TRANSACTIONS

OF THE

AMERICAN SOCIETY

OF

MECHANICAL · ENGINEERS.

VOL. XII.

XXIId Meeting, Richmond, Va., November, 1890.
XXIIId Meeting, Providence, R. I., June, 1891.

NEW YORK CITY:
PUBLISHED BY THE SOCIETY,
From the Library Building,
No. 12 West 31st Street,
1891.

OFFICERS

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

1890—1891.

PRESIDENT.

Robert W. Hunt...Chicago, Ill.

VICE-PRESIDENTS.

Joel Sharp...Salem, Ohio.
Geo. W. Weeks...Clinton, Mass.
De Volson Wood...Hoboken, N. J.

Term expires at Annual Meeting of 1891.

Stephen W. Baldwin...New York City.
Jno. F. Pankhurst...Cleveland, Ohio.
Alexander Gordon...Hamilton, Ohio.

Term expires at Annual Meeting of 1892.

MANAGERS.

Frank H. Ball...Erie, Pa.
Geo. M. Bond...Hartford, Conn.
Wm. Forsyth...Aurora, Ill.

Term expires at Annual Meeting of 1891.

Jas. E. Denton...Hoboken, N. J.
Carleton W. Nason...New York City.
H. H. Westinghouse...Pittsburgh, Pa.

Term expires at Annual Meeting of 1892.

Andrew Fletcher...New York City.
Worcester R. Warner...Cleveland, Ohio.
Coleman Sellers, Jr...Philadelphia, Pa.

Term expires at Annual Meeting of 1893.

TREASURER.

Wm. H. Wiley...New York City.

SECRETARY.

F. R. Hutton..No. 12 West 31st St., and 50th St. and 4th Ave., New York City.

PAST OFFICERS.

(EXECUTIVE.)

R. H. THURSTON	*President*	April 7th, 1880—Nov. 3d, 1882.
E. D. LEAVITT, JR.	"	Nov. 3d, 1882—Nov. 3d, 1883.
JOHN E. SWEET	"	Nov. 3d, 1883—Nov. 7th, 1884.
J. F. HOLLOWAY	"	Nov. 7th, 1884—Nov. 13th, 1885.
COLEMAN SELLERS	"	Nov. 13th, 1885—Dec. 2d, 1886.
GEO. H. BABCOCK	"	Dec. 2d, 1886—Dec. 1st, 1887.
HORACE SEE	"	Dec. 1st, 1887—Oct. 18th, 1888.
HENRY R. TOWNE	"	Oct. 18th, 1888—Nov. 22d, 1889.
OBERLIN SMITH	"	Nov. 22d, 1889—Nov. 14th, 1890.
LYCURGUS B. MOORE	*Treasurer*	April 7th, 1880—Dec. 2d, 1881.
" " "	*Acting-Sec'y*	April 7th, 1880—Nov. 4th, 1880.
THOS. WHITESIDE RAE	*Secretary*	Nov. 4th, 1880—March 1st, 1883.
CHAS. W. COPELAND	*Treasurer*	Dec. 2d, 1881—Nov. 7th, 1884.

MEMBERS OF PREVIOUS COUNCILS.

VICE-PRESIDENTS.

HENRY R. WORTHINGTON, COLEMAN SELLERS, ECKLEY B. COXE, Q. A. GILMORE, WM. H. SHOCK, ALEX. L. HOLLEY, F. A. PRATT, W. P. TROWBRIDGE, E. D. LEAVITT, JR., CHAS. E. EMERY, JOHN FRITZ, HENRY MORTON, WM. METCALF, S. B. WHITING, A. B. COUCH, W. R. ECKHART, J. V. MERRICK, CHARLES W. COPELAND, OLIN LANDRETH, HENRY R. TOWNE, C. H. LORING, HORACE SEE, ALLAN STIRLING, JOS. MORGAN, JR., C. T. PORTER, HORACE S. SMITH, W. S. G. BAKER, H. G. MORRIS, C. J. H. WOODBURY, THOS. J. BORDEN, WM. KENT, and CHAS. B. RICHARDS.

MANAGERS.

W. P. TROWBRIDGE, T. N. ELY, J. C. HOADLEY, WASHINGTON JONES, WM. B. COGSWELL, F. A. PRATT, CHAS. B. RICHARDS, S. B. WHITING, J. F. HOLLOWAY, GEO. W. FISHER, ALLAN STIRLING, GEO. H. BABCOCK, S. W. ROBINSON, JNO. E. SWEET, R. W. HUNT, CHAS. T. PORTER, C. J. H. WOODBURY, W. F. DURFEE, OBERLIN SMITH, C. C. WORTHINGTON, WM. LEE CHURCH, WM. HEWITT, C. H. MORGAN, H. A. HILL, WM. KENT, S. T. WELLMAN, F. G. COGGIN, J. T. HAWKINS, T. R. MORGAN, SR., S. W. BALDWIN, FRED'K GRINNELL, and MORRIS SELLERS.

NOTE.

THE increasing bulk of the annual volume of *Transactions* has induced the Publication Committee to discontinue the insertion of the full list of members among the preliminary matter therein. The list which would appear is that which was published under date of July 1, 1891, as the second edition of the Twelfth Catalogue. The following summary records the members in each grade:

SUMMARY.

Honorary Members	18
Life Members	13
Members	1,119
Associates	52
Juniors	142
Total	1,344

AMENDED.

RULES

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

[*Adopted November 5th*, 1884.]

OBJECTS.

ART. 1. The objects of the AMERICAN SOCIETY OF MECHANICAL ENGINEERS are to promote the Arts and Sciences connected with Engineering and Mechanical Construction, by means of meetings for social intercourse and the reading and discussion of professional papers, and to circulate, by means of publication among its members, the information thus obtained.

MEMBERSHIP.

ART. 2. The Society shall consist of Members, Honorary Members, Associates and Juniors.

ART. 3. Mechanical, Civil, Military, Mining, Metallurgical and Naval Engineers and Architects may be candidates for membership in this Society.

ART. 4. To be eligible as a *Member*, the candidate must have been so connected with some of the above-specified professions as to be considered, in the opinion of the Council, competent to take charge of work in his department, either as a designer or constructor, or else he must have been connected with the same as a teacher.

ART. 5. *Honorary Members*, not exceeding twenty-five in number, may be elected. They must be persons of acknowledged professional eminence who have virtually retired from practice.

ART. 6. To be eligible as an *Associate*, the candidate must have such a knowledge of or connection with applied science as qualifies him, in the opinion of the Council, to co-operate with engineers in the advancement of professional knowledge.

ART. 7. To be eligible as a *Junior*, the candidate must have been in the practice of engineering for at least two years, or he must be a graduate of an engineering school.

The term "Junior" applies to the professional experience, and not to the age of the candidate. Juniors may become eligible to membership.

ART. 8. All Members and Associates shall be equally entitled to the privileges of membership. Honorary Members and Juniors shall not be entitled to vote nor to be members of the Council.

ELECTION OF MEMBERS.

ART. 9. Every candidate for admission to the Society, excepting candidates for honorary membership, must be proposed by at least three members, or members and associates, to whom he must be personally known, and he must be seconded by two others. The proposal must be accompanied by a statement in writing by the candidate of the grounds of his application for election, including an account of his professional experience, and an agreement that he will conform to the requirements of membership if elected.

ART. 10. All such applications and proposals must be received and acted upon by the Council at least thirty days before a regular meeting, when the Secretary shall at once mail to each member and associate, in the form of a letter ballot, the names of candidates recommended by the Council for election.

