthur S. Mann, Schenectady, N. Y.

CAST-IRON FITTINGS FOR SUPERHEATED STEAM

By Prof. Ira N. Hollis, Boston, Mass.
Member of the Society

The failure of a number of large cast-iron fittings in use with superheated steam has rightly created a widespread suspicion of this material when exposed to high temperature. Yet on this subject there is very little information of a character to justify the wholesale substitution of steel castings for the ordinary heavy cast-iron fittings. The latter have been used with success for many years at all degrees of temperature below actual redness, and in many stations now in operation with moderate degrees of superheating (say 100 deg. fahr.) cast iron has never given the slightest trouble beyond the ordinary were and tear.

2 The doubt as to the reliability of cast iron has seemed to spring up with its use in long pipe lines to steam turbines where the temperature has ranged from 550 deg. to 600 deg. This would lead one to ask if the difficulty has not been in the design of the piping systems rather than in the character of the material. Has not the cast iron taken the brunt of a new service and has it not suffered in the estimation of the engineering public because the conditions of that service were not fully understood?

Presented at the Boston Monthly Meeting (December 1909) of The American Society of Mechanical Engineers.

3 A vast amount of experiment and investigation would be required for the satisfactory reply to this question, and this brief paper is not intended as a reply, but rather to place before the Society a record of some tests that may throw light on the subject. These tests were made for the Edison Illuminating Company of Boston for the purpose of determining the bursting strength under hydraulic pressure of some large fittings which were replaced with steel castings.

4 It may be well before giving the result of the tests to inquire what is actually known about cast-iron fittings subjected to high temperature; that is, known without the possibility of controversy:

a Fittings have developed cracks and small changes of shape after a few months of actual service.

b Fittings exposed separately to superheated steam at a temperature exceeding 500 deg. fahr. have shown a permanent increase of some dimensions.

c The tensile tests of pieces cut from fittings that have failed in service indicate in some cases the possibility of permanent loss of strength.

5 The remainder of the evidence in the case may be classed as good deductions from laboratory tests of specimens previously exposed to high temperature, or from some preconceived theory as to the behavior of the constituent parts of cast iron in a rising temperature.

6 One of the curious and interesting qualities of cast iron is its permanent increase of dimensions under high temperature. This is paralleled by the permanent set of cast-iron test pieces when subject to very moderate tensile stresses. In both cases the cast iron apparently continues to grow at a decreasing rate, at least in some dimensions, when the high temperature or tensile stress is repeated.

7 How long this growth would continue is not known. Its probable limit is the flow of the material under the ultimate breaking stress. Cast iron may not be peculiar in this respect and all materials may change their dimensions permanently under moderate stress, the change growing with each imposition of the same stress. There is no doubt of this where the yield point has been exceeded. It may also be true that all materials change permanently under repeated application of high temperature.

8 The cause of the persistent expansion under high temperature is still very hazy, but two possible agencies have been mentioned in a number of discussions:

a A chemical, or physical, change in the relation of the iron to the various foreign substances which fix it as cast iron.

b A molecular change due to the fact that cast iron has no well-defined elastic limit or modulus of elasticity.

9 Both causes may be in operation at the same time, but the theory of chemical change has far less standing than that relating to the stresses produced by unequal expansion. While there is a temperature at which carbon changes its relation to the iron, superheated steam is probably well below that point except under very unusual conditions.

10 That the strength of cast iron is materially reduced when exposed to superheated steam at 600 deg. is not conclusively proved. Test specimens taken out of cast-iron fittings after one year or more of exposure to a temperature of 550 deg. to 600 deg. have shown a surprising irregularity of strength in the same casting, but there is nothing to prove that new cast-iron fittings have not a great lack of homogeneity. Irregularities exist in every casting owing to the inability of the metal to flow when cooled below a certain temperature. Furthermore, the strength of a test piece cast from a given heat can rarely be taken as that of any selected part of the fitting cast from the same heat. It is common experience to find variations of strain in castings as well as variations of texture. Were any large, irregular casting cut into small test pieces, the variations of strength would probably be found to be quite as great as that reported later on in this paper. The demonstration of the loss of strength after long service with superheated steam does not seem complete, in spite of the fact that some qualities of cast iron have shown a loss in the laboratory.

11 A very brief description of the essential features of the Edison station at South Boston where the condition of fittings has been investigated will help to make clear what follows. The new equipment of the station is arranged in a series of complete units each consisting of one vertical Curtis turbine and eight boilers set in pairs. The main steam line extends along the rear ends of the boilers just beneath the brick work, four 8-inch vertical steam mains connecting each pair of boilers with the main line. Three of the vertical mains discharge through gate valves into tee's, and the fourth, at the end of the line, through a gate valve into a bend.

12 The first turbine units were provided throughout with cast-iron fittings, which were ultimately replaced with steel fittings. No expansion or slip joints are used. The main steam line (something

over 103 ft. long) is anchored at the turbine end and is allowed to expand freely in a longitudinal or horizontal direction carrying the lower ends of the vertical mains with it. The steam pressure is 175 lb., and the superheating generally amounts to 150 deg. fahr., all though it is not constant. The actual temperature of the steam varies from 500 deg. to 580 deg., so that the main line is changing in length from time to time, thus moving the lower ends of the vertical mains back and forth. A series of variable stresses are consequently introduced into all parts of the pipe system, probably affecting most seriously the tees. It is this aspect of the case, namely, the effect of varying stresses upon cast iron at high temperature, that must be studied before a sound verdict can be reached.

13 The castings in the South Boston station were first suspected of failure when nearly a year after the turbine plant had been in operation one of the 8 in. by 6 in. by 6 in. tees near the boiler showed signs of deterioration, cracks appearing near the junction of the offset with the body of the tee and in the flanges. Another fitting of the same dimensions and location began to fail and was taken out after fourteen months' service. Both these tees were cut up for testing and the results were so much alike that only the second is given here as that tee had been exposed the longest to the strain.

14 A chemical analysis gave the following: carbon, 3.47; manganese, 0.10; phosphorus, 0.366; sulphur, 0.062; silicon, 1.41. The tensile strength of six pieces taken from different parts of the tee was found to be, 12646, 14295, 26080, 27270, 27440, 28280 lb. per sq. in. There thus appears to have been considerable variation of strength in the tee unless the first two results are errors due to some faults in testing. Not considering them, the four other pieces do not appear to indicate any great falling off in service. They are as near together as would commonly be found in cast iron from the same heat. The material was supposed to be a first-rate quality of air-furnace gun iron which should have been good for 25,000 to 30,000 lb. per sq. in., but no tests or analyses of the heat from which this tee was poured are on record.

15 Four test pieces cut from a larger tee, 14 in. by 12 in. by 8 in., which had had about the same service as that from which the foregoing test pieces were cut, gave a tensile strength of 23130, 23480, 23875, 24170 lb. per sq. in. Here again there was absolutely no proof that the material had deteriorated.

16 Three test pieces were taken, for comparison, from a large manifold which had been seven years in service with saturated steam,

and the tensile strength was found to be 16413, 16550, 17000 lb. per sq. in. The nature of the cast iron was not known positively, but it was bought as air-furnace gun iron.

17 It was fully recognized in the first of the foregoing tests that, while the material might not have suffered in service, nevertheless parts of the casting might have been weakened by the expansion stresses. For the purpose of testing this, two tees were removed from the line and broken by internal hydraulic pressure, thus affording a definite idea of the strength of the castings as a whole. A third casting, an elbow not previously in use, was added for comparison. The three fittings are shown in Fig. 1 in which the measured dimensions and the location of the fractures are given.

18 The material was the same as that used for all the fittings of the turbine unit, air-furnace gun iron, and the chemical constituents were probably about the same as given for the 8-in. tee in Par. 14. The two tees had been exposed to superheated steam of 578 deg. fahr. and less, for fifteen months or longer, and when removed had given no indications of weakness. A careful examination disclosed no appreciable distortion except in the faces of the flanges which were no longer plane surfaces. There were several high spots that could not have existed when the flanges were faced off.

19 No. 1 was a 14-in. tee with an 8-in. offset. The openings were closed by heavy cast-iron plates fitted to the flanges and bolted. The pressure was produced by a steam-driven outside-packed plunger pump, and was measured by means of a small conical safety valve, one-tenth of a square inch in area, and directly loaded by dead weight applied as the pressure increased. The indications of the small valve were constantly compared with a hydraulic gage previously tested and calibrated at the Crosby manufactory. The fitting broke as shown at an internal pressure of 1650 lb. per sq. in. The plates covering the openings did not reinforce the tee to any great extent as the bolts were smaller than the holes and the joints around the flanges were leaking appreciably when the fracture occurred.

20 No. 2 fitting was an 8-in. by 6-in. tee. It was broken in precisely the same manner as No. 1 and gave way at an internal pressure of 3100 lb. per sq. in.

21 No. 3 fitting was a 12-in. elbow. Its two openings were closed with cast-iron plates and it was burst in the same way as the others. The joints practically gave out at a pressure of 2000 lb. per sq. in. although the pressure was kept on for some minutes. For the second attempt to run the pressure up, the bolts were set up with a very

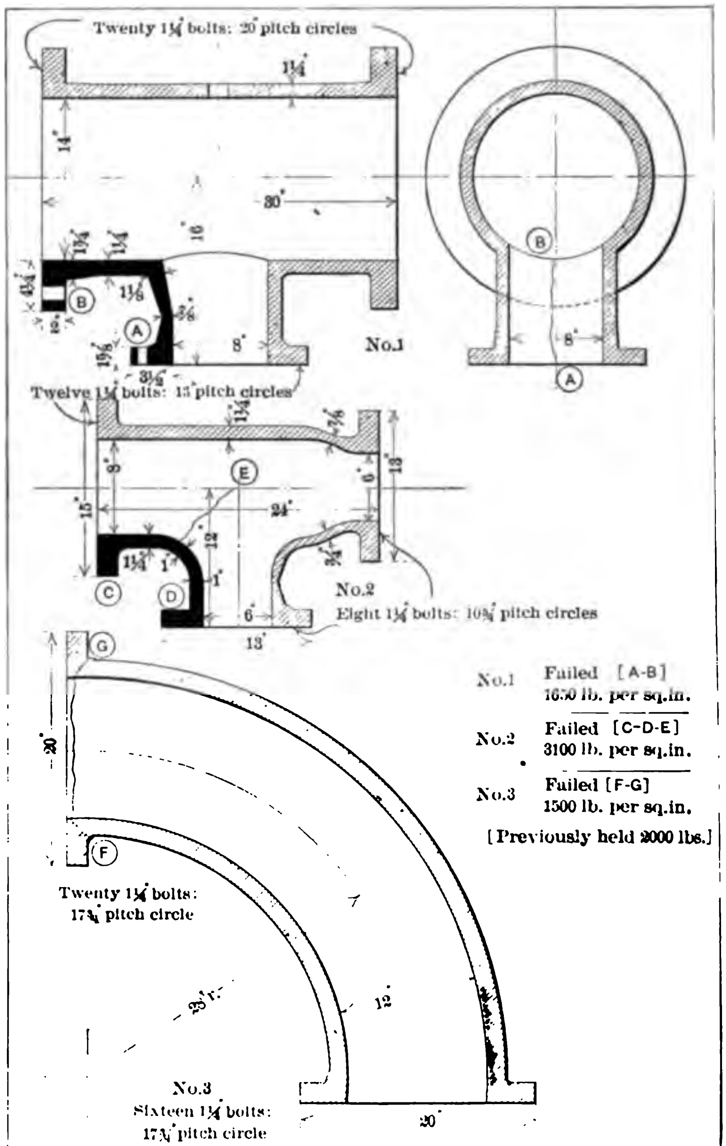

Fig. 1 Showing Points of Failure under Hydraulic Pressure of Three Cast-Iron Fittings used with Superheated Steam

heavy wrench, which undoubtedly put a bending moment on the flange. The fitting finally parted all around the root of the flange at a pressure of 1500 lb. per sq. in.

22 Four test pieces were cut from the larger tee and broken under tensile stress. Their dimensions were almost exactly $\frac{3}{4}$ in. in diameter by 6 in. between fillets. Two of them were broken cold and gave a tensile strength of 22,150 lb. per sq. in., and two were broken at a temperature of 590 deg. at 20,050 lb. per sq. in. The temperature of the latter was maintained by means of a cylinder-oil bath, the oil being placed in a large tube surrounding the test piece and kept hot by a gas flame.

23 No information could be obtained as to the original strength and chemical composition of the iron and it would be impossible to prove that it had changed either in strength or in composition. There is ground, however, for believing that it had changed, as the 8-in. tee gave as high as 28,000-lb. tensile strength in one specimen.

24 A comparison of the larger tee with results of tests reported in the Valve World for November 1907, throws an interesting light on the subject. The formula there published as derived from a very large number of tests of cast-iron and ferro-steel fittings may be taken as a basis for calculating what should have been the bursting pressure of the 14-in. tee. This formula is $B = \frac{T}{D} \times S$, where

D = inside diameter of the T
T = thickness
S = tensile strength of the material multiplied by 60 per cent
B = bursting pressure in lb. per sq. in.

25 Taking the tensile strength of the cast iron when hot at 20,000 lb. per sq. in. the diameter of the tee at 14 in. and the thickness at $1\frac{1}{4}$ in. the value of B is 1070 lb. per sq. in., whereas the tee actually burst at 1650 lb. This did not seem to indicate weakness or deterioration.

26 It is interesting to inquire here what stress existed in the tee during its service. That due to the steam pressure was small when compared with the actual bursting pressure, but that due to expansion may have been serious in its effect. The first tee in the main steam line was located at 4 ft. $8\frac{5}{8}$ in. from the anchorage, the second 37 ft. $8\frac{5}{8}$ in., the third 70 ft. $8\frac{3}{8}$ in., and there was no expansion joint to ease off the pressure on the vertical mains. Taking the third tee for purposes of illustration, certain suppositions can safely be made:

a The lower flange of the vertical pipe moves in a horizontal plane as the main pipe expands and therefore the lowest part of the axis of the pipe moves parallel to itself.

b The upper end of the vertical pipe is practically fixed. The expansion of the main steam pipe thus puts an S bend in the vertical pipes and introduces large bending moments into both ends of it and into the tee.

27 The actual length of the pipe between its lowest flange and the upper end is 26 ft., but the length between the upper surface of the tee and the upper end of the pipe is about 28 ft. The linear expansion of the main steam line is about 3 in. when heated to 578 deg. fahr. The effect of this is supposed to have been halved by cutting the pipe short and springing the flanges into place when making the joints. There is thus an initial deflection in the vertical pipe. This is overcome as the pipe is heated and carried as much farther on the other side.

28 The value of the deflection in the lower end of the pipe is then taken at 1.5 in. The formula for the maximum deflection of a beam fixed at one end and moved parallel to itself at the other end is

$$Y = \frac{Wl^3}{12\,EI}$$

W = load in pounds or push of the horizontal pipe.
l = length in inches.
E = modulus of elasticity.
I = moment of inertia of the pipe.

The inside diameter of the pipe is 8 in. and its thickness is 0.322 in giving the value of I = 72.5. E is taken at 30,000,000.

29 The equation for the deflection is then

$$Y = 1.5 = \frac{312^3\,W}{12 \times 72.5 \times 30{,}000{,}000}$$

and W is found to be 1288 lb. Thus if the expansion of the pipe is exactly split by cutting the main line short, the push on the lower end of the vertical mains is 1288 lb. The point of contrary flexure in the S bend of the pipe is about 179 in. above the junction of the offset of the tee with its main body. The bending moment in the offset of the tee is therefore 1288 × 179 inch-pounds and the stress set up is

$$S = \frac{Mc}{I} = \frac{1288 \times 179 \times 5}{290} = 3975 \text{ lb.}$$

In this formula, 290 is the moment of inertia of the offset taken as 8 in. inside diameter and 10 in. outside diameter. The distance to the remotest fiber from the neutral axis (or center of the offset) is the radius 5 in.

30 While this calculation is not entirely reliable on account of the uncertainty as to the elastic curve of the vertical pipe, nevertheless it is a fair indication of the stress to be expected in this tee when the temperature of the pipe reaches 578 deg. Furthermore, it is made under the supposition that the expansion of the pipe was lessened by an initial pull and that all the joints came exactly fair before setting up the bolts. It is easy to imagine how serious the stresses might have become under actual conditions of inaccurate fitting. The one element of splitting the expansion is very uncertain. The foregoing stress might easily have been doubled, resulting in pulling the tee quite out of shape and in setting up internal strains certain to weaken the material in places.