ART. 11. Any member or associate entitled to vote may erase the name of any candidate, and may, at his option, return to the Secretary such ballot enclosed in two envelopes, the inner one to be blank and the outer one endorsed by the voter.

ART. 12. The rejection of any candidate for admission as member, associate, or junior, by *seven* voters, shall defeat the election of said candidate. The rejection of any candidate for admission as honorary member by *three* voters shall defeat the election of said candidate.

ART. 13. The said blank envelopes shall be opened by the Council at any meeting thereof, and the names of the candidates elected shall be announced in the first ensuing meeting of the Society, and also in the first ensuing list of members. The names of candidates not elected shall neither be announced nor recorded in the proceedings

ART. 14.—Candidates for admission as honorary members shall not be required to present their claims; those making the nominations shall state the grounds therefor, and shall certify that the nominee will accept if elected. The method of election in other respects shall be the same as in case of other candidates.

ART. 15. All persons elected to the Society, excepting honorary members, must subscribe to the rules and pay to the Treasurer the initiation fee before they can receive certificates of membership. If this is not done within six months of notification of election, the election shall be void.

ART. 16. The proposers of any rejected candidate may, within three months after such rejection, lay before the Council written evidence that an error was then made, and if a reconsideration is granted, another ballot shall be ordered, at which thirteen negative votes shall be required to defeat the candidate.

ART. 17. Persons desiring to change the class of their membership shall be proposed in the same form as described for a new applicant.

FEES AND DUES.

ART. 18. The initiation fees of members and associates shall be $15, and their annual dues shall be $10, payable in advance. The initiation fee of juniors shall be $10, and their annual dues $5, payable in advance. A junior, being promoted to full membership, shall pay an additional initiation fee of $5. Any member or associate may become, by the payment of $150 at any one time, a life member or associate, and shall not be liable thereafter to annual dues.

ART. 19. Any member, associate or junior, in arrears may, at the discretion of the Council, be deprived of the receipt of publications, or stricken from the list of members, when in arrears for one year. Such person may be restored to membership by the Council on payment of all arrears, or by re-election after an interval of three years.

OFFICERS.

ART. 20. The affairs of the Society shall be managed by a Council, consisting of a President, six Vice-Presidents, nine Managers and a Treasurer, who shall also be the Trustees of the Society.

All past (Ex) Presidents of the Society, while they retain their membership therein, shall be known as Honorary Councillors, and shall be entitled to receive notices of all meetings of the Council and may take part in any of its deliberations; they shall be entitled to vote upon all questions except such as affect the legal rights or obligations of the Society or its members.

ART. 21. The members of the Council shall be elected from among the members and associates of the Society at the annual meetings, and shall hold office as follows:

The President and the Treasurer for one year; and no person shall be eligible for immediate re-election as President who shall have held that office for two consecutive years; the Vice-Presidents for two years and the Managers for three years; and no Vice-President or Manager shall be eligible for immediate re-election to the same office at the expiration of the term for which he was elected.

ART. 22. A Secretary, who shall be a member of the Society, shall be appointed for one year by a majority of the members of the Council at its first meeting after the annual election, or as soon thereafter as the votes of a majority of the members of the Council can be secured for a candidate. The Secretary may be removed by a vote of twelve members of the Council, at any time after one month's notice has been given him by a majority of its members to show cause why he should not be removed, and he has been heard to that effect. The Secretary may take part in any of the deliberations of the Council, but shall not have a vote therein. His salary shall be fixed for the time he is appointed by a majority vote of the Council.

ART. 23. At each annual meeting, a President, three Vice-Presidents, three Managers and a Treasurer shall be elected, and the term of office of each shall continue until the end of the meeting at which their successors are elected.

ART. 24. The duties of all officers shall be such as usually pertain to their offices or may be delegated to them by the Council or by the Society. The Council may, in its discretion, require bonds to be given by the Treasurer.

ART. 25. The Council may, by vote of a majority of all its members, declare the place of any officer vacant, on his failure for one year, from inability or otherwise, to attend the Council meetings, or to perform the duties of his office. All such vacancies and those occurring by death or resignation shall be filled by the appointment of the Council, and any person so appointed shall hold office for the remainder of the term for which his predecessor was elected or appointed; *provided* that the said appointment shall not render him ineligible at the next annual meeting.

ART. 26. Five members of the Council shall constitute a quorum; but the Council may appoint an Executive Committee, or business may be transacted at a regularly called meeting of the Council, at which less than a quorum is present, subject to the approval of a majority of the Council, subsequently given in writing to the Secretary and recorded by him with the minutes. Absent members of the Council may vote by proxy upon subjects stated in the call for a meeting, said proxy to be deposited with the Secretary.

ART. 27. The President on assuming office shall appoint a Finance Committee and a Publication Committee and a Library Committee of five members each. The appointment of two members of each Committee shall expire at the end of each year. The Secretary shall, *ex officio*, be a member of all three Committees.

ART. 28.—The Finance Committee shall have power to order all ordinary or current expenditures, and shall audit all bills therefor. No bill shall be paid except upon their audit. When special appropriations are ordered by the Society, they shall not take effect until they have been referred to the Council and Finance Committee in conference.

ART. 29. It shall be the duty of the Publication Committee to receive all papers contributed, to decide which shall be published in the *Transactions*, and which shall be read in full at the meetings.

ART. 30. It shall be the duty of the Library Committee to take charge of the collection of all material for the Library of the Society, and to supervise all regulations for its use.

ELECTION OF OFFICERS.

ART. 31. At the regular meeting preceding the annual meeting a nominating committee of five members, not officers of the Society, shall be appointed, and this committee shall, at least thirty days before the annual meeting, send to the Secretary the names of nominees for the offices falling vacant under the rules. In addition to such regularly appointed committee, any other five members or associates, not in arrears, may constitute an independent nominating committee, and may present to the Secretary, at least thirty days before the annual meeting, all the names of such candidates as they may select. All the names of such independent nominees shall be placed upon the ballot list with nothing to distinguish them from the nominees of the regular committee, and the Secretary shall at once mail the said list of names to each member and associate in the form of a letter ballot, it being understood that the assent of the nominees shall have been secured in all cases.

Art. 32. In the election of Vice-Presidents, each member and associate may cast as many votes as there are Vice-Presidents to be elected. He may give all these votes to one candidate, or distribute them among more, as he chooses. Managers shall be voted for in the same way.

Art. 33. Any member or associate entitled to vote may vote by retaining or changing the names on said list, leaving names not exceeding in number the officers to be elected, and returning the list to the Secretary—such ballot inclosed in two envelopes, the inner one to be blank and the outer one to be indorsed by the voter. No member or associate in arrears since the last annual meeting shall be allowed to vote until said arrears shall have been paid.

Art. 34. The said blank envelopes shall be opened by tellers at the annual meeting, and the person who shall have received the greatest number of votes for the several offices shall be declared elected.

MEETINGS.

Art. 35. The annual meeting of the Society shall be held on the first Thursday in November of each year, in the City of New York, unless otherwise ordered, at which a report of proceedings and an abstract of the accounts shall be furnished by the Council. The Council may change the place of the annual meeting, and shall, in that case, give timely notice to members and associates.

Art. 36. Other regular meetings of the Society shall be held in each year at such time and place as the Council may appoint. At least thirty days' notice of all meetings shall be mailed by the Secretary to members, honorary members, associates and juniors.

Art. 37. Special meetings may be called whenever the council may see fit; and the Secretary shall call a special meeting at the written request of twenty or more members. The notices for special meetings shall state the business to be transacted, and no other shall be entertained.

Art. 38. Any member, honorary member or associate may introduce a stranger to any meeting; but the latter shall not take part in the proceedings without the consent of the meeting.

Art. 39. Every question which shall come before the Society shall be decided, unless otherwise provided by these rules, by the votes of a majority of the members and associates present, provided there is a quorum.

ART. 40. At any regular meeting of the Society thirteen or more members and associates shall constitute a quorum.

ART. 41. Unless otherwise ordered, papers shall be read in the order in which their text is received by the Secretary. Before any paper appears in the *Transactions* of the Society a copy of the paper shall be sent to the author, and, so far as possible, a copy of the reported discussion shall be sent to every member who took part in the same, with requests that attention shall be called to any errors therein.

ART. 42. The Society shall claim no exclusive copyright in papers read at its meetings, nor in reports of discussions, except in the matter of official publication with the Society's imprint, as its *Transactions*. The Secretary shall have sole possession of papers between the time of their acceptance by the Publication Committee and their reading, together with the drawings illustrating the same; and at the time of such reading, or as soon thereafter as practicable, he shall cause to be printed, with the authors' consent, copies of such papers, "subject to revision," with such illustrations as are needed for the *Transactions*, for distribution to the members and for the use of technical newspapers, American and foreign, which may desire to reprint them in whole or in part. The policy of the Society in this matter shall be to give papers read before it the widest circulation possible, with the view of making the work of the Society known, encouraging mechanical progress, and extending the professional reputation of its members.

ART. 43. The author of each paper read before the Society shall be entitled to twelve copies, if printed, for his own use, and all members shall have the right to order any number of reprints of papers at a cost to cover paper and printing; *provided*, that said copies are not intended for sale.

ART. 44. The Society is not, as a body, responsible for the statements of fact or opinion advanced in papers or discussions, at its meetings; and it is understood that papers and discussions should not include matters relating to politics or purely to trade.

AMENDMENTS.

ART. 45. These rules may be amended, at any annual meeting, by a two-thirds vote of the members present; *provided*, that written notice of the proposed amendment shall have been given at a previous meeting.

CONTENTS OF VOLUME XII.

* Revised and reprinted in final form from Vol. XI.

LIST OF ILLUSTRATIONS—VOL. XII.

PAPERS

OF THE

RICHMOND MEETING

(XXIIId)

NOVEMBER 11th TO 14th, 1890.

BEING ALSO THE ELEVENTH ANNUAL MEETING OF THE SOCIETY.

CCCCXIV.*

PROCEEDINGS

OF THE

RICHMOND MEETING

(XXIId)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS,

November 11th to 14th, 1890.

LOCAL COMMITTEE OF ARRANGEMENTS:—MESSRS. E. F. C. DAVIS, *Chairman;* A. H. Raynal, *Secretary;* Archer, Brooks, Burgwyn, Delaney, Mellin, Greenwood, Sherrell, and Simpkin.

FIRST DAY. TUESDAY, NOVEMBER 11, 1890.

The XXIId meeting of the American Society of Mechanical Engineers was also its Eleventh Annual Meeting, and was convened in the city of Richmond, Va., the sessions being held in the Assembly Hall, in connection with the Exchange Hotel, of that city.

The opening session was called to order by Mr. E. F. C. Davis, Chairman of the Committee of Arrangements, who introduced his Honor Mayor J. T. Ellyson, of Richmond.

After an appropriate address of welcome by the mayor, the retiring president, Oberlin Smith, delivered his annual address, entitled "The Engineer as a Scholar and a Gentleman."

The tellers to count the ballots cast for officers of the Society for the ensuing year were appointed under Article 34, and the session adjourned.

In the evening a reception was tendered to the Society at the house of Governor Philip McKinney, of Virginia, in the historic mansion in Capitol Square.

SECOND DAY. WEDNESDAY, NOVEMBER 12.

The first session for business was convened in the Assembly Hall at half past nine o'clock. The Secretary's register in

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

headquarters showed the following members in attendance during the convention:

Adams, Wm. H.................................New York City.
Alberger, Louis R.............................New York City.
Allison, Robert...............................Port Carbon, Pa.
Almond, Thos. R...............................Brooklyn, N. Y.
Albree, Chester B.............................Allegheny, Pa.
Ashworth, Daniel..............................Pittsburgh, Pa.
Archer, Edward R..............................Richmond, Va.
Babbitt, Geo. R...............................Providence, R. I.
Ball, Frank H. (*Manager*)....................Erie, Pa.
Bang, Henry A.................................New York City.
Barnes, Abel T................................Jamaica Plain, Mass.
Barr, William M...............................Philadelphia, Pa.
Barrus, Geo. H................................Boston, Mass.
Beach, Chas. S................................Bennington, Vt.
Bennett, Frank M..............................Washington, D. C.
Bond, Geo. M. (*Manager*).....................Hartford, Conn.
Bray, Chas. W.................................Youngstown, Ohio.
Britton, J. W.................................Cleveland, Ohio.
Brooks, Wm. B.................................Richmond, Va.
Brotherhood, Fred.............................Richmond, Va.
Bulkley, Henry W..............................New York City.
Campbell, Andrew C............................Waterbury, Conn.
Carr, C. A....................................Washington, D. C.
Coleman, Isaiah B.............................Elmira, N. Y.
Comly, George M...............................Edgemoor, Del.
Cooper, John H................................Philadelphia, Pa.
Cullingworth, Geo. R..........................New York City.
Dallett, W. P.................................Philadelphia, Pa.
Darling, Edward...............................Pawtucket, R. I.
Dashiell, Benj. J., Jr........................Baltimore, Md.
Davis, E. F. C................................Richmond, Va.
Delaney, Alexander............................Richmond, Va.
Denton, James E. (*Manager*)..................Hoboken, N. J.
Dick, John....................................Meadville, Pa.
Doran, Wm. S..................................New York City.
Draper, T. W. Morgan..........................Norfolk, Va.
Eberhardt, F. L'H.............................Newark, N. J.
Engel, Louis G................................Brooklyn, N. Y.
Field, Cornelius J............................New York City.
Fladd, Frederic C.............................New York City.
Foster, Charles E.............................Washington, D. C.
Gantt, Henry L................................Philadelphia, Pa.
Gobeille, Jos. Leon...........................Cleveland, Ohio.
Graves, Erwin.................................Camden, N. J.
Gray, G. A....................................Cincinnati, Ohio.
Greenwood, P. F...............................Richmond, Va.
Grimm, Paul H.................................Glen Cove, L. I., N. Y.
Hand, Frank Ludlam............................Philadelphia, Pa.