31 Under such conditions, it was, and would generally be, wise to replace the cast-iron tees with cast steel which would yield more readily to expansion and which would be safer at much higher tensile stresses. The reason for the substitution ought not to be lost sight of in such a case, if cast iron is to be judged fairly. It is made because it is cheaper on the whole to replace the cast iron with steel rather than to put in expansion or slip joints. Perhaps the steel casting is also much easier to take care of than any form of expansion joint. The unreliability of cast iron in such a service has nothing to do with the case: it is merely that the design usually adopted for steam piping does not quite fit cast iron.

THE EFFECT OF SUPERHEATED STEAM ON THE STRENGTH OF CAST IRON, GUN IRON AND STEEL

By Edward F. Miller, Boston, Mass.
Member of the Society

The object of this paper is to describe some experiments made to determine the effect of superheated steam on cast iron, gun iron and steel. From each piece to be tested two tension specimens were made, one to be subjected to the action of superheated steam, and one to be used in obtaining the original strength of the piece.

2 All of the specimens were made with screwed ends in accordance with the specification prepared by The American Society for Testing Materials. The tension tests were made on a 100,000-lb. Olsen testing machine, the specimens being screwed into spherical holders attached to the heads of the testing machine, thus ensuring a straight tension pull without any bending.

3 The specimens to be subjected to superheat were placed on a wire grating suspended at the center of a 12-in. iron pipe about 3 ft. long, supported horizontally on brackets. The ends were closed by blank flanges. Steam was supplied by a small pipe, a flow at low velocity being maintained at all times. The under side of the pipe was heated by Bunsen gas burners. Thermometers, in wells reaching down to the grating on which the specimens were placed, gave the temperature of the steam, the pressure being read from a steam gage on the supply pipe.

4 For the tests plotted in Fig. 1, the average gage pressure in the superheating pipe was 93 lb. and the average temperature 660 deg. fahr. The gas flame was extinguished at 5 p.m. and lighted again at 7 a.m. The temperature reached 660 deg. fahr. by 11 a.m. and by 5 p.m. would be as high as 700 or 720 deg. fahr. Steam was kept in the superheater during the night. The total time these specimens were exposed to superheated steam was 260 hours, and the exposure to saturated steam was 460 hours. A chemical analysis of the iron tested is given in Table 1.

5 For the tests plotted in Fig. 2 the average gage pressure was 82 lb. and the average amount of superheat about 390 deg. fahr. These

TABLE 1 CHEMICAL ANALYSIS OF CAST-IRON SPECIMENS, FIG. 1

	Phosphorus	Total Carbon	Graphitic Carbon	Manganese	Silicon	Sulphur
Cast Iron Gibby Foundry		3.51	3.02	0.37	1.88	0.05
Gun Iron Hunt Spiller	0.41	3.25	2.60	0.24	0.54	0.09
Cast Iron Broadway Foundry		3.34	2.84	0.38	2.26	0.09

specimens were subjected to superheated steam for 520 hours, and to saturated steam for 920 hours. A chemical analysis of three of the semi-steel specimens is given in Table 2. This semi-steel was

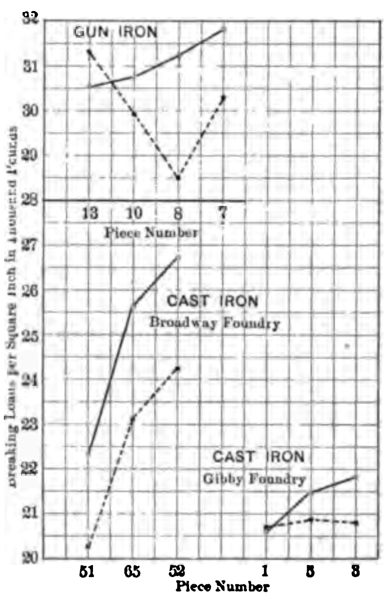

Fig. 1 Breaking Loads of Gun-iron and Cast-iron Test Pieces Subjected to Action of Superheated Steam

made by adding 200 lb. of steel to 1500 lb. of cast iron. The analysis of the gun-iron showed, total carbon, 3.37; graphite, 2.44; manganese, 0.34; sulphur, 0.11; silicon, 1.65.

6 Four grades of steel were tested; two pieces from a bar of 65,000 to 70,000-lb. tensile strength, two from a bar of 75,000 to 80,000-lb.

tensile strength, two each from three bars of about 90,000-lb. tensile strength, and two from a bar of over 100,000-lb. tensile strength.

7 The composition of two of the rolled-steel pieces No. 26 and No. 27 was as follows:

	Phosphorous	Total Carbon	Manganese	Sulphur	Silicon
No. 26	0.85	....	0.73	0.026	0.026
No. 27	0.116	....	0.90	0.057	0.031

8 In Fig. 1 and Fig. 2 the open circles represent the ultimate strength per sq. in. of the original specimen while the dots on the same ordinate gives the strength per sq. in. of the comparison specimen which had been subjected to the action of superheated steam. By figuring the per cent loss in strength in each specimen and then

TABLE 2 CHEMICAL ANALYSIS OF SEMI-STEEL SPECIMENS, FIG. 2

Phosphorus	Total Carbon	Graphite	Manganese	Sulphur	Silicon
.....	....	2.64	....	0.11	....
0.24	3.48	2.39	0.35	0.11	1.91
0.61	3.22	2.83	0.44	0.49	2.62

taking the average of these per cents it appears that the cast iron from the Broadway Foundry (Fig. 1) lost 9.5 per cent; that of the Gibby Foundry 2.4 per cent. The cast iron of Fig. 2 came from the Waltham Foundry; here there is apparently a gain in strength of 1.8 per cent. Fig. 1 and Fig. 2 show that gun-iron loses strength, Fig. 1 showing a loss of about 3.5 per cent, and Fig. 2 about 2.1 per cent.

9 The tests on semi-steel show an average reduction of strength due to exposure to the steam, of about 0.4 per cent, four out of six pieces showing quite a reduction. If piece No. 154 is not considered, the percentage reduction of strength would be much greater.

10 Of the bar steel tested that of 65,000 to 70,000 lb. tensile strength showed a loss of 1.8 per cent due to exposure to the steam, the 75,000 to 80,000-lb. steel a loss of 1.9 per cent, the 90,000-lb. steel a loss of 1.5 per cent, and the 100,000-lb. a loss of 24 per cent.

11 While one is not justified in drawing many conclusions from the results of as few tests as are quoted here, still it is evident from Fig. 1 and Fig. 2 that the metals tested have suffered a loss in strength due to their exposure to the steam. A paper bearing on this subject Materials for the Control of Superheated Steam, by M. W. Kellogg,

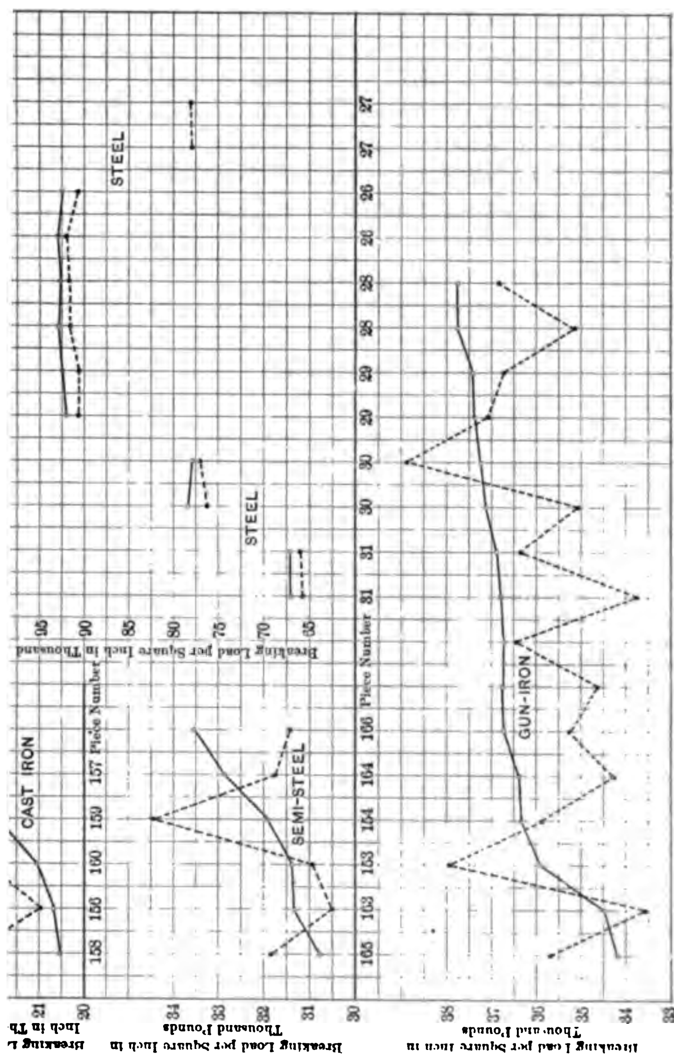

FIG. 2 BREAKING LOADS OF GUN-IRON, SEMI-STEEL AND CAST-IRON TEST PIECES SUBJECTED TO ACTION OF SUPERHEATED STEAM

appeared in the 1907 Transactions of the Society. In the Valve World, March 1908, are given the results of tests on cast iron taken from the body of a 14-in. valve which had been in use for four years on a main carrying steam at 200 lb. pressure and superheated to a temperature of 590 deg. fahr. A number of test bars cut from the body of the valve showed a loss of strength of 41 per cent when compared with the strength of the original metal as determined from coupons tested at the time the valve was made.

12 Fig. 1 in this paper formed part of the thesis of H. A. Terrill, M. I. T. '07, and Fig. 2 part of the thesis of F. M. Heidelberg, M. I. T. '09.

CAST-IRON VALVES AND FITTINGS FOR SUPERHEATED STEAM

By Arthur S. Mann, Schenectady, N. Y.

Member of the Society

There have been many failures of cast-iron valves and fittings in piping systems carrying steam of high pressure and high superheat. The ordinary extra-heavy flanged cast-iron fittings which are listed in many manufacturers' catalogues as suitable for 200 lb. pressure and which have to meet a close price competition, have successfully carried a pressure perhaps as high as 150 lb. or more. No doubt the fittings and valves can support a steady pressure of 200 lb. without bursting, but there have been many failures when carrying superheated steam of lower pressure.

2 These fittings are not well suited for permanent work of even 150-lb. pressure, and many engineers in control of such matters in stations of a representative type prefer to design their own parts rather than to trust the usual run of commercial extra-heavy fittings.

3 Probably on account of the advertised ability to support a high steady pressure these extra-heavy fittings and valves have been used in a number of instances for superheated work. After a short time, six months or perhaps even less, cracks make their appearance; valves leak, seats become loose, castings grow in length and surface cracks become so large in size and in number that the casting is removed from the line.

4 A few repetitions of this experience seem to justify the conclusion that cast iron is not fit material for high-temperature steam. The natural substitute is steel, which is used with fair, even complete, success in many cases.

5 It is known that cast iron will grow with repeated heatings and coolings, often observed in the ordinary straight grate bar. When the bar is first heated it expands and cools as it contracts; but if the temperature has been high, the bar will increase in length. With a second heating, a further increase takes place, followed by many others. As a consequence the long single, straight, flat grate

warps and proves the wisdom of McClave's rule "Keep your long lines of metal away from the fire."

6 This subject of growth has been treated very completely by A. E. Outerbridge in his excellent paper published in the Journal of the Franklin Institute for February 1904. Mr. Outerbridge heated his samples to redness or above, temperatures greatly exceeding that to which a steam-pipe fitting is subjected.

7 A rough experiment on this line was tried by the writer with two samples, one of an ordinary cast iron and a second of a high-grade cast iron, which has proved itself capable of carrying superheated steam and of which a detailed analysis is given in the following pages of this paper. The two samples were each 6 in. long and 1 in. in diameter. They were placed in a banked fire over night, reaching a dull red heat, and were allowed to cool in the air. A slight growth as measured by micrometer was found in each piece.

8 This treatment was followed for two or three nights and the growths were measured. There was an increase in the length of each of the samples, the high-grade iron having increased in length slightly more than did the ordinary iron. The experiment so far as it went tended to show that the growth of cast iron does not necessarily unfit it for the usual degree of superheat in power-house work.

9 Many grades of brass will crumble when heated in a forge to a barely visible red, and are quite unfitted to support any stress at such a temperature. But this characteristic in no way unfits very ordinary cast brass for saturated steam work, and one should not hesitate to use a valve of cast brass up to 3 in. in diameter for 150 lb. saturated steam pressure. Three inches is not usually exceeded because large brass bodies are expensive.

10 Articles have appeared in various publications showing the unsuitability of cast iron, tensile tests being made before and after the use of fittings of ordinary iron. Cases of bronze seats dropping from valves were cited and it was not difficult to prove that something better than ordinary cast iron was needed for steam of 180 lb. pressure and 250 deg. superheat. These failures came from two causes. In the first place the iron itself was not of sufficiently good quality; and, secondly, the parts were not thick enough. The static stress probably did not exceed 1000 lb. in the body: but static stress is not the important load which fittings have to support.

11 Stresses from expansion and contraction within and without the casting and stresses from pulling up joints no doubt greatly exceed the static load even in pipe very carefully erected. The troubles are

aggravated by the action of the steam itself, but it is yet to be proved that the steam or its high temperature will of itself start cracks in a properly designed fitting.

12 The ordinary commercial extra-heavy flanged tee, 8 in. inside diameter, has a body $\frac{7}{8}$ in. and flanges $1\frac{5}{8}$ in. thick. It is made of common iron, having a tensile strength of about 18,000 lb. Such a fitting will fail with superheated steam at 175 lb. pressure and 200 deg. superheat. Within a year the inner surfaces will have a network of cracks, some of which will increase in depth till they extend through the body. The flanges will crack outward from the bolt holes and the fitting will become not only leaky but dangerous as well. The writer has observed just such castings, an analysis of some of them being given later in this paper. Similar effects have been experienced by a great many steam users. The fittings are inherently weak to begin with, so that the failures do not prove that a heavier fitting of better iron is unsuited for superheated steam work.

13 Within the experience of the writer steel fittings have failed with superheated steam. Out of twenty-five steel gate valves, 6, 8 and 10 in. in diameter, not more than four were fairly tight after one year's service, the bodies themselves yielding enough to leak badly. Some defects in the castings developed allowing steam to pass straight through the walls, when they left the foundry. Some of these defects were such that the fittings and valves could not be repaired. In some cases seats were scraped in once or twice and holes were plugged up or patched, but the material would not have been satisfactory without this working over. Yet all these castings were heavy, materially thicker than the commercially extra-heavy cast-iron product, and had passed a rigid inspection.

14 Fig. 1 shows a 10-in. steel fitting; the irregular line at A showing a defect developed after use. The line does not pass clear through the casting, and no doubt the piece was amply strong to resist rupture even after the fault developed. Some of these fissures went 3 in. back and were 5 in. broad. Such a large opening in a shell is objectionable for there are blow-holes enough adjacent to it to pass steam in large quantities. Some fittings of this kind were removed from the line entirely, while others were plugged or patched.

15 No doubt a thoroughly sound steel casting is able to withstand highly superheated steam. There are several connected to the system under discussion. So far as it has been possible to observe, superheated steam does not of itself initiate defects and it is not supposed that the sound metal undergoes a change, either chemically or

structurally. But if there is an initial defect, superheated steam is much more active in bringing out the objectionable features of that defect. It may well be that the material within the body, and not a part of the actual metal, suffers through change of some sort. This material does not add to the strength of a casting but it may serve to stop up holes if allowed to lie undisturbed.

16 It would appear that some material better than the ordinary steel casting was desirable for high temperature work. Such a material is found in gun iron. Gun iron is nothing more than a high-grade cast iron, which any first-class iron foundry can produce. In the days of the smooth bore cannon, a few foundries discovered that it was possible to produce an iron having a tensile strength of 30,000 lb. or more. The government specified it for its guns and it was called gun iron. Probably a tensile strength of 30,000 lb. is not needed in steam fittings, but iron of that quality is well adapted for 180 lb. steam with 300 deg. superheat.