Hand, S. Ashton..............................Philadelphia, Pa.
Hawkins, John T..............................Taunton, Mass.
Hemenway, F. F..............................New York City.
Henderson, Alexander..............................Boston, Mass.
Henderson, Geo. R..............................Roanoke, Va.
Hillard, Chas. J..............................Pittsburgh, Pa.
Holland, John..............................Dover, N. H.
Holloway, J. F. (*Past-President*)..............................New York City.
Hunt, Charles Wallace..............................New York City.
Huston, Chas. L..............................Coatesville, Pa.
Hutton, Frederic R. (*Secretary*)..............................New York City.
Hyde, Chas. E..............................Bath, Me.
Jacobi, Albert W..............................New York City.
Jacobus, D. S..............................Hoboken, N. J.
Jewett, L. C..............................Erie, Pa.
Jones, Willis C..............................Cincinnati, Ohio.
Laforge, Fred'k Henry..............................Waterbury, Conn.
Laird, John A..............................St. Louis, Mo.
Lane, Harry M..............................Cincinnati, Ohio.
Leonard, Samuel A..............................Washington, D. C.
Lipps, Henry, Jr..............................Raleigh, N. C.
Low, Fred. R..............................New York City.
McBride, James..............................Brooklyn, N. Y.
McElroy, Samuel..............................New York City.
MacFarland, Walter M..............................Washington, D. C.
McKinney, Robt. C..............................Hamilton, O.
Mattes, William F..............................West Superior, Wis.
Mellin, Carl J..............................Richmond, Va.
Mirkil, Thos. H., Jr..............................Philadelphia, Pa.
Moore, D. G..............................Elizabeth, N. J.
Mumford, Edgar H..............................Detroit, Mich.
Nason, Carleton W. (*Manager*)..............................New York City.
Painter, William..............................Baltimore, Md.
Parker, Chas. D..............................Worcester, Mass.
Parks, Edward H..............................Providence, R. I.
Passel, George W..............................Cincinnati, Ohio.
Phillips, Geo. H..............................Newark, N. J.
Philp, C. V..............................Bethlehem, Pa.
Pickrell, James M..............................Richmond, Va.
Raynal, Alfred H..............................Richmond, Va.
Redwood, I. I..............................Brooklyn, N. Y.
Reiss, Geo. T..............................Hamilton, Ohio.
Roberts, William..............................Waltham, Mass.
Rogers, Winfield S..............................Troy, N. Y.
Rood, Vernon H..............................Luzerne Co., Pa.
Ross, Edward L..............................Indian Orchard, Mass.
See, Horace (*Past President*)..............................New York City.
Shirrell, David..............................Richmond, Va.
Simpkin, William..............................Richmond, Va.
Smith, Geo. H..............................Providence, R. I.
Smith, Jesse M..............................Detroit, Mich.

Smith, Oberlin (*President*)	Bridgeton, N. J.
Smith, Scott A.	Providence, R. I.
Snell, Henry I.	Philadelphia, Pa.
Spaulding, H. C.	Boston, Mass.
Sperry, Charles	Queens Co., L. I., N. Y.
Stillman, Francis Hill	New York City.
Stirling, Allan	New York City.
Svenson, John	Scranton, Pa.
Swasey, Ambrose	Cleveland, Ohio.
Thomson, John	New York City.
Tompkins, S.	Charlottesville, Va.
Torrey, Herbert Gray	New York City.
Tucker, Wm. B.	Elizabeth, N. J.
Vanderbilt, Aaron	New York City.
Webster, John H.	Boston, Mass.
Wellman, Samuel T.	Cleveland, Ohio.
Wheeler, F. Meriam	New York City.
White, William, Jr.	Pittsburgh, Pa.
Whitehead, Geo. E.	Providence, R. I.
Whitney, Baxter D.	Winchendon, Mass.
Whitney, W. M.	Winchendon, Mass.
Wightman, D. A.	Allegheny, Pa.
Wilcox, John F.	Pittsburgh, Pa.
Wiley, Wm. H. (*Treasurer*)	New York City.
Wood, W. H.	Philadelphia, Pa.
Wood, Walter	Philadelphia, Pa.
Woolson, Orosco C.	Newark, N. J.
Wolcott, Frank P.	Carteret, N. J.

There was also a number of guests present and a large delegation of ladies.

The first order of business was:

THE ANNUAL REPORT OF COUNCIL.

The Council would beg leave to present its Annual Report as follows:

It has held seven meetings during the year for the transaction of business, and the following is a summary of its action, in addition to the usual routine of labor:

The committee proposed, in a resolution of Henry R. Towne, at the annual meeting of 1889, to consider the advisability of establishing in this country, what has been called provisionally "An Institute or Academy of Engineering," was duly appointed by the Council, to consist of Messrs. Towne, Thurston, and Sellers, of this Society, and at their request three members were appointed from each of the other Engineering Societies to confer with them.

The Society of Civil Engineers appointed Messrs. Brush, Collingwood, and Michaelis.

The Institute of Mining Engineers appointed Messrs. McDonald, Rölker, and Kirchhoff; and the Institute of Electrical Engineers appointed Messrs. Martin, Upton, and Crocker.

The committee of this Society reported several times during the year the steps which they had taken in connection with the individual members of the other committees, and a draught for a proposed scheme for such an organization was prepared by the committee of this Society, and submitted, confidentially, for criticism.

The Association of Engineering Societies of Chicago appointed a committee to prepare an address to the Engineering Societies of the United States, also with reference to the formation of a National Engineering Society. This committee, since March 12, 1890, has consisted of Messrs. J. B. Johnson, Wm. B. Knight, and B. Williams.

On April 25, 1890, the committee of the Institute of Mining Engineers wrote: "We feel that we cannot pledge the Institute to the conclusions reached by your deliberations, or that we could add much of value to the latter on the specific points considered in the proposals."

On June 10, 1890, the Society of Civil Engineers, through its Secretary, reported that they deemed the consideration of the proposed organization inexpedient at the present time.

Whatever, therefore, may be planned hereafter by the committee of this Society will obviously be upon different lines from those proposed when the committee was first constituted.

The committee of the Society on "Standards" received from the Council an appropriation, not to exceed $250, for the furtherance of the work intrusted to them.

The committee appointed to express to the mayors of the leading American cities their sympathy with the movement to hold a World's Fair in the near future, presented a report at the meeting in May, 1890, which will be found in Volume XI.

The Society has moved during the year from its former quarters at 64 Madison Avenue to the house owned by the Mechanical Engineers' Library Association at No. 12 West Thirty-first Street The steps leading to this action have been very carefully considered at special meetings of the Council, and the action was only consummated after very careful advice taken from interested members of the Society. The Society rents from the Mechanical Engineers' Library Association the parlor floor, which it uses as an office and members' rendezvous, and the auditorium at the rear. On the floor below, it occupies the front room as a stenographer's office and writing-room for the transaction of office routine; and the large room in the rear, under the auditorium, has been fitted up as a banquet or collation room, for social purposes in connection with reunions in the auditorium above.

The Society also has the privilege of the use of certain rooms on the third and fourth floors, as bachelor apartments or sleeping rooms for members of the Society, at the rate of $1.50 per night, or $10 per week for longer occupancy. While these rooms are plainly furnished, the beds have been made particularly luxurious; and it is the desire of the Council that the members of the Society coming to New York should make the house their home during their stay in the city. This move was particularly intended to secure for the non-resident membership benefits in the occupancy of the house which it is hoped they will appreciate.

The Society Library is rented to the trustees of the Library Association, who maintain it as a free public library, and it is open to every one between the hours of ten in the morning and ten in the evening. The evening opening of the library and building has proved itself a conspicuous success, and will be continued indefinitely, except upon the evenings of the Fourth of July, Thanksgiving, and Christmas.

The coöperation of this Society was requested through its Council in the matter of extending its courtesies to the Iron and Steel Institute of Great Britain, and the cognate organization from Germany which made its visit with them to the United States.

Besides the coöperation extended by individuals, the Council put its house quarters at the service of the Institute before they left England, and repeated these offers on the arrival of the party in this country. Quite a number of them made it a point to visit the Society's house during their stay.

The Council has received from the committee managing the details of the Cincinnati convention a gift of the surplus remaining in their hands after all the expenses of the meeting had been paid. This gift, amounting to $218.90, the Council were requested either to apply to the library or the house fund. The Council directed that it should be turned over to the library trustees to meet the obligations which they had incurred.

In response to an overture from the committee of the Western Society of Engineers of Chicago, a committee of this Society was appointed to confer with the representatives of other societies, in reference to holding an international congress of engineers, and providing an international headquarters during the Columbian Exposition. William Forsyth, Jesse M. Smith, Henry B. Stone, R. H. Thurston, and Henry R. Towne were appointed such committee, and the report of this joint committee will be presented to the Society at a later point in the meeting.

The Hon. B. F. Tracy, Secretary of the United States Navy, requested that the Council of this Society should appoint a commission of three to decide upon the relative merits of the designs of lathes for the new gun factory of the Ordnance Bureau. That commission was appointed, to consist of John E. Sweet, S. T. Wellman, and Charles H. Morgan, who met in Washington, and performed its delicate duty to the satisfaction of all concerned.

The Council has appointed a committee of the Society to report upon standard methods for testing the efficiency of locomotives, to consist of William Forsyth, Chairman; F. W. Dean, J. W. Cloud, A. S. Vogt, Allan Stirling, J. E. Denton, and R. H. Soule.

The memorial adopted by the Society's committee, and approved at the May meeting of 1890, to memorialize the United States Government in regard to a suitable monument or other recognition of the important services to the nation of Captain Ericsson, late member of the Society, have directed that the memorial be printed and circulated where it would do good, and the Council have directed that an engrossed letter of transmittal of that memorial to the Secretary of the Navy be sent to him, in the furtherance of the work of the committee.

The issue of a second edition of the first three volumes of this Society's Transactions has been decided upon, the present membership having subscribed for an amount in excess of two-thirds of the expenses of such re-issue. These first three volumes will therefore be issued during the year and printed from plates, so that there will be no difficulty in the matter of subsequent issues.

The Council has directed that its Library Committee have power to expend, for the purchase of books, such proportion of the fund in the savings-bank to the credit of the Library Association as the discretion of the committee may indicate. That sum is now in excess of $2,100, as will appear from the report of the Finance Committee; and the Council, while thanking the members for the suggestions which they have made as to desirable books, will be glad to have a wider interest taken in this matter by members other than those from whom they have heard.

A fine life-sized portrait of the late Alexander L. Holley, deceased founder of the Society, and created honorary member in perpetuity, has been presented to the Society by Mrs. Mary H. Bunker, formerly Mrs. A. L. Holley. This portrait will be hung upon the walls of the assembly room of the Society, and the Council has directed that at an early day suitable ceremonies of unveiling be held in the Society's house, to give expression to the sentiment of recognition for the gift and of the distinction which the donor has conferred.

By the kindness of Mr. James Dredge, honorary member of this Society, a portrait of Sir Henry Bessemer, who was elected honorary member of the Society by the Council at its meeting, April 20, 1882, has been presented to the Society, and is also hung upon its walls.

The Council have directed that a suitably engrossed minute be sent to Sir Henry by the executive of this Society, copies of which are appended below.

The Council has had presented to it the question of the advisability of the issue of a handbook of engineering similar to that issued by the cognate society in Germany. They have directed that the question of the expediency of such procedure be presented to the Society under the head of new business at this meeting.

The Council, in addition to the routine of scrutiny of applications, has considered favorably a number of requests to present sets of its Transactions to libraries of technical schools in this country and abroad, and is now considering the question of increasing its exchange list, particularly with French societies and organizations.

The Council has passed favorably during the year upon 204 applications for membership in the several grades.

The losses by death during the year since the last annual report have been as follows:

Fred. B. Rice,	Hector V. Havemeyer,	Chas. A. Ashburner,
Horatio Allen,	Gustave Adolph Hirn,	Chas. B. Smith,
Henry J. Davison,	Edward H. Owen, Jr.	

The present membership of the Society, including those joining at this meeting and favorably acted upon by the voting members, is distributed among the grades as follows:

Honorary members	16
Life members	10
Members	1,025
Associates	52
Juniors	117
Total	1,220

The Council would also present the report of the tellers of election as follows:

The undersigned were appointed a committee of the Council to act as tellers, under Rule 13, to count and scrutinize the ballots cast for and against the candidates proposed for membership in the Society of Mechanical Engineers before the XXIId meeting of the Society in November, 1890.

They would report that they have met upon the designated days in the office of the Society and proceeded to the discharge of their duties.

They would certify, for formal insertion in the records of the Society, to the election of the appended named persons to their respective grades upon lists Nos. 3 and 4, respectively green and blue.

There were 427 votes cast in the ballot upon the green list, of which 23 were thrown out because of informalities.

STEPHEN W. BALDWIN, } *Tellers.*
CARLETON W. NASON, }

There were 452 votes cast in the ballot upon the blue list, of which 34 were thrown out because of informalities.

STEPHEN W. BALDWIN, } *Tellers.*
CARLETON W. NASON, }

November 7, 1890.

The lists are appended below.

MEMBERS.

Adams, Wm. H.	New York City.
Adsit, John O.	Hornellsville, N. Y.
Andrews, Geo. C.	Minneapolis, Minn.
Archer, Edward R.	Richmond, Va.
Bailey, Cyrus	Akron, O.
Bartlett, John E. T.	Navy Yard, B'klyn, N. Y.

Berg, P. T. .... Homestead, Pa.
Bixby, Edgar M .... Boston, Mass.
Blake, John H .... New York City.
Boyden, N. N. .... Macon, Ga.
Breckenridge, L. P. .... Bethlehem, Pa.
Brotherhood, Fred. .... Beaufort, S. C.
Bull, Goold H. .... Cambridgeport, Mass.
Bull, Storm. .... Madison, Wis.
Cake, H. W .... Lake Linden, Mich.
Carnegie, Andrew .... New York City.
Christie, E. W. .... Jersey City, N. J.
Christensen, Chas. C .... Chicago, Ill.
Colahan, Chas .... Cleveland, O.
Crooker, Ralph, Jr. .... Pittsburgh, Pa.
Davis, Wm. C .... Denver, Col.
Dripps, Wm. A. .... Philadelphia, Pa.
Ferguson, Geo. R. .... Brooklyn, N. Y.
Foster, Chas. F. .... St. Louis, Mo.
Freeman, F. J .... Warren, O.
Fry, Chas. A. .... New York City.
Gandy, Fred'k. .... Hamilton, O.
Greenwood, P. F. .... Richmond, Va.
Grover, Lewis C .... Hartford, Conn.
Hammond, Richard .... Buffalo, N. Y.
Henderson, Geo. R. .... Roanoke, Va.
Hobart, Frank G. .... Beloit, Wis.
Hoffecker, W. L .... Elizabeth, N. J.
Holman, M. L .... St. Louis, Mo.
Hoppes, John J. .... Springfield, O.
Houston, Chas. Robb. .... Cincinnati, O.
Jackson, D. C. .... New York City.
Jarvis, Chas. M. .... E. Berlin, Conn.
King, Wm. R. .... New York City.
Leonard, Sam'l H. .... Washington, D. C.
Lyon, J. Wyckoff. .... Brooklyn, N. Y.
Mackiewicz, Victor. .... New York City.
Marshall, Geo. .... Dayton, O.
Marshall, Robt. E. .... Wilmington, D. C.
Mason, Frank S. .... Jersey City, N. J.
Maury, Dabney H. .... New York City.
Maxwell, James R. .... Cincinnati, O.
Melcher, Chas. W. .... St. Louis, Mo.
Morison, Geo. S. .... Chicago, Ill.
Morton, Geo. L .... Washington, D. C.
Newell, Augustus. .... Chicago, Ill.
Padgham, Frank W. .... Oil City, Pa.
Painter, Wm. .... Baltimore, Md.
Philp, C. v. .... Bethlehem, Pa.
Pickrell, Jas. M. .... Richmond, Pa.
Pike, Wm. A. .... Minneapolis, Minn.
Pollock, James. .... Wilkesbarre, Pa.