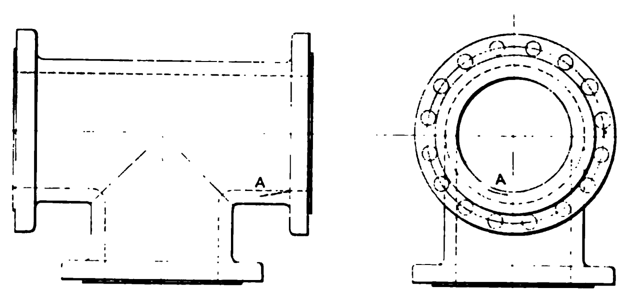

FIG. 1 A 10-IN. STEEL FITTING, THE IRREGULAR LINE AT *A* SHOWING POINT OF FAILURE UNDER SUPERHEATED STEAM SERVICE

17 From such observations as have been thus far possible it appears that certain elements in the iron are liable to cause trouble when present in excess, and perhaps the worst of these is silicon. It is at present going too far to say that every high silicon iron will fail and that every low silicon iron will prove successful, but there is much evidence pointing toward the correctness of such a surmise. In any event iron of low silicon, low phosphorus and low carbon—in other words, gun iron—has proved successful.

18 The following analysis shows the character of a casting which failed at 250 deg. superheat:

Silicon	2.40 per cent	Manganese	0.52	per cent
Sulphur	0.067 per cent	Total carbon	3.19	per cent
Phosphorus	0.94 per cent	Combined carbon	0.25	per cent

19 A second failure developed in this iron:

Silicon	1.98 per cent	Manganese	0.42 per cent
Sulphur	0.068 per cent	Total carbon	3.31 per cent
Phosphorus	0.65 per cent	Combined carbon	0.24 per cent

20 In each of these cases a sample was taken by drilling a hole straight into the body after the part had been in service a year or more and was in bad condition.

21 The following analysis is of an iron that has been successful in every respect for four years under 300 deg. superheat:

Silicon	1.72 per cent	Manganese	0.48 per cent
Sulphur	0.085 per cent	Total carbon	2.45 per cent
Phosphorus	0.89 per cent	Combined carbon	0.17 per cent

22 The latter sample is from an 8 in. valve and it is tight today, no repairs whatever having been made upon the valve during the four years though the bonnet was taken off once to permit internal examination. The outer surface of the valve was covered with 85 per cent magnesia insulation, four and one-half inches thick. The inner surface appeared sound; a microscope revealed no cracks or other defects. The unfinished surfaces were struck several sharp blows with a ball-peen hammer, a hand chisel was driven straight at the surface and some thick chips were cut off from the rough portion. If the metal had suffered to such an extent as cast iron is supposed to suffer, some of the defects would have made themselves manifest. After these treatments the valve was reassembled and has continued to perform its work properly.

23 Foundrymen are not afraid to attempt to produce this iron. No difficulty whatever was encountered in securing bids for valves made of the following mixture:

Silicon	1.40 per cent to 1.60 per cent
Phosphorus	0.20 per cent to 0.40 per cent
Sulphur	0.06 per cent to 0.09 per cent
Manganese	0.45 per cent to 0.75 per cent
Total carbon	3.00 to 3.25 per cent

It will be noted that the percentages of silicon and phosphorous are low.

24 There is of course a decided advantage in depending upon chemical analysis for determining the suitability of fittings. A hole can be drilled at any time in the actual fitting and a few grams of

sample secured. Very few of us are willing to destroy a fitting to obtain a test bar, and test coupons cast in the foundry may or may not represent the actual piece.

25 Superheated steam was in commercial use in Europe before the practice had gained its present hold here. England and Germany were using superheated steam twenty or more years ago. The writer has not discussed this subject with engineers from abroad, but wishes to quote briefly those who have.

26 E. D. Dickenson, of Schenectady, on a recent trip abroad asked a great many manufacturers whether they used steel for their superheated work and received a negative reply in each instance. When the manufacturer was questioned in regard to his iron mixture he shrugged his shoulders and replied that he made his iron fit his needs, be it gas-engine cylinder or steam pipe.

27 John Primrose, in Power and the Engineer, for June 8, 1909, states that he discussed the matter with English and German engineers. In one instance a well-known German engineer, who had used superheat for twenty-five years, was surprised that he had not learned of the effect of superheated steam upon cast iron. The engineer promised to investigate the matter in Germany, but he could find nothing to bear out the contention, and could find no one who believed that such a thing was possible.

28 It is not the author's intention to state that steel of good quality will not do for superheated work. Some manufacturers are putting out fittings of open-hearth steel which are doubtless good; but any foundry can make gun iron if it will, and delay and uncertainty will be decreased by its use.

29 It is clear that a pipe system for conducting steam should be designed properly for its work. Expansion stresses must be taken care of and water must be kept away from it. While steel has a much higher breaking strength than cast iron, fittings that are properly designed, properly installed and properly used may be very well made of a good grade of cast iron.

DISCUSSION ON THE THREE PRECEDING PAPERS

B. R. T. Collins. Last summer I ran across three valves on pipe lines from boilers on the main steam header so located as to be subjected to excessive expansion strains as described by Professor Hollis. They were 10-in. extra heavy valves, with ribs running between the end flanges and also between the bonnet flange and end flanges. These latter ribs were cracked from 1 in. to 1½ in. deep on all three valves. In addition one valve had a crack 1 in. deep in one of the longitudinal ribs, and in one place on the body showed small criss-cross cracks when examined with the miscroscope.

2 The face-to-face length of this valve was originally 18 in. but after two and one-half years' exposure to a superheat of 150 deg. this had increased to $18\frac{7}{32}$ in. This valve was removed, broken up, and pieces sent to Professor Miller for testing, which showed a tensile strength of 11,300 lb. per sq. in. The iron was very coarse, with crystals something like ⅛ in. across. This valve evidently was of very poor material to start with, or else it was seriously affected by expansion strains due to its location or to the superheat. Probably all three of these conditions had their share in producing the result obtained.

George A. Orrok. When we first considered the use of superheated steam in our power stations a few years ago there had been developed a type of steam piping which most engineers considered excellent. The piping itself was of steel with VanStone flanges, the flanges being of sufficient thicknesses to prevent buckling. The fittings were all cast iron of a carefully worked out pattern, much stronger than the ordinary high-pressure fittings. The valves were of similar design and the whole piping system was bolted together with steel bolts of larger size and greater number than the ordinary extra heavy standard required.

2 This piping system gave absolutely no trouble with saturated steam. The up-keep of such a system under power station conditions with 200 lb. steam pressure over a period of a number of years was almost nothing; in fact less than $100 was spent on one pipe line in about three years time.

3 Superheated steam, however, introduced another factor, and a very important one. From certain tests made by the General Electric Company it was considered that this superheat might vary over a range of more than 200 deg. and the temperature strains brought upon the piping and the valves would be severe. It was finally determined to make the entire pipe line of steel. The prices on steel valves and fittings were only a little higher than if of a good quality of cast iron of the thickness required for the high pressure and excessive temperature strains. We adhered to the steel piping with the VanStone joint, but made the VanStone flange of cast steel from the cast iron pattern. The steel fittings were not as heavy as cast iron ones of the same size but differed considerably in the detail of design. The steel valves followed the design of the fittings and were of various makes, both single and double wedge.

4 Our experience with the steel valves has been good and we feel that they are giving better satisfaction than was to be expected under the circumstances. Troubles developed from blowholes, however, which led to an investigation of the subject about a year and a half ago. We traced most of the blowhole difficulties to improper moulding, improper gating and to over-oxidized metal in the case of Bessemer steel and cold metal in the case of open-hearth steel. The valves and fittings are about equally divided between Bessemer and open-hearth steel, all of the former being made, however, on the baby converter by the Tropenas and Zenzes process. The manufacturers understand better today how to handle the work, and the castings which we are receiving are much better than they were two or three years ago. I believe our troubles with the steam lines resulting from superheat are now practically over, and on one steam main in particular we have done nothing in a year and a half. Whether or not the valves can be shut off absolutely tight I do not know as we have had no reason for doing this during the time.

5 After our first installation of steel fittings and valves I had occasion to look up a number of power stations in which cast iron fittings and valves had been installed for use with superheated steam. In one of these stations I saw fittings which had been under the action of superheat for approximately nine months and had been removed because of the many leaks which had developed. The castings were supposed to have been made from the best air furnace iron, but were swollen and bulged practically all over, the outside being covered with fine hair cracks. None of the castings had gone to pieces but practically all had developed leaks, and were being replaced with steel.

6 At another station a cast iron valve had gone to pieces causing quite a little damage and many other valves and fittings had been seriously affected. There were a number of vertical engines in this station in which superheated steam had been used. All the high-pressure cylinders had cracked in two or three places and they were replacing the cylinders and had so arranged their pipe line that no more superheated steam could get to them. The fittings which I examined, taken from the superheat line, had all undergone a growth in size and the outside was covered with fine hair cracks and seemed very much swollen. Analyses of the metal showed a silicon content of from 1.88 per cent to 2.33 per cent, phosphorus about 0.7 of 1 per cent, low manganese and almost no combined carbon. The tensile strength of the material after its exposure to superheat was in the case of the iron with the silicon content of 1.88 per cent about 4500 lb. per sq. in.; in the case of the silicon content of 2.33 per cent it averaged about 8500 lb. per sq. in. We have no means of knowing what this was when it was first made. Microphotographs of the etched surfaces of this metal show the essentially open character of the iron. In this particular station the superheaters have been removed and their troubles have ceased.

7 In view of the many and excessive strains likely to come on a pipe main with 200 lb. pressure and more or less superheat I have not felt that we are justified in installing cast iron valves and fittings. Even with saturated steam at the above pressure and with the length and size of mains which we are using today in our modern stations it seems to me that the extra expense for steel is justifiable and might probably be saved many times over in the cost of up-keep during the life of the station.

8 A few years ago it was the general impression that superheated steam could not exist in the presence of water. This statement has been made many times and no longer ago than at the Annual Meeting. That this idea is fallacious is, I think, the generally accepted belief today, and we have good evidence that it is possible in a steam pipe carrying steam at 200 lb. pressure and 200 deg. superheat to have a stream of water flowing along the bottom of the pipe. In this case the bottom of the pipe would be at a temperature of possibly 380 deg. fahr., while certain other portions of the pipe in contact with the superheated steam might have a temperature between that of saturated steam and the maximum temperature of superheat.

9 Regarding the difference between European cast iron and American cast iron, it has been my impression that the pig-iron

manufacturer here is always trying to make a grade of iron which will command a high price in the market. This iron must be an open iron with reasonably high silicon and almost no combined carbon, the carbon content being in the graphitic state. This iron will sell readily. If the quality of the iron fell off, and because of a lower silicon content more of the graphite was converted into combined carbon, the iron would become harder—more difficult to machine—and would not command as ready a sale. In Europe it is my impression that they make much harder iron and are willing to spend the money to machine it. In America we demand an open iron that can be machined easily.

10 If Mr. Mann continues his researches and considers his test specimens in the light of the volumetric composition of the iron; that is, the volume which the compounds of iron and silicon and of iron and carbon occupy in the cast iron, in comparison with the volume occupied by the iron itself, he may find some interesting results.

11 Referring to air-furnace iron or gun iron as it has been called, I think the great difficulty is the fact that it is almost impossible to control the regular composition of the product. The reverberatory furnace, while a comparatively simple piece of apparatus, is remarkably delicate, and uniform results are obtained only when the very best of care is taken. It is a comparatively easy thing to refine high silicon iron to some kind of refined iron, but it is a much harder thing to get a uniform result from each heat.

W. K. Mitchell. The following notes are taken from several years' personal experience with superheated steam and its effect on cast-iron valves, fittings, etc.

2 Our first intimation that cast-iron fittings and valves gave trouble under superheated steam conditions occurred about three years ago and came in the nature of a surprise, as we had been using superheated steam for some years previous.

3 The first case was in a railway power plant for a high-speed electric line. The plant had been running for several months under a fairly constant load, but owing to a falling off in traffic it was decided to cut down the service to one-half or less, which made the load quite variable. Three months after this had been done the trouble with the fittings and valves began to develop. It was first found that the valves could not be closed tight, and gaskets were giving trouble. Then fittings began to show signs of weakness, cracks appearing on the outer surface.

4 Fortunately these cracks never extended through. In an 8-in. by 6-in. double tee, the metal of which was about $\frac{7}{8}$ in. thick, the cracks did not extend more than half way through, which indicates, that there is no advantage in very thick castings under such conditions. The most serious of these cracks occurred at the junction of the flange and fittings, and kept growing to so alarming an extent that several fittings were replaced. It was then noticed that the old fittings had lengthened considerably. The original length of some 8-in. by 6-in. double-tee fittings was 35 in., and when taken out and cooled they measured $35\frac{5}{8}$ in. to $35\frac{3}{4}$ in. They had been in service about nine months. Open hearth cast-steel fittings and valves were substituted for those of cast iron and have been working satisfactorily ever since.

5 About the only information we could get bearing on the cause of this growth was from a paper by A. E. Outerbridge, read before the mining and metallurgical section of the Franklin Institute in January 1904. Mr. Outerbridge stated that by repeated heating and cooling of bars he had caused the metal to grow to an almost incredible extent. He exhibited a test bar, the original dimensions of which were 1 in. square cross-section and $14\frac{15}{16}$ in. long, which had been heated some 27 times to a temperature of about 1450 deg. fahr., and cooled again by various methods, some slow and some fast, until at the end of the treatment it had grown to a length of $16\frac{1}{2}$ in. and a cross-section of $1\frac{1}{8}$ in. square. The similarity between the action of the fittings above referred to and the test bars which Mr. Outerbridge exhibited caused us to investigate further along similar lines.

6 The railway plant was designed for a steam pressure of 175 lb. per sq. in. and superheat was intended to be 150 deg. fahr. That this temperature had been greatly exceeded, however, was made evident by the discovery of a board that had been charred by contact with the steam trap which rested on it. This trap was connected to a drip pipe running from one of the elbows next to the strainer on a steam turbine and was about 10 ft. below the elbow. Investigation showed that the trap had been so hot that its legs had burned holes through the board until the trap was not resting on the board at all but was suspended by the pipe.

7 The president of the company that installed the superheaters said that while they were built to give an average of 150 deg. superheat, "the real question was not one of the amount of superheat but of velocity." This seems reasonable when one considers that if the

load should fall very low, the velocity of the steam through the superheaters would be considerably reduced and its temperature correspondingly raised. Again, a sudden increase in the demand for steam would result in a rapid flow through the superheater, and steam at much lower temperature, and these recurring changes of temperature must necessarily cause rapid changes in the lines due to expansion and contraction. We therefore concluded that in this particular plant, at least, the damage to the fittings and valves was not caused by the high temperature itself, but by the constantly changing temperatures due to the change of load.

8 Our contention that the damage was due to variable temperatures seemed to be borne out by the fact that in a cotton mill plant installed three years previously, where the steam requirements for pressure and superheat were higher than those mentioned (the pressure being 200 lb. per sq. in., and superheat 200 to 250 deg. fahr.), the fittings and valves were of regular cast iron, and there had been no trouble to speak of. The load was practically constant, however, varying not more than 15 per cent at any time.

9 On account of the discussion in several publications in the spring of 1908 regarding the disastrous effects of superheat on cast iron, the owners of the mill grew anxious about their piping and asked the writer to look over the system. He found everything normal; the fittings were tight, valves could be operated freely, and in a general way the plant was in good condition. The first installation had been made in 1903, and a second one in 1906, using the same class of fittings and valves.

10 The writer suggested that measurements be taken of all the fittings and valves in the plant and records kept of changes. The original dimensions were determined as closely as possible from the patterns and records of construction, and beginning with July 1908 records were kept of the dimensions of the fittings for a period of nine months. Although the changes in the dimensions were slight, the increase in length of certain of the fittings was such that it was thought unsafe to continue them in use and steel fittings were substituted throughout. Most of the valves, however, are still in service and there have been no failures in either fittings or valves. The following will give an idea of the changes which occurred from the dates of installation to the last date given:—

A 12-in. by 10-in. by 8-in. by 6 in. cross installed in 1903 measured 24. in. in length, and in March 1909 measured $24\frac{21}{64}$ in.

A 10-in. by 8-in. by 8-in. tee increased in the same time from 24 in. to 24 $\frac{5}{16}$ in., and

A 12-in. by 12-in. by 8-in. by 6-in. cross, 20 in. long when installed in 1903, was 20$\frac{9}{64}$ in. long in March 1909.

A 12-in. by 10-in. by 8-in. by 6 in. special cross, 50 in. long when installed in 1906, had grown to 50$\frac{1}{2}$ in. in 1909, also

A 10-in. by 8-in. by 8-in. tee, 24$\frac{1}{2}$ in. long, measured 24$\frac{47}{64}$ in. in March 1909.