Porter, William ....Chester, Pa.
Potis, Salvator, Jr.............................Chicago, Ill.
Reed, W. I.............................St. Paul, Minn.
Richards, Chas. R.............................Brooklyn, N. Y.
Schmid, Albert.............................Pittsburgh, Pa.
Sheldon, Wm. H.............................New York City.
Simpson, Wm. L.............................Philadelphia, Pa.
Smith, Robert W.............................Wilmington, Del.
Spaulding, H. C.............................Boston, Mass.
Suppes, Max. M.............................Johnstown, Pa.
Swenson, Wathier.............................New York City.
Thackray, Geo. E.............................Johnstown, Pa.
Theil, Chas.............................Chicago, Ill.
Tobey, Wm. L.............................Boston, Mass.
Vivian, Simon.............................Brooklyn, N. Y.
Wales, Chas. M.............................New York City.
Wallace, Wm.............................Ansonia, Conn.
Whitney, Edwin H.............................Providence, R. I.
Wright, Ernest N.............................Boston, Mass.

ASSOCIATES.

Foster, Chas. E.............................Washington, D. C.
Foster, Rufus J.............................Scranton, Pa.
Roux, Paul.............................Paris, France.
Stratton, W. H.............................Providence, R. I.
Ware, Justin A.............................Worcester, Mass.

JUNIORS.

Bailey, Chas. L.............................Washington, D. C.
Bardwell, A. F.............................Stamford, Conn.
Bissell, Geo. W.............................Ithaca, N. Y.
Buchanan, A. W.............................St. Louis, Mo.
Chamberlin, Paul M.............................Waynesboro, Pa.
Cushman, C. G.............................Roanoke, Va.
Grist, James E.............................Philadelphia, Pa.
Hollingsworth, Loftus, Jr.............................Holyoke, Mass.
Inowye, Yasumaro.............................Japan.
Johnson, Arthur E.............................Stamford, Conn.
Lockwood, E. H.............................New Haven, Conn.
Shaw, Edwin C.............................Binghamton, N. Y.
Shepherd, Wm. G.............................Trenton, N. J.
Snyder, Robert M.............................Wilkesbarre, Pa.
Trask, Geo. F. D.............................Louisville, Ky.
White, Ambrose H.............................Trenton, N. J.

PROMOTION TO FULL MEMBERSHIP.

Bailey, W. H.............................New York City.
Edwards, Victor E.............................Worcester, Mass.

The Council would further report that an invitation has been received from the members resident in the city of Providence, R. I., inviting the Society to hold its spring meeting, or XXIIId convention, in that city during the month of June, 1891. It has been thought advisable that the annual meeting in the autumn of 1891 should be held in the Society's house in New York, to introduce its facilities and attractiveness to the membership in due form.

Respectfully submitted

By the Council.

As appendix to report of the Council were presented the following copies of minutes referred to therein:

The Council of the American Society of Mechanical Engineers desire, on behalf of that Society, to express their indebtedness to Mrs. Mary H. Bunker for the kind feelings which prompted her to present to the Society the well-executed portrait of her late husband, Alexander Lyman Holley, one of its founders and most steadfast friends.

While the occasion is one which will not permit an extended review of the efforts of Mr. Holley, in the early formation of the Society, we cannot but congratulate its members, now that they are so pleasantly installed in a house of their own, in having hung on their walls the portrait of one whose earnest wish was for its growth and success, and whose pleasant face will serve as a reminder of the past, as well as an inspiration for the future.

In tendering the donor, on behalf of ourselves and our associates, our warmest thanks for this most treasured gift, we hold ourselves as among those who will ever keep in pleasant remembrance the name of Alexander Lyman Holley, honorary member in perpetuity, deceased founder of the Society.

By order of the Council,

OBERLIN SMITH, *President.*
F. R. HUTTON, *Secretary.*

NEW YORK, *October*, 1890.

TO THE HON. BENJAMIN F. TRACY, SECRETARY OF THE NAVY, WASHINGTON, D. C.

SIR:—By direction of this Society, a committee, especially appointed for the purpose, and consisting of friends of the late Captain Ericsson, and officers or past officers of the Society, has prepared the accompanying memorial, asking that Congress take action looking toward the proper acknowledgment of the great indebtedness of our country, in its time of greatest need, to that great inventor and engineer.

This memorial is intended to present, as well as language permits, a statement of the earnest desire on the part of the members and the colleagues of that distinguished man, that some fitting monument be erected in memory of the man and the engineer, to testify to the gratitude which his adopted country feels toward him; a permanent memorial of the people and the Government of the United States; erected, not to immortalize a name already too well known to be forgotten, but to give never-failing testimony to the fact that our gratitude is equally enduring.

In compliance with the suggestion implied in the communications of the department to the chairman of the committee, and by special direction of the Council of this Society, this memorial is now transmitted to the Navy Department, with the request on the part of its framers, and of the undersigned, representing the Society, that it be forwarded to the President, to be by him communicated to Congress ; preferably with a special message, together with such suggestions or recommendations as your department may see fit to offer.

The work of Ericsson was done for the Navy Department directly, and it is, as you have already remarked, fitting that this document should be passed through that channel, that it may have the indorsement and reënforcement of that branch of the Government to which his great work was most important.

We remain, sir, with greatest respect,

Your most obedient servants,

OBERLIN SMITH, *President.*
F. R. HUTTON, *Secretary.*

TO SIR HENRY BESSEMER, LONDON, ENGLAND.

DEAR SIR : The Council of the American Society of Mechanical Engineers acknowledge with great pleasure, on behalf of that Society, their obligations to you for the portrait of yourself which they have just received through the kindness of Mr. James Dredge, of London.

The American Society of Mechanical Engineers have but recently established themselves in their own house, in the city of New York, and they are now engaged in decorating and embellishing the same, with the hope that for long years to come it will be the home, and as well the centre of interest, of the Mechanical Engineers of the United States.

Among the regrets of the engineers of our country is that fact that you could not have made a visit, in connection with the other members of the British Iron and Steel Institute of Great Britain, and have seen for yourself, and known as eye-witness, something of the extent and growth of this comparatively young land. This marvellous growth is, we are glad to think, to a large extent due to the important invention which bears your honored name.

While the written thanks of a society whose members may doubtless be somewhat unknown to you may not, of itself, convey much of pleasure, we cannot but hope that there will come to you something of delight and satisfaction in the thought of the fact that your portrait on our walls, three thousand miles away, will be gazed upon by a vast number of engineers, who hold in kindly appreciation one who, through difficulties and disappointments and trials which would have shaken the faith and made despondent many another, has persevered, upheld by an inspiration unfelt by others, until at last triumphant, you won, not only for your countrymen, but for all the world, an inestimable boon, and for yourself a renown which naught else could equal.

It is with this hope and belief we send you our hearty thanks for your kind remembrance, and to which we add our own earnest wish that you may live long to enjoy, not only the honors of our motherland, but as well the respect of engineers the world around.

By order of Council,

OBERLIN SMITH, *President.*
F. R. HUTTON, *Secretary.*

At the close of the report by Council, the second order of business was the report of the Finance Committee of the Society, presented as follows:

The Finance Committee of the American Society of Mechanical Engineers would respectfully report to the Council the following statement of the receipts and expenditures on behalf of the Society, under their direction, during the Society year from November, 1889, to November, 1890:

ANNUAL REPORT

Receipts.

Initiation Fees	$2,595 00
Current Dues	10,162 08
Past Dues	547 70
Advance Dues	167 71
Sales of Publications	658 21
Binding	503 69
Library, Permanent	85 00
Library, Current	395 00
Badges	714 58
Engraving	118 01
Life Membership	800 00
Sinking Fund	808 50
Profit and Loss	5 00
Rent	347 50
Work of Committees	5 00
Interest in Savings-Bank	35 08
Balance, November, 1889	277 34
	$17,720 40

Expenditures.