11 These facts seem to show that even at high steam temperatures, if cast iron can be kept at a uniform temperature and not cooled off too frequently or too rapidly, it will meet the requirements of superheated steam for a long period; but if the temperature is subject to frequent changes such as occurred in the railway plant, the cast iron will become disintegrated and ultimately fail within a short period of time.

12 In another street railway power house which had been in operation for a number of years with saturated steam at 200 lb. pressure, cast-iron fittings, valves and pipe were used successfully. During 1906 fourteen new boilers were added, making a total of thirty-two. The new boilers were equipped with superheaters intended to superheat to about 50 deg. fahr. New piping was installed similar to the old, with cast-iron fittings and valves, and steel pipe with steel flanges. The fittings were unusually heavy and strong. In the original installation of this piping a white metal gasket had been used, about $\frac{1}{16}$ in. thick, which was very satisfactory for saturated steam. These gaskets had a melting point of about 650 deg. fahr., and no sooner had steam been turned in from the new boilers than the gaskets began to melt, and in the course of a month or six weeks it became necessary to replace every one with material that would stand the temperature of the superheated steam. As the majority of the boilers had no superheaters it was hard to understand how sufficient superheat could be generated by the new boilers to do any harm.

13 Two years later a 16-in. tee in one of the connecting pipes between the main headers was found to be leaking. The leak becoming worse, the covering was taken off and the tee was found to be covered with small cracks or fissures similar to the cracks that had occurred in the fittings taken out of the power house first mentioned, except that a few of the fissures had worked through to the inside. The tee was replaced with a new one of the same material and dimensions. When the defective fitting was examined it proved to be some-

thing of a curiosity. Its original dimensions were 31 in. face to face by 15½ in. centre to face. It had grown on one side to 31⅞ in. and on the other side to 32½ in. The flanges, which were originally 25 in. in diameter, had grown to 25¾ in. and as they had been bolted to steel flanges that had not changed under the superheat conditions they had become dished to a depth of about ⅛ in. The original thickness of the body of this fitting, as near as could be determined from the pattern, was about 1¾ in., but where the surface cracks were most numerous careful measuring gave a thickness of almost 2¼ in. Of course, there is always the possibility of the core moving when a casting is being made, but the thickness of this fitting was quite uniform ithroughout.

14 In this plant, the trouble did not stop with the fitting. Several valves began to show cracks and were replaced. Then the high-pressure cylinders of the engines became affected and several had to be renewed. The engineers finally decided to take out the superheaters and the plant is now running without superheat. This is a typical street railway plant, subject to changes of temperature similar to the one first mentioned. Fig. 1 shows the original dimensions of the 16-in. tee and its dimensions after coming out of the line.

15 It is of interest to note that at the same time these superheat boilers were installed in the railway plant, similar boilers with engines, piping, valves and fittings of the same type and material were installed in a lighting plant in the same city, where practically no trouble of any kind had developed. This seems to indicate that a much more constant load is maintained.

16 It is my opinion that in plants where the load is constant and the temperature of the steam therefore constant, properly designed piping with cast-iron fittings of good material will do the work satisfactorily and be safe for a long time. I believe there is no advantage in using cast-iron alloys known as semi-steel, ferro-steel or gun iron. In the railway plant first referred to, the fittings were of cast iron from one foundry; gate valves of semi-steel from another; and stop, check and emergency stop valves also of semi-steel from a third. The results in each case were practically the same. A specimen from an 8 in. by 6 in. double tee which had been in service about nine months gave a tensile strength of 13,750 lb. per sq. in. The chemical analysis of this piece gave the following: Carbon, 2.502; Phosphorus, 0.461; Sulphur, 0.083; Silicon, 2.435.

17 Two test pieces from the 16-in. tee showed tensile strengths of 4970 lb. and 4340 lb. Chemical analysis: Silicon, 2.33; Sulphur, 0.07; Phosphorus, 0.68; Manganese, 0.39; Total Carbon, 3.18.

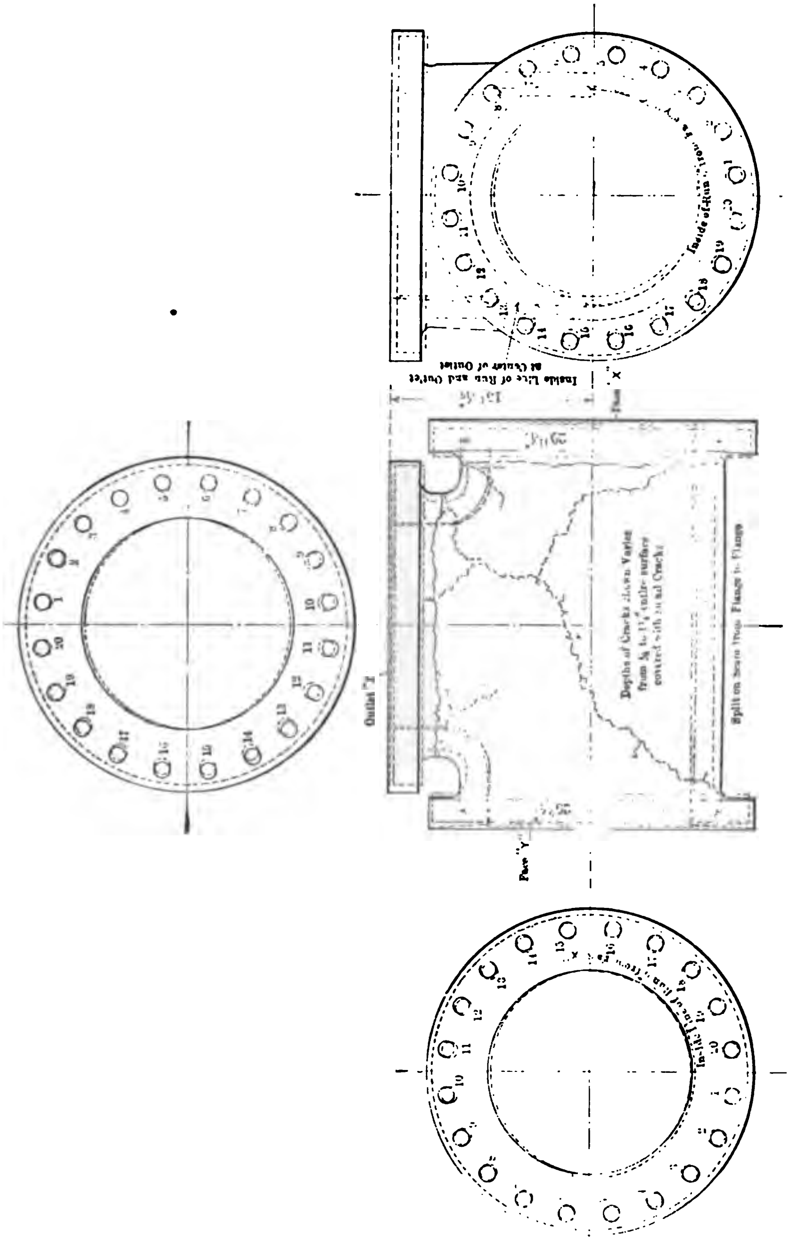

FIG. 1 CHANGES IN DIMENSIONS OF 16-IN. TEE REMOVED FROM STREET RAILWAY POWER STATION

18 In Fig. 2 is shown the first fitting listed in Par. 10, taken from the cotton mill plant where it had been in use several years under superheated steam, the temperature of which, however, was nearly constant. This fitting was tested under hydraulic pressure. At first the pressure was put up to 1100 lb. per sq. in., when the gaskets leaked and the pressure had to be reduced to zero in order to tighten up the gaskets. The fitting was then tested again and broke at a pressure of 1250 lb. There were no serious defects in the fitting. One small surface crack was of so little moment as not to

Fig. 2 12 in. by 10 in. by 8 in. by 6 in. Extra-Heavy Cast Iron Fitting in Use with Superheated Steam, 1903-1908.

require special attention. A testpiece from the fitting showed a tensile strength of 15,900 lb. per sq. in. The chemical analysis was as follows:

Total carbon	3.05
Phosphorus	0.769
Sulphur	0.06
Silicon	2.07

19 It will be noted that the silicon in this specimen is lower than in the two castings just mentioned and I think this had a good deal to do with the case as well as the fact that the load and temperature were constant.

20 While open-hearth steel castings seem to be successfully used under superheated steam conditions, I do not believe they will last

indefinitely because of their extreme thickness. There must be changes taking place similar to those in cast iron, due to temperature changes, but the ductility of open-hearth steel will undoubtedly delay the process of disintegration for a longer period. The material which we recommend and use today for high-pressure superheated steam, is wrought steel throughout, with welded nozzles instead of fittings, and steel flanges, using bends in all cases in preference to short elbows.

JOHN PRIMROSE. During the past eight years the writer has been in close touch with many plants, containing upwards of fifteen hundred installations using superheated steam, and in a position where troubles would be promptly reported to him. Almost without exception these plants use cast-iron fittings in their pipe connections. The fact that no one of these plants has reported troubles with its fittings is in striking contrast to the comparatively few instances where superheated steam has been charged with being the cause of trouble with cast-iron fittings. In order that there should be no doubt about the absence of trouble due to superheat, letters were written to ten concerns known to have been passing superheated steam through cast-iron fittings for the past eight years, at from 100 to 150 deg. superheat, asking the following questions:

Question One. Are not the tees, elbows and valves of cast-iron in the branch and main steam lines leading from the boilers? Seven answered yes, two replied that some fittings and valves were of cast iron and some of cast steel, and one replied that while the fittings were originally of cast iron some tees had been changed to cast steel, but stating positively that the change was not made because of any ill effects of superheated steam.

Question Two. Are fittings of extra heavy or standard weight? Nine replied that they used extra heavy fittings, and one standard weight.

Question Three. What steam pressure do you ordinarily carry? One used steam pressure of 100 lb., six used 150 lb., one 165 lb., one 185 lb. and one 200 lb.

Question Four. Have you ever noticed any injurious effect of the superheated steam on valves or fittings? Eight answered no, one that no trouble was experienced in fittings, but that valves with cast-iron bodies and brass seats were difficult to keep tight, and one reported no trouble further than the baking of a hard deposit on inside.

Question Five. Have you ever found it necessary to replace any of these valves or fittings with cast steel? Eight answered that no fittings or valves had been replaced on account of superheated steam. One answered that they had replaced no fittings, but some globe valves, and one answered that they were replacing some fittings with cast steel, but upon further inquiry it was found that this was not because of the ill effects of superheat, but because the steam mains were being changed to contain VanStone joints and they wished to change the fittings to standard length and deemed it advisable to use cast steel.

Question Six. Of what material are the gaskets in the steam line? Seven use corrugated copper or bronze, two sheet packing, and one asbestos.

2 The chief engineer in charge of a plant in the middle west, of some 20,000 h.p., writes that nothing has developed in any of the cast-iron fittings to show that they are in any way affected by the use of superheated steam. This plant has been in operation about five years.

3 Such evidence as the foregoing proves pretty conclusively that superheated steam does not have an injurious effect on cast iron. There seems to be no very good reason why it should. There is nothing extraordinary in the fact that several cast-iron fittings have failed when passing superheated steam. The failures were probably due to inferior metal, or to strains developed by expansion or contraction of the pipe lines, as suggested by Professor Hollis and Mr. Mann. These are much more plausible theories than that superheated steam at a temperature of 500 deg. to 600 deg. fahr. has any effect on the metal. In investigations by Mr. Outerbridge and Professors Rugan and Carpenter on the growth of cast iron when repeatedly heated, their experiments were started at 900 deg. C. or 1652 deg. fahr. Such instances as the growth of grate bars, etc., are all at temperatures far exceeding anything used in superheated steam work for power plants.

4 Samples of cast iron taken from fittings passing superheated steam for years have been polished and micro-photographed before and after etching, and compared with samples treated in the same way, taken from fittings passing saturated steam. The report states that there is no evidence of a change in the carbon conditions, or of exposure to superheated steam, and in support of this a well known foundryman gives his opinion that a temperature below 900 deg. fahr. would not produce any effect in cast-iron.

5 The tests of the famous Crane valve so often quoted are no proof of superheated steam being responsible for the failure. Test bars from the broken valve were compared with test bars taken from the same heat that the valve was made from, and the valve was said to have weakened. This is no real test, because castings from different parts of the same heat, or, in fact, different parts of the same casting are known to vary in strength, and it is quite likely that fittings passing saturated steam, if compared on the same basis, would be found to have suffered greatly from the effect of saturated steam! It is unquestionably true that this valve must have been subjected to other influences besides superheated steam. It is rather remarkable that the body of the valve is said to have been weakened more than the flanges—the reason given being that the metal of the body was nearer the superheated steam. Is it not more reasonable to suppose that the metal of the body weakened more than that of the flanges because it was subjected to greater fatigue on account of expansion and contraction of the pipe?

6 The writer's experience with a great number of steel fittings used for pressure parts of superheaters exposed to hot gases, has led him to conclude that the metal in steel castings is anything but satisfactory for fittings. It unquestionably has greater tensile strength than cast-iron, which appears to be its only advantage. On the other hand, it is difficult to get steel castings sufficiently homogeneous to hold the pressure. A large percentage of castings are "doctored" before leaving the foundry, but new openings frequently develop on test after machining, and even after the castings are in place, causing the charge to be made that the castings have not been tested before sending out. This fact is further evidence of strains developing in service, other than those produced by internal pressure, which open up cavities or spongy places not discovered by shop test. Steel castings vary greatly in toughness, as shown by the great variation of elongation on test; others are so hard that machining is very difficult. While they can be bought on very careful specifications to guard against these faults, there is always the chance of porosity. The high tensile strength of the steel is not a necessity, and cast-iron made to careful specifications is amply strong. It machines well, is not porous, and can be relied on to hold the pressure.

7 Care should be exercised in the design of pipe lines to guard against straining the fittings from movement of the pipe due to expansion and contraction. Where long radius bends are the means of taking up this movement, pipe of the lightest possible weight consistent

with safety should be used, thereby lessening the force required to spring the pipe. With properly connected flanges, full weight pipe is amply strong for all ordinary working pressures, and if drawn tubing is used, even lighter metal may be adopted. In this connection the design and arrangement of the superheater is of great importance, and should be such that sudden and frequent changes in the temperature of the steam do not occur; otherwise the changes in the length of the pipe will be more frequent, resulting in a more rapid fatigue of the metal of the fittings.

8 A better way of taking care of expansion than with long radius bends, is to use ball and socket expansion joints, which have the additional advantage of reducing the amount of piping.

9 The writer agrees with Professor Hollis in charging strains due to expansion and contraction with the failure of certain fittings, and with Mr. Mann when he charges inferior fittings with the cause of failure in other cases and recommends the use of a good cast iron containing a percentage of steel scrap for fittings passing superheated steam. This is entirely in accord with the writer's experience.

H. S. Brown believed the troubles with cast iron would be eliminated if the temperature could be kept constant, and further said:

2 I think the discusssion boils down to this, that under certain conditions steel castings will give a more satisfactory performance than cast iron. The company with which I am connected has found it necessary to replace a large number of cast-iron fittings with steel castings, where superheated steam was used; and the performance of these steel fittings, under the same conditions under which the cast-iron fittings were working, has been satisfactory.

3 In a large number of other plants where cast-iron fittings are used with saturated steam, the design of the piping is such that the stresses set up on account of expansion and contraction are very much worse than in this system; and we do not get into troubles as we did in using superheated steam. It may be that steel fittings will form a more practical and less expensive way of taking care of the conditions set up by the use of superheated steam than elaborate precautions in the way of expansion joints and the like.

E. H. Foster. Cast iron is much too useful a metal to receive general condemnation for steam pipe fittings, whether for superheated or saturated steam, without very good reasons and the writer is firm in this opinion that such reasons have not yet been advanced.

2 Having devoted the greater part of his time for the past ten years to the study of superheated steam and the manufacture of superheaters, the writer has eagerly followed up every report of the failure of a steam pipe fitting, where superheated steam was used, and it can fairly be said that no instance has yet occurred where the weakness has not been readily explained by the poor quality of the iron, or by lack of provision for expansion and contraction without straining the metal. The many instances where cast-iron fittings are habitually subjected to steam of varying degrees of superheat up to final temperatures close to 1,000 deg. fahr. leave no doubt that good cast-iron is equal to, if not better than, any other metal for making steam fittings for superheated steam as well as for saturated steam, especially in smaller sizes.