General Printing and Stationery	$1,617 71
Printing Transactions	4,761 15
Postage	635 59
Library	164 55
Salaries	3,913 25
Office Expenses	429 42
Engraving	1,057 30
Contingencies	61 25
Binding	470 70
Meetings	1,041 75
Work of Committees	167 72
Furniture and Fixtures	686 89
Badges	695 25
Interest Deposited in Savings-Bank	35 08
Travelling	38 00

Brought forward	$15,775 61
Savings-Bank (Library account)	315 45
Insurance	24 00
Rent	1,568 31
Balance, November 1, 1890	37 03
	$17,720 40

There also remains uncollected dues from members to the amount of $176, from nine members resident in this country and four resident abroad. There is no doubt of the collectibility of this sum, and the committee would call the attention of the membership to the fact that there has been inaugurated the practice of drawing upon the members for their dues when eleven months overdue.

The amount in previous years uncollected at the end of the fiscal year has almost always been in excess of $700, and the Society is to be congratulated upon the healthy and interesting condition of the organization, when out of a membership of nearly twelve hundred, when this report is made, there should be only nine persons behind in their annual dues.

There stands also to the credit of the Society, for its Library Fund, the sum of $2,163.48, deposited in savings-banks and drawing interest.

Respectfully submitted

By the Finance Committee.

The next order of business was the report of the Society's Committee on Library, presented as follows:

REPORT OF LIBRARY COMMITTEE.

While the incorporation of the Mechanical Engineers' Library Association, acting in the interest and sympathy with the Society of Mechanical Engineers in the development of an engineering library, may perhaps appear to render of less moment the work of the Library Committee of the A. S. M. E., yet it is the intention of those in authority in both organizations that their work should be completely distinct, and that there should be no abatement in the efforts of the Library Committee to secure, in particular, gifts of books to the library, and an increase of the fund for current expenses, for binding of periodicals, exchanges, etc., which will necessarily draw upon the finances of both organizations.

While the Council of the Society of Mechanical Engineers has directed that the books under the control of its Library Committee should be loaned to the trustees of the Library Association, that arrangement is purely a business matter, such loan being a part of the consideration passing between the two organizations in the form of a rent charge for the space occupied in the library building.

The plan outlined in the original report, in Vol. VI. of the Transactions, page 11, has been continued through the year, and, with the interruptions incident to a change of location, with a reasonable degree of success. Circulars were sent out in the beginning of the year, with the bill for the Society dues, to each member who had not heretofore subscribed. These circulars explained the scheme of the committee, and were accompanied by a form of agreement as to contributions to the library, in one of three forms.

First: A subscription to a permanent fund, in installments, if preferred, for the purchase of books.

To this fund since the last report there have been subscribed from members as follows:

Vincent G. Hazard,	H. H. Scoville.

Second: Subscriptions to an amount of $2 to a fund for current library expenses, binding of periodicals, exchanges, etc., payable as an increase of the dues and at the same time.

To this call there have been responses since last report in Vol. XI. from members as follows:

James Atkins,	W. S. G. Baker,	William Burnham,
D. L. Barnes,	Percy M. Blake,	Francis J. Cole,
Walter L. Clark,	James Christie,	J. J. Dekinder,
W. P. Dallett,	Edward L. Dent,	H. A. Gillis,
Julius S. Hornig,	Edward L. Jones,	W. V. Lowe,
Asa M. Mattice,	Edward F. Miller,	James McBride,
Edward S. Renwick,	David W. Robb,	Edwin Ruud,
Louis Schutte,	Coleman Sellers,	Frank L. Shepherd,
Geo. A. Suter,	Jesse M. Smith,	William W. Snow,
Isaac G. Sowter,	Henry R. Towne,	A. Verastegui,
Ezra J. Whitaker,	Chas. H. Wilcox,	H. H. Westinghouse.

There are therefore now 213 members regularly contributing to this fund, and members not now interested in it are urged to cooperate in the further extension of this plan, and thus induce even more widespread interest in the thoroughness of the library.

The total available annual income from this fund is now $432.

Third: Direct contributions of books, photographs, drawings, and manuscripts of value. Under this subdivision there have been many responses during the year, which the following list is intended to catalogue, and to cover contributions received since last report in Vol. XI.:

From R. H. Davies:
- Link-Motion and Expansion Gear, by N. P. Burgh, Engineer.
- Philosophia Britannica, 1771, 2 vols., by B. Martin.
- A Practical Treatise on Railroads, by Nicholas Wood.
- Britannia and Conway Tubular Bridges, by Wm. Fairbairn, C.E.
- Essays on Millwork and other Machinery, by Robertson Buchanan.

From H. L. Binsse:
- International Centennial Exhibition, 11 vols., by Dorsey Gardner.

From W. C. Lambert:
- The Technologist, Vols. 1 and 2, 1870–71, Industrial Pub. Co., New York.

From W. J. Silver:
- Key to the Universe, by Orson Pratt.
- Current volume of Deseret Weekly.

From B. E. Fernow:
- 3 copies of Tratman's Report on Metal Track.

From R. H. Thurston:
- Patent Office Reports from 1847 to 1855, inclusive.
- Journal of Franklin Institute for 1863–64–65–67, and two numbers of 1860, viz.: July and August.
- Set of Spanish drawings.

From Warren S. Locke:
- The Five Orders of Architecture, 1 vol., by Giacomo Barezzi.

From Aug. W. Colwell:
- Patent Office Reports and Official Gazette, 91 vols., including General Index from 1790 to 1878.

From J. M. Allen:
- Hartford, Conn., as a Manufacturing, Business, and Commercial Centre, 1 vol., by Hartford Board of Trade.

From Mrs. Mary H. Bunker, through Mr. L. G. Laureau:
- 8 years of Zerah Colburn's American Railway Review.
- 2 years of Mann & Holley's Railroad Advocate.

From S. W. Robinson:
- Appendix, 1884–5, Special Reports to the Commissioner of Railroads and Telegraphs of Ohio.

From R. H. Thurston:
Paris/Bibliotheque Scientifique Nationale.
Histoire de la Machine à Vapeur, Vols. 1 and 2, bound.

From J. Hirsch:
Leçons sur les Machines à Vapeur. Paper, Vol. I.
Reports du Jury Internationale, Group 4, Class 54.
Les Machines et les Appareils. De La Mechanique Generale.

From M. J. Hirsch:
Notice sur les Elevateurs et Plas Inclines pour Canaux.
Note sur L'Explosion D'eine Chaudiere à Vapeur.
Frein Continu Systeme Wagner.
Theorie des Machines Aerothermiques.
Rapport Commission Centrale des Mach à Vapeurs.
Rapport Congress Internationale de Mechanique Applique.
Annales des Conservatoire des Arts et Metiers.
Reservoir des Mettersheim Deversoir Syphon.
Rapports Deleguees du Ministre des Tnaux Publics, de France sur les Traaux du Congres.

From J. F. Klein:
The Law of Proportionate Resistances.

From Anon:
Catalogue Junior Engineers Society of London.
Electric Cable Traction.
Journal of Engineer's Society, Lehigh University.

From C. E. Billings:
Hartford, Conn., Board of Trade Circular, 1889, bound.

From De Volson Wood:
Thermodynamics, Second Edition, bound.

From Dwelshauvers-Dery:
La Machine a Vapeur. Pamphlet.

The library still needs Vol. I. of *Engineering* of London, having acquired Vol. II. during the course of the year.