3 In the writer's experience it is as important to have regard to the mixture of the iron to be used in cast-iron fittings as it is to secure the proper mixture for concrete work.

4 The suggestion that better provision should be made for free expansion and contraction of steam pipes, is, in my opinion, very much to the point. More care applied to this feature of the design of power plants would remove entirely from the shoulders of cast-iron the odium of being unsuitable for carrying superheated steam.

L. B. Nutting stated that superheaters installed by his company nine years ago, and since then in constant use, have caused no trouble and have not changed their dimensions. These superheaters were made entirely of cast iron, the tubing having a smooth bore and corrugated exterior.

2 He also reported a great many superheaters installed and in operation delivering steam at a temperature of 1000 deg. fahr On these the users have employed, without any distortion or without any evidence of weakness developing, standard makes of cast iron valves (globe valves and angle valves) under 1000 deg. final temperature. But the temperature is maintained at 1000 deg., without a variation of 25 deg. These illustrations seem to trace the cause of the trouble with cast-iron fittings directly to widely fluctuating temperatures.

3 In regard to Mr. Mitchell's suggestion that provision should be made to obviate troubles from varying temperatures on cast iron Mr. Nutting asked, why not make provision to keep the temperature constant. The art of superheater construction has advanced to such a point that a uniform temperature should safely be counted

on. The plant Mr. Primrose referred to, a 20,000-h.p. boiler plant has a record of variation not exceeding 10 deg. either way from the desired amount at any time during the year, although the loads have a fluctuation of from 5000 to 35,000 kw.

Andrew Lumsden.[1] In one case we have eight boilers of the Babcock & Wilcox type, equipped with heaters, part of which were made and installed by the boiler company. These boilers were all connected to one 12-in. main through long radius bends, valves, tees, etc. On the superheaters installed by the boiler company there were usually 150 deg. of superheat and on the others about 90 deg.

2 When this plant had been in service about two years some of the fittings in the main were found to leak just back of the fillets and small cracks were discovered extending around one side of the tees. Some long bolts were made to go the whole length of the tees and through the end flanges, using them to make the joints. Steel tees were also ordered of the same dimensions as the cast-iron ones to replace all the fittings in the main. When the old fittings were removed, however, it was found they were from ⅜ in. to ½ in. longer than when first installed. This is a turbine station and they have had quite a little trouble with the admission valves on some of the turbines, those directly opposite the boiler carrying the 150 deg. superheat giving by far the most trouble.

3 At another plant there are boilers of the Babcock & Wilcox type and Curtis steam turbines, installed seven years ago, with a separately fired superheater on which exhaustive tests were made. The temperature of the steam leaving the superheater reached as high as 750 deg. The superheater was run for about six months and at the end of that time all the copper gaskets in the main were destroyed and the joints had to be remade. The superheater was shut down and has not been operated since.

4 The writer visited this plant a few days ago and found that no large joints had been made since the superheater was shut down. There are about ninety joints ranging from 8-in. to 12-in. diameter and none have leaked, but three 12-in. valves were leaking badly through the body on the under side and about the point where the seat rings are screwed in. These valves are laid on their sides and are on the boiler side of the superheater and have never had superheat in them nor in

[1] President, Lumsden and Van Stone Co., 69-71 High Street, Boston, Mass.

the writer's judgment can their troubles be due to expansion as particular care was taken to allow free movement in all the piping of the plant.

5 At another plant where they have Babcock & Wilcox boilers, Curtis turbines, etc., they have carried 150 lb. steam pressure and 150 deg. of superheat for about six years, with cast iron fittings, etc., and have had no trouble with the fittings. The valves have given them some trouble with loose seat rings and by being badly cut, and some of them have been replaced.

JOHN C. PARKER. Professor Hollis draws attention to the necessity for greater allowance for expansion in piping for superheated steam. This is important and where sufficient flexibility cannot be put into the design expansion joints should be installed. My experience accords with his statement that cast-iron fittings have been largely and successfully used for superheated steam.

2 Six or seven years ago I was called on to furnish superheaters with some of our boilers but could find none in the market to meet my ideas of what a superheater should be. A design was worked out and forty or fifty thousand horsepower of boilers have been built with these superheaters. Two plants are above ten thousand horsepower. In six years experience we have had no trouble either with cast-iron or steel fittings, valves or cylinders. I ascribe the result to the steadiness of the superheat and to the fact that condensed steam in the superheater is not intermittently carried into the steam line.

3 In one of our first installations the men started to flood the superheaters without my knowledge whenever the boilers were banked and the result taught me the effect of suddenly injecting water at 360 deg. fahr. into piping and headers which had been raised to 500 deg. fahr. Leakage started at the joints but stopped as soon as the practice was stopped.

4 I believe there is no connection between the troubles which I have been cognizant of with some designs of superheaters and the expansion of the steam mains. I believe the troubles have been due solely to fluctuations in temperature and temperature shocks caused by frequent injection of condensed steam from incorrectly designed superheaters. I recently went into a plant where a superhea er had been in use for about four years. It was an independently fired superheater and the engineer had had so much trouble with piston rings in an engine with poppet valves designed especially for superheated steam, that he had cut the superheat from 600 deg. fahr. to 500 deg. fahr. and then to 400 deg. fahr.

5 There is a superheater in the market that uses cast-iron to protect wrought iron tubes. I have seen some of these removed from boilers on account of overheating and, while the cast-iron had been red hot it had cracked less than some steam pipe fittings which had been subjected to water jets and fluctuations under 600 deg. fahr.

6 I do not think conclusions can be drawn from Professor Miller's experiments. It would require at least half a dozen tests of the same sample of cast-iron at progressively increasing temperatures and periods to obtain results of value. Ten of the tests show increased strength of cast-iron while all the steel has lost strength. One steel test (100,000 lb.) is unreasonable.

7 We have sixteen 800 h.p. boilers with superheaters directly over the fire running at 175 lb. pressure and up to 170 deg. superheat with no such trouble as Mr. Mann mentions in Par. 12.

Albert A. Cary said that after investigating a number of plants having trouble with the use of superheated steam, he had been led to the conclusion that many if not most of their troubles have been due to bad design in the piping arrangements.

2 Far greater care and better judgment is called for in designing pipe systems for superheated steam than for similar systems using saturated steam, as the strain due to expansion and contraction is greatly increased. The piping on each side of every offset should be carefully considered to see that excessive stress is not thrown upon the flanges and threads by the lever which is developed there. Several special forms of flanges which avoid the screw connection are now used to excellent advantage, with high superheat.

3 Continued flexing on one side of the flanges of fittings, due to the cooling and high degree of heating of the pipe system as steam is turned on and shut off will cause ruptures not unlike those shown in the illustrations accompanying these papers and will be apt to change the internal structure of the metal itself.

W. E. Snyder. I may be able to touch upon certain particular phases of the papers and the discussion, in such a way as to contribute some of the results of actual experience covering a wide variety of conditions and several years' practice. The consideration of this subject in the discusssions seems to have broadened to include the effect of unequal heating of metal and also the designs of systems of steam piping. These matters are both directly related to the use of cast iron fittings for superheated steam, as failures may in some cases be due to the improper design of the steam piping; also under some conditions to unequal heating of such irregular castings.

2 A common connection between engine and main steam pipe is by a branch running horizontally and at right angles to the main pipe, out directly over the high-pressure cylinder and turning down by a bend to connect with the throttle valve on top of the cylinder. When this branch is long the expansion in the steam line does not exert any harmful effect in the throttle valve, but the expansion in the branch is taken up by the change of curvature of the bend over the throttle valve, and this puts a strain directly on this valve. In two or three instances where this kind of connection was in use, the throttle valves were cracked immediately under the flange, and serious accidents narrowly averted.

3 Where the branch to the engine is short the expansion in the branch itself does not require any consideration, but the longitudinal movement of the steam main due to its expansion and contractions, transmits strains through the branch pipe directly to the throttle valve and flange on the steam chest. In one instance this resulted in a very serious accident, as the cast-iron flat top of the steam chest was broken in by the expansion of the main steam pipe several feet away.

4 In another installation a 24-in. cast-iron Y with an 18-in. branch split in the fork of the Y, while under 150 lb. steam pressure. Fortunately it was possible to take the line out of service before the fitting exploded, but it was a very narrow escape. All the accidents mentioned above were the direct results of the installation of systems of steam piping without proper consideration of the effects produced by expansion and contraction. All were in systems using saturated steam, and they emphasize the necessity of using great care in arranging the piping that the expansion and contraction may take place without throwing the severe strains on the cast metal members, which are always liable to failure under such conditions. Consideration of this feature of design is of still greater importance in piping systems using superheated steam, on account of the higher temperature used and the consequently greater expansion. The avoidance of expansion strain on castings in a system of steam piping is of fully as great importance as is the selection of the material from which these castings are made.

5 The effect of unequal heating of metal has been investigated by engineers in the French Navy (See Marine Boilers by Bertin &

Robertson, p. 201). The theory advanced there, which seems reasonable and is also confirmed by experience, is this: When one side of a piece of metal or a boiler tube is heated to a higher temperature than the other, the hot side tends to expand, and the expansion is resisted by the metal on the cold side. This condition puts the metal on the hot side in compression, and the metal on the cold side in tension, and if the temperature difference is great enough the metal will be strained beyond the elastic limit. When the hot side is allowed to cool it is shorter than the cold side because of the strain beyond the elastic limit which has been undergone by both sides. This results in the piece taking a permanent set, or becoming "bow shaped" away from the side that has been heated.

6 The bend away from the fire, of boiler tubes in some types of boilers after they have been in service for some time, seems to be a good example of the results of unequal heating. Another example is the cracking of the large cast iron mud drums used in some types of water tube boilers. Under ordinary operating conditions, in boilers having vertical baffling, the hot gas does not come in contact with the mud drum of the boilers until it has passed at least twice across the tubes, and has thus been greatly reduced in temperature. At times, however, holes are formed in the front baffle, or through the top of the bridge wall allowing the hot gases to pass directly from the furnace to the back part of the boiler setting, where they strike the cast-iron mud drum, heating it to a considerably higher temperature on the front side than on the side away from the fire. These conditions have resulted, in a number of instances, in the mud drum cracking perpendicularly to its axis, causing serious accidents.

7 This matter of the unequal heating of metal is one of the most serious with which designers of large engine cylinders have had to contend. Features of the design of pipe fittings are very similar to those mentioned in the design of gas-engine cylinders, the irregular castings having flanges and other forms of construction which make it practically impossible to avoid having the metal considerably thicker in some places than in others. This irregularity causes internal strains in the metal when heat is applied to one side. It has been the experience of European designers of gas-engine cylinders that one of the greatest difficulties they have had to overcome is this one of distributing the metal so as to avoid the small cracks resulting from irregular expansion, which destroy the cylinder. Where trouble has occurred in the use of cast iron for large fittings in superheated-steam piping, it is possible that the experience of the gas-engine engineers will suggest the remedy.

8 As bearing upon the use of cast iron for superheated steam, particularly upon the much discussed question of the possibility of having superheated steam that is in contact with water in the boiler, an experience of the speaker may be of interest.

9 A large furnace used for heating slabs for a plate mill was equipped with a small vertical Cahall boiler for the purpose of utilizing part of the waste heat. This furnace was fired with under-feed stokers, using forced blast, so that it was possible to obtain a very high temperature both in the furnace and in the boiler, also in the stack. The boiler was set in the usual way for waste heat, i.e., with the large end down. The steam pipe was connected to a flange on the top of the boiler, this connection being made inside the conical-shaped base of the stack which rested on top of the circular boiler setting. By this arrangement a cast-iron elbow on this steam pipe, and about two or three feet of pipe on each side of the elbow were located in the hot gas directly over the upper drum of the boiler.

10 Tests on this furnace and boiler were continued for two weeks, observations being taken every 30 minutes. Frequently the stack temperature would rise to 1000 and 1100 deg. fahr. The thermometer on a Carpenter throttling calorimeter, connected to the steam pipe just outside the stack breeching mentioned above, ranged from 380 to 600 deg. fahr., depending on the rate of working of the furnace. This variation at times occurred very rapidly, and at the time the thermometer readings were high, the steam escaping from the calorimeter was as completely invisible as though it were natural gas. For 12 hours in succession the average superheat of the steam was 140 deg. and during the entire time the tests were being made, the superheat of each 12-hour period averaged 120 deg. or over; the steam pressure being about 95 lb.

11 This boiler has been in operation under conditions similar to the above for at least 15 years. The cast-iron elbow and flange at the top of the boiler, although subjected to such severe service as that described, has never given any trouble. A number of other Cahall boilers using blast-furnace gas, with steam pipe connections made in the same way, have been in operation for about the same length of time, and no troubles have occurred due to the fittings. The service under blast-furnace gas conditions are not so severe as the heating-furnace installation described above, on account of the stack temperature being somewhat lower, from 700 to 900 deg. The heating-furnace conditions mentioned above are unusually severe and for that reason have been described fully. It may be added that the

boilers using blast-furnace gas produced superheated steam, notwithstanding that the water level was only a short distance below the connection to which the calorimeter was attached. (This must not be understood as being a special feature of the Cahall boiler, as in fact it is only incidental to its operation under these conditions.)

J. S. SCHUMAKER called attention to the fact that in the cases cited no difficulty had been experienced with the fittings of superheaters, cast iron or otherwise, but that there had been a great deal of difficulty with steam-pipe fittings. This seems to be the result of high temperature on one side of the fitting only. In the superheater itself the temperatures are balanced, to some extent at least.

DR. D. S. JACOBUS. In a large power plant that I have in mind, where the fittings are all of cast iron, and where the superheat averages 150 deg. fahr., repeated examinations have failed to reveal any deterioration. In other cases, however, where there has been less superheat, and even where a single boiler with superheated steam has been connected into a common main with a number of other boilers furnishing saturated steam, there has been every indication that a small amount of superheat has had an injurious effect. It therefore seems that a difference in the quality of the cast iron may affect the results, and by making a careful study of the matter and knowing the analysis of the cast iron there is a possibility that its action under superheated steam may be predicted. In the meantime, we are furnishing cast-steel fittings for all superheated steam work, as we do not know of a single case of the failure of such fittings that can be attributed to the action of the superheat.

2 The stresses due to expansion, as pointed out by Professor Hollis, may tend to produce failures. In the case of fittings broken in superheated steam lines we have found there was a stress at the point of rupture entirely apart from the stress produced by the steam pressure. In the ordinary flanges the tension of the bolts produces cross strains and the fittings give way where they would naturally fail through this strain. We have given considerable thought to the construction of flanges in which such cross strains are eliminated, but have not pushed the matter forward as we have decided to eliminate all doubt as to the safety of the fittings by employing steel castings.

3 Professor Miller's tests bear out what we have observed regarding the different results to be expected from cast iron, as they show that although there is a general falling off in strength in one case the

cast-iron specimens did not lose in strength by being subjected to a high degree of superheat. In connection with such tests it would be interesting to investigate the action of superheat when the metals are under stress.

4 Mr. Mann's conclusion that gun iron is better than cast steel is indeed interesting, but we would not think of changing our present practice of using cast steel until gun iron is thoroughly tried out in the practical field and demonstrated all right for the work. The proper method of determining the quality of the gun metal which is used must also be developed by the necessarily slow process of observing the action of the fittings in service. It would indeed be a simple matter if bids for the fittings could be based on an analysis of the metal, and I hope Mr. Mann may be right in this belief.

Prof. H. F. Rugan. While investigating the phenomenon of the increase in cubic dimensions of cast iron as a result of repeated heatings it became evident that the test pieces deteriorated in strength. I am of the opinion that the influences at work producing such growth at high temperatures are the same that cause the failure of cast iron fittings at lower temperatures, say at from 500 deg. fahr. to 600 deg. fahr. The effect of the higher temperatures is merely to increase the extent of the changes, producing a maximum growth per heat.

2 Further experiments to determine the length of time required to produce maximum growth developed the fact that a change in the temperature was necessary to produce continued growth. No apparent difference in growth was observed between pieces heated at the same temperature for periods of 3 hours and 17 hours respectively.

3 The test pieces were heated in cast iron muffles, carefully luted with fire clay, to protect them from contact with the furnace gases, to a temperature of from 850 deg. cent. to 950 deg. cent.

4 Experiments were made with nine iron carbon alloys (A to I) containing no graphite, the carbon content changing by 0.5 per cent from 4.03 per cent to 0.15 per cent. Other constituents were low and constant. Four bars of each alloy were cast in both sand and chill moulds. These proved to be all white irons, the samples with low carbon content being full of blow holes. No growth was observed in any save the sample A which contained 4.03 per cent carbon. This sample shrank for the first 12 heats, afterwards expanding, ultimately becoming 6.88 per cent larger than its original volume.

5 Four alloys (J to M) were also tested. Of these J, K and L were grey irons while M was a white iron. It was observed that M

followed along lines closely approximating the action of A, shrinking slightly during the early heats but growing after 12 to 19 heats had been taken; ultimately becoming 6.2 per cent larger than the original. Pieces of the bars from which the A and M test pieces were made were inserted in the muffle to be sampled for chemical analysis after successive heats. These analyses showed that the appearance of free carbon (or temper carbon) coincided with those heats which produced growth in the test pieces. Free carbon was in this way proved to be in some way an indispensable factor in the growth of cast iron when under heat treatment.

6 The grey irons J, K and L, grew from the start, and their progress indicated a close relation between their respective growth and their silicon content.

7 To check these indications a series of alloys, with all the constituents constant save silicon, having the following analyses, were used to test the part played by silicon:

Alloy	Total Carbon	Combined Carbon	Graphite	Si.	Mang.	Sulph.	Phos.
N..............	3.98	0.64	3.34	1.07	0.25	0.01	0.013
O..............	3.98	0.68	3.30	1.79	0.23	0.01	0.013
P..............	3.79	0.30	3.49	2.96	0.25	0.01	0.012
Q..............	3.76	none	3.76	4.20	0.27	0.01	0.012
R..............	3.79	none	3.79	4.83	0.30	0.01	0.012
S..............	3.38	none	3.38	6.14	0.30	0.01	0.013

8 It will be observed that the total carbon in the series is approximately constant, that alloys N and O contain about the same amount of combined carbon, that alloy P contains about half the quantity, and that the remaining alloys contain none at all. The silicon in O and R is 0.2 per cent lower, in Q, 0.2 per cent higher than was desired. The remaining constituents are satisfactorily low and constant.

9 Test pieces N to S, measuring 6 in. by about 0.88 in., were machined from the castings. They were not taken from similar portions throughout, but haphazard, some from the gate, others from a riser either near to or at some distance from the gate. When the growth of these alloys was investigated it became evident that the locations from which a test piece had been cut had a considerable influence on the rate of expansion. It was found that specimens taken from the gate end of the casting grew more rapidly than those taken from the top of the riser.

10 In test pieces from the same part of the bar, however, these inequalities disappeared and a like growth was obtained in each alloy. A slight falling off was observed in the closer-grained irons.

11 The results obtained are plotted in Fig. 3, the coördinates being percentage of growth and number of heats. In this way the rate of growth is clearly seen. In the case of samples N, O and P, curves are plotted from the data obtained. It will be observed that the growth

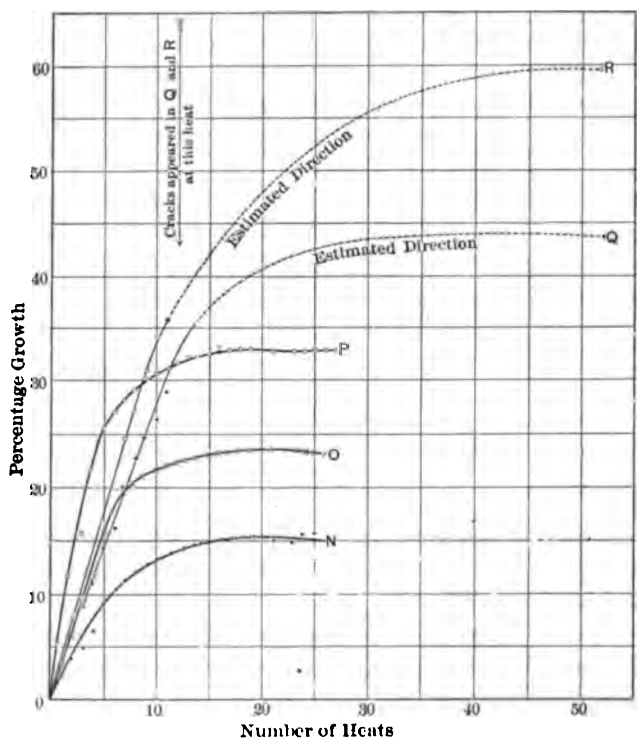

FIG. 3 CURVE SHOWING RATE OF GROWTH OF ALLOYS *N* TO *S*

is rapid at first, diminishes after about the seventh heat and stops at the sixteenth heat.

12 In the case of Q and R curves are plotted in full lines from the data obtained up to the point at which cracks appeared, viz. the twelfth heat. Beyond this the direction of the curves can only be guessed and this is indicated by dotted lines. The following table summarizes the results in relation to the silicon content:

Alloy	Percentage of Silicon	Percentage Growth on Heating
N	1.07	15.40
O	1.79	23.46
P	2.96	32.85
Q	4.20	43.90
R	4.83	59.50
S	6.14	63.00

13 It is quite clear from these tests that silicon is a most important constituent of cast iron from the standpoint of growth under repeated heatings. If the ultimate growths and percentages of silicon are plotted as coördinates, the curve in Fig. 4 is obtained, which shows that, broadly speaking, the growth is proportional to the percentage of silicon.

14 To settle the question as to the influence exerted by graphite and at the same time determine if iron-silicon alloys, containing little carbon and no graphite, would grow, three alloys (T, U and V) were experimented with, having the following analyses:

Alloy	Silicon	Carbon	Mang.	Sulp.	Phos.
T	0.65	0.17	0.17	0.045	0.017
U	1.10	0.18	0.19	0.049	0.022
V	2.71	0.19	0.20	0.051	0.033

15 Microscopic examination showed in all three cases a solid solution of iron silicide in iron. There were no traces of graphite or any other structural constituent.

16 Machined bars were heated fifteen times under the same conditions as the previous alloys. A summary of the final values is contained in the table below:

Alloy	Percentage Silicon	Percentage Change of Volume after Fifteen Heats	Percentage Change of Weights after Fifteen Heats
T	0.65	−0.025	−0.04
U	1.10	0.000	−0.03
V	2.71	+0.394	−0.02

17 It will be seen that the only alloy of the three which showed any tendency to grow was V, with 2.71 per cent of silicon. The expan-

sion, however, was very slight, and compared with that of P (2.96 per cent silicon and 3.79 per cent carbon) after the same number of heats was almost negligible, amounting to but 0.394 as compared with 31.35 per cent, the mean figure of P and PP.

18 Alloys K, N and P correspond closely to alloys T, U and V in silicon content. They also contain about 3.9 per cent of carbon,

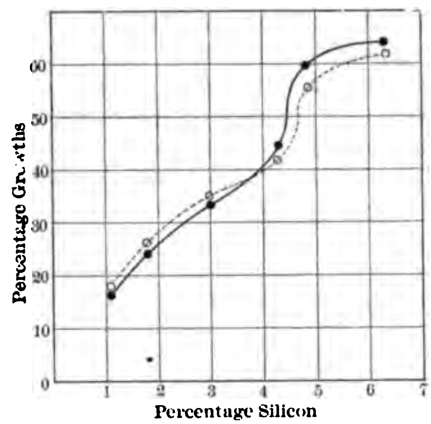

FIG. 4 CURVES ILLUSTRATING RELATION BETWEEN PERCENTAGE GROWTH AND PERCENTAGE SILICON

mostly in the form of graphite, as compared with a mean figure of 0.18 per cent carbon, none of which is present as graphite in the other series. A comparison can thus be made between the changes of volume of the two series under similar tests after fifteen heats, by means of the following:

Alloy	Carbon	Silicon	Per Cent Change in Volume	Alloy	Carbon	Silicon	Per Cent Change in Volume
T.........	0.17	0.65	−0.025	K..........	3.90	0.69	+ 5.40
U.........	0.18	1.10	0.000	N..........	3.98	1.07	+15.20
V.........	0.19	2.71	+0.394	P..........	3.97	2.96	+31.35

19 This comparison serves to emphasize anew that free carbon, even in the form of graphite, is one of the essential factors in the growth of cast irons under heat treatment. The previous series of alloys, however, N to S, brought out clearly the fact that in the constant

graphite series the growth is roughly proportional to silicon present, graphite becoming merely the agent or forming the avenues by means of which the silicon present can be acted upon. It is clear, therefore, that both graphite and silicon are involved in these changes of volume after repeated heatings.

20 A sample of test piece S, known as "S ½," was heated to constant volume in vacuo. This resulted in a shrinkage of 0.04 per cent. The same sample was afterwards heated in the muffle to a constant volume, when a growth of 67.70 per cent was obtained. Fig. 5 shows curves plotted from these data. It will be seen that S ½ grew rapidly during the later heats, with no cracks developed, and the sample retaining its original form throughout. S, however, grew rapidly during the earlier heats, cracks developing during the first heat, and finally breaking in two.

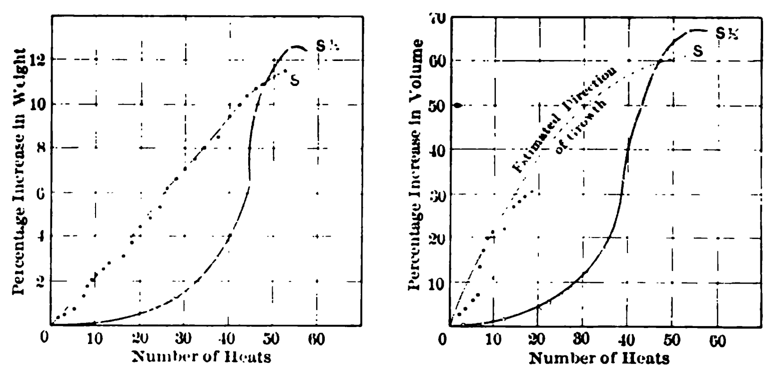

FIG. 5 CURVES SHOWING PERCENTAGE INCREASE IN WEIGHT AND VOLUME AND NUMBER OF HEATS: TEST PIECES *S* AND *S*½

21 Mr. A. Wolfe, superintendent of motive power of the United Railway and Electric Company, Baltimore, commenting upon the growth and final failure of some fittings in one of their power houses says, "The temperature was not constant, varying between that of the temperature of saturated steam at 175 lb. per sq. in. to superheated steam running between 500 deg. fahr. to 550 deg. fahr. total temperature."

22 From the experiments I have made there is considerable evidence indicating that gray cast iron subjected to changing temperatures from 450 deg. fahr. and up gives evidence of an oxidation of the silicon present, forming silica in a micro-crystal form, which upon

cooling causes a disintegration of the surface exposed, ranging generally along the planes formed by the graphite, changing an apparently solid wall into one showing many cracks. It is the constant recurrence of these conditions, produced by the changing temperatures, that, in time not only produces growth, but breaks down the structure of the metal.

23 The experiments conducted by Prof. E. F. Miller were so treated, cooling down each night to the temperature of saturated steam. No mention is made of any growth of these specimens. One would expect to find a marked relation between growth and loss of strength.

24 In a comparison between gray iron samples, those subjected to a heating and cooling treatment totaling 1000 hours would become weaker and larger than those kept constantly at the temperature of superheated steam for a like period. I believe that actual contact with steam is not a necessary condition in this experiment, neither is it a comparative test of the metal, which in service has one surface exposed to atmospheric conditions.

AUTHOR'S CLOSURE[1]

Prof. Ira N. Hollis. The discussion of these papers brings out certain interesting and valuable conclusions which cannot fail to assist in the proper use of cast iron for parts of machinery and boilers. Previously existing differences of experience with this metal under a high temperature are shown to be due to fundamental differences of chemical composition or to variations in the temperature. From this point of view, existing data, even though conflicting, can probably be reconciled. The following conclusions as the result of the papers and discussions may be studied with profit in connection with new construction.

a Cast iron varies in its behavior under high temperature, starting from about 450 deg. fahr. In many cases it deteriorates in structure and strength to a marked degree.

b The effect of high temperature is independent of the medium producing it, whether superheated strain, hot gases or solids.

c. The change of structure or deterioration is much increased by a fluctuating temperature.

[1]Professor Miller and Mr. Mann did not desire to present closures.

2 Where the temperature is constant, even though as high as 600 or 700 deg. fahr., the change in cast iron is not serious enough to prohibit its use, but where the temperature varies considerably, the metal is certain to develop cracks and distortion that render it unsuitable for steam pipes and other parts under steam pressure.

d Cast iron of certain chemical constituents increases materially in volume when subjected to fluctuating temperatures above 500 deg. fahr.

e The chemical composition of the cast iron has a material bearing upon the change of shape and volume and upon the development of imperfections.

3 Certain facts in this connection are well shown by Professor Rugan's experiments. As he states, cast iron containing only combined carbon does not change even under high fluctuating temperatures unless the carbon begins to separate into a graphitic form. Free carbon is one of the factors assisting in the deterioration under high temperature, especially when associated with silicon. The latter seems to be the chief cause of increase in volume. Where the free carbon is constant, the growth in dimensions is roughly proportional to the percentage of silicon.

f The use of cast-iron fittings for superheated strain is inadvisable where the temperature is likely to fluctuate, but it can be safely used where the temperature is to be constant.

g Cast-iron fittings should not be placed in any parts of a steam-pipe line where there are serious bending stresses in addition to the stresses produced by internal pressure, unless the combined stresses are fully allowed for, or neutralized by expansion joints.

No. 1270

NECROLOGY

WALTER MORRISON ALLEN

Walter Morrison Allen, works manager of the Warner & Swasey Company, died February 8, 1909, at his home in Cleveland, O. He was born in Bristolville, O., December 14, 1866, and received his early education in the local schools of Cherryfield, Me.

He evinced an interest in mechanics early in life and when only sixteen years of age frequently went to the nearest railway station, a distance of thirty miles, to study and make drawings of the locomotives that passed that point. In 1885, he began work as an apprentice to the machinist's trade in the works of Warner & Swasey. He was given special opportunities in the drafting-room, and at the completion of his term of apprenticeship was kept in this department, of which he was made head in 1891. During the next two years, the details of the design and construction of the 26-in. telescope of the Naval Observatory and the 40-in. telescope of the Yerkes Observatory came largely under his direction.

In 1893 he had charge of the firm's exhibit at the Chicago exposition and during the following six years was superintendent of their works. In 1904 he was made works manager, the position which he held at the time of his death. He had traveled much in the interests of the company, visiting England and the Continent in 1897–1898 and again in 1900.

Mr. Allen was a member of the Cleveland Engineering Society, the Cleveland Chamber of Commerce, the Colonial Club and the Automobile Club of Cleveland.

A. KENNEDY ASHWORTH

A. Kennedy Ashworth, manager of the filter department and traveling engineer of the Pittsburg Gage and Supply Co., died January 20, 1909, at his home in Crafton, Pa. He was born May 26, 1873, in Covington, Ky., receiving his education in the schools of Pittsburg, Pa., and in Cook Academy; he was also a graduate in mechanical engineering of Rose Polytechnic Institute, Terre Haute, and studied at the Western University of Pennsylvania.

He first entered the employ of the Joseph Horne Company as mechanical engineer, and became chief engineer of their steam and electric plants. After about two years he became a member of the firm of D. Ashworth & Son, consulting engineers. He associated himself with the Pittsburg Gage and Supply Company, and was actively engaged in engineering work after that time, except for a short period when he was connected with the Buckeye Engine Company, establishing a Boston office. He also established offices for the Pittsburg Gage and Supply Company in Philadelphia, New York, Chicago and other large cities.

He was a member of the Engineers' Society of Western Pennsylvania, the order of Free and Accepted Masons, and the Sons of the American Revolution.

ARCHIBALD W. BLAIR

Archibald W. Blair, Junior member of the Society, died July 17, 1909, at the home of his father, Dr. W. W. Blair, in Princeton, Ind., where he was born June 4, 1865. He was a graduate of the Princeton High School and took a special course in mechanical engineering at Purdue University. After serving an apprenticeship in the foundry, machine, boiler and smith shops of the Princeton Foundry and Machine Works, Mr. Blair went in 1890 to Cincinnati where he attained high standing as a mechanical draftsman. At the time of his death he was chief draftsman of the American Laundry Machinery Company and also instructor in the Ohio Machinists' Institute, and a member of the American Association of Engineers. He was a man of many excellent traits and high attainments and will be greatly missed by his associates. Mr. Blair entered the Society in 1898.

FRANCIS H. BOYER

Francis H. Boyer died at his home, Somerville, Mass., February 21, 1909. He was born at Manheim, N. Y., in 1845, and at the age of ten years went to Greensburg, Ind., where he remained until he was 18 years old. He learned the trades of millwright, carpenter and architect. On his return to the East he entered the steamboat transportation business in Brooklyn, N. Y., in which he was engaged during the Civil War.

At the age of 23, he went to the frontier, settling at Seneca, Nemeha

Co., Kan., where he carried on a stock and land business for a few years. He then returned to Brooklyn and engaged in the refrigerator building business, and was associated with the building of the first ship refrigerator for carrying beef to Europe. He eventually became superintendent of the De La Vergne & Mixer Company, refrigerator builders, directing the construction of machinery for brewery refrigerators. Mr. Boyer built the first brewery refrigerator in Boston in 1884. His work called him to Washington, Baltimore, Newark and New York. He settled in Boston in 1890. and was appointed master mechanic of the John P. Squire Company. He designed the big chimney at that company's plant in East Cambridge and remained with the company until its assignment in 1900, when he went into business for himself, with his son as a member of the firm.

Mr. Boyer was a Manager of this Society, 1899–1902, and was chairman of the Local Committee at the time of the meeting of the Society in Boston in 1902. He also did important work on the Committee of the Society for Determining Standard Methods for Conducting and Reporting Steam Engine Trials. He was a member of the Boston Society of Civil Engineers; the American Society of Refrigerating Engineers; honorary member of the National Association of Stationary Engineers and Somerville Council, Royal Arcanum; and president of the Somerville Board of Trade for two years.

THOMAS HALLETT BRIGGS

Thomas Hallett Briggs was born on August 21, 1870, in New York. He received his education in the public schools of Brooklyn, N. Y., and finished with a technical course at Cooper Union. During his studies at Cooper Union, he was employed in the drafting room of the Logan Iron Works of Brooklyn, with which company he remained for fourteen years. His work there covered not only drafting-room work, but shop inspection, and he finally became outside representative. He became associated with the M. H. Treadwell Company, New York, as salesman, in 1904, which position he held until the time of his death on September 24, 1909. Mr. Briggs was a member of the Society of Gas Engineers, and entered The American Society of Mechanical Engineers as an Associate in 1900.

ANDREW JAMES CALDWELL

Andrew James Caldwell, Manager of the Society, died in New York City, May 10, 1909. Mr. Caldwell was born in Brooklyn, N. Y., May 1, 1858, and was graduated from the University of Maine in 1878, with the degree of B.M.E. He also took a graduate course at Cornell University.

He first entered the employ of the Delamater Iron Works, and later that of one of the regular line steamship companies on the New York and New Orleans route. In 1880 he became connected as draftsman with the firm of Henry R. Worthington, and was closely identified with the designing and development of the Worthington high-duty pumping engine. He was soon made chief of the erecting and testing departments, which dealt with water-works pumping machinery; then assistant to the president, Charles C. Worthington; and finally general manager of the hydraulic works.

Soon after the formation of the International Steam Pump Company, he resigned to take a similar position with the Crane Company of Chicago, by which he was later sent to Bridgeport, Conn., to take charge of the new plant of the Eaton, Cole & Burnham Company, controlled by them. In 1902 he resigned, accepting a position with the National Foundry Association, and later entered the service of the Standard Oil Company, with which he was identified at the time of his death.

Mr. Caldwell was elected Manager of the Society in 1906, and was still serving at the time of his death. He was a member of the Engineers' Club in New York, and one of the organizers and the first president of the Brooklyn Engineers' Club.

KENTON CHICKERING

Kenton Chickering, vice-presicent of the Oil Well Supply Company, died December 9, 1908, at his residence in Oil City. He was born in Worcester, Mass., May 16, 1847, and received his education in the Massachusetts public schools.

In 1863 he became a dispatch bearer for General Clark of the United States commissary department in New York City, and remained in the government service for a time after the war. In 1870 he represented Eaton and Cole, dealers in brass and iron goods, at Titusville, remaining with the company when it became Eaton, Cole and Burnham Company, with offices at Oil City. In 1878 Mr. Chickering was made secretary of the Oil Well Supply Company, Ltd., which was formed at this time. This new company absorbed the Eaton, Cole and Burnham Company and others. In 1891, when the Oil Well Supply Company was organized in its present corporate form, Mr. Chickering was elected vice-president, the position which he held at the time of his death.

He patented a number of useful inventions in connection with oil well machinery, and planned the large manufacturing plant erected by the company in 1901–1902, known as the Imperial Works. He also designed a number of special machines to increase the output and improve the quality of product of the plant.

Mr. Chickering was very active in church, civic and fraternal organizations.

GEORGE W. CORBIN

George W. Corbin was born in New Britain, Conn., March 3, 1859. He attended the local schools and Wilbraham Academy until 18 years of age. His first business connection was with P. & F. Corbin, who later organized the Corbin Cabinet Lock Company, making Mr. Corbin manager, and later secreatry and president. He resigned to become president of the Union Manufacturing Company, and held this position until his death, Novemember 30, 1908. He organized several other manufacturing corporations, and took active part in municipal affairs—the savings banks, local government and schools.

He was connected with several social orders, among them the Masonic order, and numerous social clubs.

ELMER G. EBERHARDT

Elmer G. Eberhardt, of Newark, N. J., an Associate Member of the Society, died at his home on November 21, 1908. He was born in Newark, April 26, 1881, and was graduated from the Newark High School in 1896. He received his technical education in Stevens Institute and Cornell University, receiving the degree of M.E. at the latter institution in 1904.

He learned the machine trade with his father, Henry E. Eberhardt, of the firm of Gould and Eberhardt, at Newark, and upon graduation from Cornell University, he formed, with his father and brothers, the firm of Eberhardt Brothers Machine Company, now the Newark Gear Cutting Machine Company. Mr. Eberhardt was vice-president of the company, and was engaged in the design of automatic gear-cutting machines, in which field he invented a number of improvements as well as made investigations along original lines. He was a frequent contributor to the technical columns of the mechanical papers. He designed the power plant and equipment of the factory with which he was connected.

Mr. Eberhardt was elected president of the Co
trical Engineers and vice-president of the Cornell
He was an Associate Member of the American I
Engineers, and a member of the University Clu
Cornell Association of Northern New Jersey.

At the time immediately preceding his death,
engaged, aside from his business connections, in
ing work, in matters relative to gears and gear c

DAVID HAMILTON GILDERSLEEV

David Hamilton Gildersleeve was born in Te
5, 1867, and died in New York City, July 30, 190
at Stevens Institute of Technology, Hoboken, N
in 1889 with the degree of M.E. For nearly te
was active in gas engineering and the constru
pumps and hydraulic machinery, being associa
Gas Improvement Company of Philadelphia, th
Company of Cincinnati, and the Snow Steam Pum
For three years, during and following the Spa
he served as first lieutenant, United States Corps
and as assistant engineer of the Department of
of the mechanical work executed and plann
he became associated with the C. W. Hunt Com
Brighton, N. Y., as sales manager, which posi
February 1909 to become one of the partners in t
of the Waters, Gildersleeve, Colver Company, of
Staten Island.

Mr. Gildersleeve became a member of the Socie
a member of the Machinery Club, the Amer
the New York Railroad Club, Dry Dock Associ
Club, Royal Arcanum, Staten Island Association
Spanish War Veterans Association, the Chi Phi
Secretary of the Stevens Alumni Association.

THOMAS GRAY

Dr. Thomas Gray, Vice-President and Profe
Electrical Engineering of Rose Polytechnic In
Ind., died December 19, 1908.

He was born in Fifeshire, Scotland, Februar
a course in engineering at the University of Glas

he graduated in 1878 with the degree of B.S. Later he took a four-year course in practical physics and telegraph engineering under Sir William Thomson (Lord Kelvin).

He was engaged by the Japanese government as instructor in telegraph engineering at the Imperial College of Engineering at Tokio, Japan. After this engagement he was employed by Sir William Thomson and Professor Fleming Jenkin, engineers of the Commerical Cable Company, to superintend the manufacture and the laying of that Company's system of transatlantic and other cables, and had sole charge under them, as resident engineer, of the whole of that work. He was later chief assistant to Lord Kelvin in his engineering work.

In 1888, he was appointed to the professorship at Rose Polytechnic Institute, and held the position until his death.

Doctor Gray was the author of Directions for Seismological Observations, in the British Admiralty Manual of Scientific Inquiry; of articles on telephones and telegraphs in the Encyclopædia Britannica, and of the Smithsonian Physical Tables. He also wrote many papers on scientific and technical subjects, and was engaged as an expert in electricity on the staff of the Century Dictionary.

LEWIS CLESSON GROVER

Lewis Clesson Grover died at Hartford, Conn., September 30, 1909, after a long illness. He was born November 26, 1849, at Springfield, Mass. After an ordinary school education there, he acted as apprentice at the Norwalk Iron Works, Norwalk, Conn., for three years; and remained with them on the expiration of his term for seven additional years. He was afterwards successively connected with the Winchester Repeating Arms Company, New Haven, Conn., C. W. Lacount of Norwalk, Conn., Smith & Wesson, Springfield, Mass., and F. C. & A. E. Rowland, New Haven, Conn.

He became general manager of the Whitney Arms Company, New Haven, Conn., about 1880, holding this position until 1886, when he went to Hartford as assistant superintendent of the Colt's Patent Fire Arms Mfg. Company. He was soon promoted to the office of superintendent, and later to that of general manager. In 1902 he was elected president and a director of the company, at the same time becoming president of the Colt's Arms Company, of New York. Because of ill health he was finally compelled to relinquish to others the active duties of management, and in January resigned the office of president, the same meeting making him chairman of the boards of directors of both corporations.

Mr. Grover served as a member of the common council board of Hartford and as park commissioner. He was a prominent Mason and a member of the Hatchetts Reef Club. He entered this Society in 1890.

CHARLES LEWIS HILDRETH

Charles Lewis Hildreth was born October 9, 1823, at Concord, N. H., and died suddenly on February 26, 1909, at his country home in Westford, Mass. He received his education at a private school in Nashua, N. H., and at the Appleton Academy in New Ipswich, N. H. In 1845 he went to Lowell, Mass., and became an apprentice in the Lowell Machine Shop, which was incorporated the same year. After a service of three years, he became a contractor on piecework. He was identified for over fifty years with this company, being superintendent for twenty-six years. During the great depression of the iron trade in 1858 he went to Philadelphia and served as foreman in the Industrial Works. He returned to Lowell, Mass., in 1860, and assumed charge of the shop as general foreman, holding this position until 1879 when he was made superintendent of the entire plant. Mr. Hildreth was identified with the textile machinery trade during almost the entire period of its development in America. In July 1905 he retired from business.

He was president of the Mechanics Savings Bank in Lowell, Mass., was closely identified with the Lowell Textile School, and while in Lowell was an attendant at the Kirk Street Congregational Church; in Westford he attended the Union Congregational Church.

WARREN E. HILL

Warern E. Hill was born in New York in 1835. In 1852 he entered the service of the Allaire Iron Works in Newark, N. J., and was associated with that company for six years. In 1858 he was appointed superintendent in charge of the installation of the Detroit, Mich., Water works, which position he held until 1862, when he returned to the East and accepted a position with the Continental Iron Works of Brooklyn. In 1888 he was made vice-president, and in 1907 president of this firm, the position he held at the time of his death. Mr. Hill was the designer of the machinery and engines of the original "Monitor," which defeated the "Merrimac" in Hampton Roads.

His death occurred in New York, December 8, 1908. He became a member of this Society in 1884.

ROBERT HOE

Robert Hoe, head of the firm of Robert Hoe & Company, of New York and London, was born in New York, March 10, 1839, and was educated in public and private schools in this city. He was grandson of Robert Hoe of the hamlet of Hoes, Leicestershire, England, who began the manufacture of printing machines in New York in 1803, constructing and introducing into America the first iron and steel machines.

Mr. Hoe at an early age entered the printing factory established by his grandfather, and devoted his life to the improvement and development of printing machinery. He developed the rotating-cylinder type of press to the present double-octuple press capable of printing, pasting, folding and delivering more than 150,000 16-page newspapers per hour. He also invented greatly improved processes of printing in colors, and is the author of several books on printing and binding.

Mr. Hoe always resided in New York, although his business interests were almost as great in London, and identified himself with its interests and prosperity. He was one of the founders of the Metropolitan Museum of Arts, founder and first president of the Grolier Club, and a member of the Engineers, Union League, Century, Players and Fencers Clubs. He joined The American Society of Mechanical Engineers in 1883.

Mr. Hoe died in London, September 22, 1909.

WILLIAM S. HUYETTE

William S. Huyette was born in Blair, Neb., November 13, 1870, and was educated in the public schools of Detroit, Mich.

He began his shop experience in the drafting department of the Detroit Blower Company, under the management of his father. He was later engaged by the engineering firm of Gilbert Wilkes Company, Detroit, leaving their employ in 1897 to open an office for the Wickes Boiler Company, in Milwaukee, Wis.

The following year, Mr. Huyette returned to Detroit, to take charge again of the business of the Gilbert Wilkes Company in the absence of Mr. Wilkes, who was commander of the Detroit Naval Militia during the Spanish-American War. Upon the return of Mr. Wilkes after the war, Mr. Huyette went back to his work with the Wickes Boiler Company, and opened their branch office in Chicago. He continued as manager of that office until his death, January 11, 1909.

His engineering work was chiefly on boiler installations, and he also designed and patented a gas engine, and designed and built steel sail boats and motor boats.

He was a member of the National Association of Stationary Engineers, and of the Chicago Yacht Club.

EDWARD L. JENNINGS

Edward Lobdell Jennings, whose death occurred on November 6, 1908, was born in North Wayne, Me., April 14, 1850. He received his education in the public schools of his native town and at the Maine Wesleyan Academy. He was apprenticed to the North Wayne Tool Company, and in 1872 he went to Boston and entered the employ of W. A. Wood & Co., and was their manager for several years. He resigned his position with this company and removed to Waterbury to take the position of purchasing agent for the American Brass Co., held for the eight years preceding his death.

Mr. Jennings was a member of the Waterbury Club, a Commandery Mason and a member of the First Church (Congregational), Waterbury, Conn.

EDWARD H. JONES

Edwin Horn Jones was born in Wilkes-Barre, Pa., April 15, 1844, and died December 2, 1908. He was educated at the Old Dow Academy on South Franklin Street, and at an early age he entered the employ of his father, Richard Jones, who then conducted the Jones Foundry, the foundation of the present extensive Vulcan Iron Works. He learned the iron business thoroughly, advanced to superintendent of the works, and at the time of his father's death in 1873 became general manager of the company and later its president, which position he held until the time of his death.

As president of the Vulcan Iron Works he consolidated the Wyoming Valley Manufacturing Company and the Pittston Iron Works with the original plant and later purchased the Tamaqua shops, all of which he consolidated as the Vulcan Iron Works.

In 1891 he was made president and general manager of the Sheldon Axle Works. In 1881 he became director of the Second National

Bank, and later its vice-president. He was interested as stockholder and director in a number of other industries and was an active member of the Wilkes-Barre Board of Trade and one of its trustees. He was a member of the Westmoreland Club and the Wyoming Valley, Country Club, the Art Club, of Philadelphia, and the Sons of the Revolution.

JANG LANDSING

Jang Landsing, whose death occurred July 10, 1909, was born in Heong San, China, October 11, 1864. He was graduated from the Worcester Polytechnic Institute in June 1887, with the degree of B.S.

He was apprenticed to the Washburn Machine Shops for three years, after which he became connected with the Pratt & Whitney Company, as draftsman, leaving after two years to accept a position as machine designer with the Brush Electric Company, of Cleveland.

About 1899 Mr. Landsing, working in cooperation with Mr. Joseph Bijur, assisted in developing a line of special machinery for the manufacture of the "Bijur" type of storage battery. Mr. Landsing became mechanical engineer and then superintendent of the General Storage Battery Company, and was in active charge of the company's large plant at Boonton, N. J., at the time of his death. This work afforded him unusual opportunity to exercise his mechanical ingenuity, and many of the company's successful machines are due to the combination of his fine mechanical judgment and designing skill. He organized and equipped a machine shop for the manufacture of the special tools used by the company and also the 300-kw. power plant. Mr. Landsing had the ability to carry out in an efficient and economical manner work with which he was not previously familiar.

Mr. Landsing was also consulting engineer of the Chinese Legation at Washington.

ROBERT B. LINCOLN

Robert B. Lincoln, president of the Waters Governor Company, Boston, Mass., died June 9, 1909, at his home in Waltham, Mass. Mr. Lincoln began his career in the Globe Works in Boston, afterwards serving throughout the Civil War. In 1868 he went to Cuba as chief engineer of the Maratanza, severing this relationship to become head draftsman at the South Boston Iron Works. In 1882 he designed the compound engine on the Cymbria at East Boston, Mass., and was subsequently connected with E. D. Leavitt of Cambridge, Mass., and later with the Portsmouth Navy Yard, where he remained nine years. At the time of his death, he had been president of the Waters Governor Company, for twenty-seven years, and during his life had held many other positions of trust which were filled with honor and fidelity.

ALEXANDER MILLER

Alexander Miller, head of the firm of Alexander Miller and Brothers of Jersey City, N. J., died May 6, 1909, at his home in New York. Mr. Miller was born in Aberdeen in 1857 and came to this country at an early age. He began his engineering career at the old Delamater Iron Works of New York, of which his father was superintendent. Later he formed a connection with the Deeley Iron Works and specialized in sugar-evaporating machinery. More recently he applied with conspicuous success the experience thus gained to the problem of the evaporation of brine in the manufacture of salt. Three of the New York fire boats were also the product of his works.

Mr. Miller was a leading factor in the organization of the Delamater Veterans Association, composed of his old associates at the Delamater works, and until the present year was its president. He was also a member of Scotia Lodge, F. & A. M., the Engineers Club, New York Athletic Club and the St. Andrew's Society.

ARTHUR WARREN KENDALL PIERCE

Arthur Warren Kendall Peirce died April 13, 1909, at Driehoek, Transvaal, South Africa, where he had been located since 1897. He was born in West Boylston, Mass., November 19, 1873. His preliminary education was acquired in the Plymouth, Mass., high school, and his technical education by home study and experience in the engineering departments of various electrical companies.

In July 1897, he was appointed electrician to the Knights Deep, Ltd., a gold mining company in the Transvaal, and a year later became consulting electrical engineer to the Consolidated Gold Fields of South Africa, Ltd., a corporation controlling a number of mines in the Transvaal, continuing in this position until 1906. After a trip to England he returned to South Africa in April 1907, to accept an appointment with the Victoria Falls Power Co., Ltd., of Germiston, Transvaal, which he retained to the time of his death.

Mr. Peirce was also a member of the American Institute of Electrical Engineers, the South African Association of Engineers, the

Mechanical Engineers' Association of the Witwatersrand, and an associate of the Chemical, Metallurgical and Mining Society of South Africa. He made many extensive trips through the various mining regions of America in search of information regarding hoisting from deep mines, in which work he was professionally interested.

JASPER R. RAND

Jasper Raymond Rand, vice-president and director of the Ingersoll-Rand Company, New York, died in Salt Lake City, Utah, March 30, 1909. He was the son of Jasper Raymond Rand, one of the founders of the Rand Drill Company, and was born in Montclair, N. J., September 3, 1874.

He was graduated from Cornell University in 1898 with the degree of Mechanical Engineer, and served in Porto Rico in the Spanish-American war as a member of the First New York Volunteer Engineers. During 1899–1900 he was president of the Imperial Engine Company at Painted Post, N. Y., which position he left to take the presidency of the Rand Drill Company. In 1905 he was elected vice-president and director of the Ingersoll-Rand Company, the position he held up to the time of his death.

Mr. Rand was a member of the Alpha Delta Phi Fraternity, the Spanish War Veterans, the American Institute of Mining Engineers, the Engineers Club, the Cornell Club and the Alpha Delta Phi Club of New York.

WILLIAM THOMAS REED

William Thomas Reed, was born June 27, 1847, in London, England. He received his education at the Commercial College, Kent, and entered railway service in 1862 as a machinist; was apprenticed to the London, Chatham & Dover Railway, 1869–1871; became a machinist on the Grand Trunk Railway, Canada, in 1871, afterwards serving the road as leading machinist, at Stratford, Ont., 1875–1877, foreman erecting and other shops, at Montreal, P. Q., 1877–1883; locomotive foreman, at Belleville, Ont., 1883–1887.

In 1887 he became master mechanic of the Western division, Canadian Pacific Railway; and from 1888 to 1894, acted as general master mechanic, St. Paul & Manitoba Railway. From 1895 to 1898 he was superintendent of the Chicago Great Western Railway, and from 1893 to 1901, superintendent of motive power and machinery, Seaboard Air Line Railway, Portsmouth, Va. He was locomotive superntendent for the Jamaica Government Railway, from 1902 to 1906, at which time he was appointed locomotive superintendent of the Gold Coast Government Railway, West Africa. He held this position until 1908, when he received another appointment on the Jamaica Government Railway. In 1890 he became a member of this Society.

Mr. Reed died of malarial fever, July 1, 1909, only a few hours after leaving Kingston wharf on leave of absence, and was buried at sea.

EDWIN REYNOLDS

Edwin Reynolds, Past-President of the Society, was born in Mansfield, Conn., March 23, 1831, and was apprenticed at the age of sixteen to A. P. Kenney, a local machinist. At the end of three years apprenticeship he started on a journeyman's tour of various shops in lower New England.

About 1857 Mr. Reynolds went West and for a time was superintendent of the shops of Steadman & Company, of Aurora, Ill., builders of engines, sawmill machinery, drainage pumps, etc., for the Southern trade. As this business practically ceased during the Civil War, he returned to the East and was employed in several machine works until, in 1867, having attracted the attention of George H. Corliss, he was offered a position in the works of the Corliss Steam Engine Company, at Providence, R. I., where he remained as general superintendent until 1877. His last notable work with the Corliss company was the design of a rolling-mill engine to run at 160 r.p.m. a speed double that of previous designs. This engine was installed at the works of the Trenton Iron Company, in 1877. In 1890 a second fly-wheel was added and the speed increased to 180 r.p.m.

In 1877 Mr. Reynolds accepted the position of general superintendent of the Edward P. Allis Company, of Milwaukee, Wis., his first work being to place the company on a paying basis, largely through the development of the Reynolds-Corliss engine. He later developed the building of a varied line of machinery, including large Corliss-engine units for pumping service, mining, air compressing, furnace blast, street-railway work and other purposes. One noteworthy achievement was the building, in 1888, of the first triple-expansion pumping engine for waterworks service. This engine was described by Professor Thurston as "doing continuously so high a duty as to place it among the most remarkable constructions of its class and time, and probably to make its record the highest to date (1894)."

The blowing engine designed by Mr. Reynolds for the original Joliet Steel Company and chosen from among competitive designs submitted by leading engineers in the United States and Europe, embodied a radical departure from accepted practice, but after more than 25 years of continuous experimenting the essential features of this design have not been improved upon.

The well-known story of his sketching the design of the horizontal-vertical four-cylinder compound engine for the Manhattan Railway Company's power house in New York City, while on the train from Milwaukee to New York, illustrates his marvelously quick inventive genius.

At the time of his death, Mr. Reynolds was consulting engineer of the Allis-Chalmers Company, although the illness of the past three years had prevented active duty. He took a great interest to the last in the magnificent new works of the company at West Allis, which had been laid out on a strictly engineering basis, after his plans, at the time the Edward P. Allis Company became merged in the new organization.

He was also interested in a number of other plants of Milwaukee and vicinity, had been the first president of the National Metal Trades Association, and bore the degree of LL.D., conferred upon him by the University of Wisconsin, on the wall of whose college of engineering his name has been carved.

Mr. Reynolds served as Vice-President of the Society from 1892–1894, and as President for the year 1902.

His death occurred at his home in Milwaukee February 19, 1909.

RICHARD HERMAN SOULE

Richard Herman Soule was born March 4, 1849, in Boston, Mass. September 25, 1875, he entered the service of the Pennsylvania Railroad, where he remained for eight years. He held this position for two years until promoted to the test department. Two years later, in 1879, he was made superintendent of motive power of the Northern Central Railway.

In 1881 and June 1882, he was superintendent of motive power of the Philadelphia and Erie division of the Pennsylvania Railroad, and in June 1882 accepted a position in the same capacity with the Pittsburg, Cincinnati and St. Louis Railway.

In 1883, when the West Shore Railway enterprise was carried through, its managers secured the best talent available in the country for their managing officers, and Mr. Soule was appointed superintendent of motive power, a position which he held until the absorption of the West Shore Line by the New York Central in 1887. From February 1887, to April, 1888, he was general manager of the New York, Lake Erie and Western Railroad, and in November, 1888, he was appointed general agent of the Union Switch and Signal Company. He was engaged in the introduction of modern interlocking and lock-signaling plants until 1891. From 1891 to 1897 he was superintendent of motive power of the Norfolk and Western Railroad, and did much to put the rolling-stock of the system, which was then coming into prominence as an important coal-carrying road, on a thoroughly sound basis.

For the next two years, Mr. Soule was in the employ of the Baldwin Locomotive Works, spending nearly a year traveling in foreign countries. He had charge of the Chicago office of this company for a year and a half.

In 1900 he opened an office in New York as a consulting mechanical engineer and practiced until, on account of ill health, he was forced to retire from active business.

Mr. Soule was a member of the Master Car-Builders Association; and author of a report on the standards of this association, which led to a radical change in the association's practice, and to a placing of the standards on a much higher basis. He was also a member of the American Railway Master Mechanics Association. He was one of the managers of this Society, from 1898 to 1901.

He was universally respected and esteemed for his many sterling qualities, which caused his acquaintance to be highly prized by his associates. In all parts of the country men are found who testify to the help given them early in life by Mr. Soule, to whom they owe much of their later success. His memory will live long in the hearts of those to whom he had endeared himself.

Mr. Soul's death occurred at his residence in Brookline, Mass., December 13, 1908.

ARVY ELROY WELLBAUM

Arvy Elroy Wellbaum was born February 12, 1881, at Brookville, Ohio, where he attended the high school. He studied at the Ohio Northern University, Ada, O., for one year, and received the degree of M.E. in 1902 from Ohio State University. During the summer vacations he was in the employ of the C. & G. Cooper Company, Mt. Vernon, O., and Platt Iron Works Co., Dayton, O.

In 1902 he became draftsman for the Morgan Engineering Co., Alliance, O. He became connected with the Foos Manufacturing Co., Springfield, O., in 1903, as designing draftsman, and in 1905 he accepted a similar position with the Foos Gas Engine Co., Springfield, O. For three years he was instructor of mechanical drawing and machine design in the Young Men's Christian Association of Springfield, O. Up to the time of his death, August 31, 1908, he was associated with The Hydraulic Press Company, Mt. Gilead, O.. having had charge of the engineering department.

GEORGE W. WEST

George Washington West died at his home in Middletown, N. Y., December 24, 1908. He was born April 3, 1847, at Troy, N. Y., and received his early education in the public schools of that city. In 1865 he entered the service of the New York Central & Hudson River Railroad at Schenectady, as machinist, and was later made foreman and master mechanic, leaving this position to accept a similar one with the West Shore.

In 1886, he entered the employ of the New York, Lake Erie and Western, now the Erie, as master mechanic of the Mahoning division, was later transferred to the main shops at Meadville, Pa., and in 1888 to the Eastern division. From 1891 until the time of his death he held the position of superintendent of motive power of the New York Ontario and Western.

Mr. West was a past-president of the American Railway Master Mechanics Association, a member and past-president of the New York Railroad Club, and past-president of the Central Railway Club. He was a member of the Masonic order and the order of Elks. The George W. West Association of Engineers at Carbondale was named for him. He was also a director of the First National Bank of Middletown, president of the Ontario and Western Savings and Loan Association, a member of the Middletown Club and a member of the Board of Water Commissioners.

ALFRED R. WOLFF

Alfred R. Wolff, who died at his home in New York on January 7, 1909, was born in Hoboken, March 15, 1859. He entered the Stevens Institute of Technology with its class of 1876, when less than fourteen years of age. He nevertheless easily carried the studies of the four years course, and was recognized as one of the leading students of a strong class.

His graduating thesis on windmills, which contributed original experimental data to the theory of the subject, was published through several numbers of the Engineering and Mining Journal of 1876, with favorable editorial comment. This thesis, supplemented by a compendium of modern American windmills, with tabular statements of their power and relative economy in practice, was published in 1885 by Wiley & Sons, in book form, under the title, The Wind Mill as a Prime Mover, and remains the only book on this subject.

After graduation, Mr. Wolff entered the office of the late C. E. Emory, then consulting steam engineer of New York and also consulting engineer to the U. S. Revenue Marine Service.

About 1880, Mr. Wolff decided to build up a practice as consulting engineer in New York. For about eight years his work consisted in the miscellaneous commissions of the steam expert. During this time he wrote several articles, among them a paper on The Value of the Study of the Mechanical Theory of Heat, presented at the first meeting of The American Society of Mechanical Engineers, of which he was a charter member; a series of editorial articles on steam and energy questions, which appeared in The American Engineer, with whose staff Mr. Wolff was connected for several years, and a supplement to Robert Briggs' essay on Steam Heating, published in the Van Nostrand Science Series.

In 1888 Mr. Wolff was engaged to assist the architect of the New York Freundschaft Club to complete its heating and ventilating plant. This engagement proved to be his opportunity to secure a lucrative specialty. At this time the architect depended for the design of the heating and ventilating plant upon the largely gratuitous plans and specifications of the prospective contractor and consequently the heating and ventilating requirements of a building were liable to be sacrificed in undue proportion to their importance. Obviously there was field for a middleman as the authorized agent of the architect.

In establishing himself in this field, Mr. Wolff encountered many difficulties which he overcame so successfully that at the end of six years he had referred to him more problems of heating and ventilating than he could execute. It was during this time that he wrote and published the pamphlet entitled The Ventilation of Buildings, an outline of the elements of physics, chemistry, and mechanics involved in the design of a heating and ventilating plant.

He introduced, from the German practice, in 1893, the "heat-unit system," under which the radiator surfaces for direct heating in a building are systematically calculated from the heat lost by the various thicknesses of walls and proportion of window surface. This was a substitute for the crude American rule, previously in vogue. allowing a square foot of radiator surface per various cubic feet of room contents, depending entirely on the judgment of the engineer.

He made popular the use of the combined plenum and exhaust system operating with "tempered" air, for ventilation, supplemented by direct radiators to supply the loss of heat by walls and windows[1] as a substitute for the "switch-damper" method of heating and ventilating formerly prevailing in the metropolitan district.

He introduced the thermostat in high-class residence work in 1893; and in 1902, he stimulated Johnson to apply this mechanism to the automatic control of humidity. This automatic "humidostat" was first successfully applied in the Carnegie residence.

Mr. Wolff introduced the use of the cheese-cloth filters, for straining the dust out of the air drawn into a building for indirect heating, into the metropolitan district in 1894, for use in the New York Life Building. He installed a heating and ventilating plant in the Board Room of the New York Stock Exchange in which the problems of constant temperature and constant hydrometric conditions were successfully met. The refrigerating feature of this plant is the only example of the artificial control of summer heat in an office building, and as such it is a unique monument to Mr. Wolff's ability.

Mr. Wolff took part in the organizations of the Ethical Culture Society, especially in the maintenance of their charitable schemes. He was also an alumni trustee of the Stevens Institute of Technology from 1893 to 1896 and a permanent trustee after 1900.

Among a great many important buildings in which Mr. Wolff installed heating and ventilating plants are the following: Century Club; Waldorf-Astoria Hotel; Carnegie Music Hall; Lakewood Hotel, Lakewood, N. J.; United Charities Building; University of the City of New York; New York Herald Building; C. Vanderbilt residence; Teachers College; J. J. Astor residence; St. Regis Hotel; the Lying-in Hospital; Princeton Library; Brooklyn Institute; Columbia University; Sherry's Hotel; Delmonico's Hotel; University Club; Hotel Martinique; American Museum of Natural History; Cornell Medical College; Carnegie residence; Library of J. P. Morgan, Esq.; Hispanic Museum; Evening Post Building; Plaza Hotel.

[1] Mr. Wolff illustrated the application of this combination together with the "heat unit system," in a lecture before the Franklin Institute in 1894, which was published in pamphlet form under the title The Heating of Large Buildings.

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NOTE

1 Names of authors and discussors and headings of groups of similar subjects are in CAPS and SMALL CAPS.
2 The straight title of each paper is in *Italics*. The cross index of a title is in CAPS and lower case.
3 The Society is not responsible as a body for the statements of fact or opinion in its papers and discussions.

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