The series of the Journal of the Franklin Institute begins with Vol. XXIX. for 1855, and is complete to the end of Vol. XXXIX., 1860. There is then a gap to Vol. LV., January, 1868, after which the series is complete to date. Members who may be able to supply the missing volumes of this series are cordially urged to interest themselves in doing so.

The following is a *résumé* of the finances of the Library Fund of the Society, from its establishment in 1884 to date:

There has been actually paid in as cash to the Library Permanent Fund and reported in previous reports of the Treasurer and Finance Committee:

For the year 1884–85		$408 40
" " " 1885–86		110 00
" " " 1886–87		145 00
" " " 1887–88		95 00
" " " " (gift of Philadelphia Committee)		206 36
" " " 1888–89		89 00
" " " 1889–90		85 00
" " " " (transfer from current fund)		230 45
" " " " (transfer from current fund)		79 25
Interest account previously rendered	$178 34	
Interest to July 1, 1890	35 08	
		213 42
Total		$1,661 88

To the fund for current expenses the payments have been as follows:

For the year 1884–85		$164 00
" " " 1885–86		254 60
" " " 1886–87		266 52
" " " 1887–88		301 00
" " " 1888–89		336 00
" " " 1889–90		395 00
Total current Expense Fund		$1,717 12
Total Permanent Fund		1,661 88
Grand total		$3,379 00
The sums which were not to be immediately expended were put in savings-banks by order of the committee, and have been there accumulating interest, as the above memorandum indicates.		
The disbursements on account of the Library Funds for the purchase of books and binding of exchanges and periodicals, has amounted in previous years to	$741 27	
Add expenditure this year	164 55	
Total expenditure	$905 82	
Transferred from Current to Permanent Fund	309 70	
		1,215 52
So that in the savings-banks is a balance of, as per the Report of the Finance Committee given elsewhere.		$2,163 48

The Library Committee would also call attention to the *résumé* at the end of its report, of the report of the Mechanical Engineers' Library Association, which is to be presented at its annual meeting in New York shortly, for the information of those who are members of that association.

The following is a list of exchanges which are continually on file in the library:

SOCIETIES, AMERICAN.

American Society of Civil Engineers, New York City.
American Institute of Mining Engineers, New York City.
American Institute of Electrical Engineers, New York City.
Associated Engineering Societies, St. Louis, Mo.
Boston Society Civil Engineers, Boston, Mass.
Society of Arts, Boston, Mass.
Canadian Society Civil Engineers, Montreal, Canada.
Civil Engineers' Association of Kansas, Wichita, Kan.
Engineers' Club of Kansas City, Kansas City, Mo.
Engineers' Society of Western Penna., Pittsburgh, Pa.
Engineers' Club of Phila., Phila., Pa.
Franklin Institute, Phila., Pa.
Indiana Society Civil Engineers and Surveyors, Remington, Ind.
Master Car Builders' Association, New York City.
U. S. Naval Institute, Annapolis, Md.
Technical Society of Pacific Coast, San Francisco, Cal.
American Society of Naval Engineers, Washington, D.C.

SOCIETIES, FOREIGN.

Iron and Steel Institute, London, England.
Institute Engineers and Shipbuilders of Scotland, Glasgow, Scotland.
Institution Civil Engineers of Great Britain, London, England.
Institution Mechanical Engineers of Great Britain, London, England.
Institution Civil Engineers of Ireland, Dublin, Ireland.
Ingenoirs, Forenginens, Forhandlinger, Stockholm, Sweden.
Liverpool Engineering Society, Liverpool, England.
Mining Institution of Scotland, Hamilton, Scotland.
N. E. Coast Inst. Eng. and Shipbuilders, Newcastle-on-Tyne, England.
North of Eng. Inst. of Mining and Mech. Eng., Newcastle-on-Tyne, Eng.
Polytechnic Society of Norway, Kristiana, Norway.
Société des Ingenieurs Civils France, Paris, France.
Annales du Conservatoire des Arts et Metiers, Paris, France.

JOURNALS, AMERICAN.

American Machinist, New York City.
American Engineer, Chicago, Ill.
American Journal of Railway Appliances, New York City.
American Miller, Chicago, Ill.
Boston Journal of Commerce, Boston, Mass.
Chicago Journal of Commerce, Chicago, Ill.
Engineering News, New York City.
Engineering and Mining Journal, New York City.
Electric Power.

Electrical Review, New York City.
Fire and Water, New York City.
Industrial World, Chicago, Ill.
Mechanics, Philadelphia, Pa.
Manufacturers' Gazette, Boston, Mass.
National Car Builder, New York City.
Power, New York City.
R. R. and Engineering Journal, New York City.
Railway News, New York City.
R. R. Gazette, New York City.
Stevens Indicator, Hoboken, N. J.
The Locomotive, Hartford, Conn.
The Locomotive Engineer, New York City.

JOURNALS, FOREIGN.

Architektu' a' Inzenyru', Prague, Bohemia.
Engineering, London, England.
Engineer, The, London, England.
Electric Review, London, England.
Giornal del Genio Civile, Rome, Italy.
Glaser's Annalen, Berlin, Germany.
Indian Engineering, Calcutta, E. I.
Iron, London, England.
Industries, London and Manchester, England.
L'Industria, Milan, Italy.
Practical Engineer, Manchester, England.
Proceedings Royal Tech. Mech. Laboratory of Instr.
Stahl und Eisen, Düsseldorf.

The Transactions of the Society may also be found in the following institutions, to whose libraries they are regularly sent, either as donations or in return for certain publications issued by them :

Stevens Inst. Tech., Hoboken, N. J.
Fisk University, Nashville, Tenn.
Vanderbilt University, Nashville, Tenn.
Royal Technical Institution of Research, Charlottenburg, Germany.
The Yorkshire College, Leeds, England.
Arkansas Industrial University, Fayetteville, Ark.
Bureau of Naval Intelligence, U. S. N., Washington, D. C.
Ohio State University, Columbus, Ohio.
American Institute, New York City.
Rensselaer Polytechnic Institute, Troy, N. Y.
Sibley College, Cornell University, Ithaca, N. Y.
University Library, Cornell University, Ithaca, N. Y.
University of Illinois, Champaign, Ill.
U. S. Naval Observatory, Washington, D. C.
U. S. Patent Office, Scientific Library, Washington, D. C.
U. S. Patent Office Library, London, England.
Massachusetts Inst. of Technology, Boston, Mass. (Society of Arts.)

Conservatoire des Arts et Metiers, Paris, France.
Free Public Library, Worcester, Mass.
Purdue University, Lafayette, Ind.
University College, London.
University of Michigan, Ann Arbor, Mich.
Columbia College Library, New York City.
Lehigh University, Bethlehem Pa.
McGill University, Montreal, Can.
Iowa Agricultural College, Ames, Iowa.
Glasgow and West of Scotland Tech. College, Glasgow, Scotland.
Smithsonian Institute, Washington, D. C.
Verein Deutscher-Eisenhuttenlente, Dusseldorf, Germany.
Mechanic's Institute, San Francisco, Cal.
Sheffield Scientific School, Yale College, New Haven.
Pratt Institute, Brooklyn, N. Y.
University of Wisconsin, Madison, Wis.
Free Public Library, Providence, R. I.
University of Minnesota, Minneapolis, Minn.
University of Tennessee, Knoxville, Tenn.
Washington University, St. Louis, Mo.

The success of the practice inaugurated in 1889, of having the Society library open in the evenings, has been unmistakable, and has had a decided influence in inducing the trustees of the Library Association to continue the practice, which the Society of the Mechanical Engineers inaugurated.

The library building and the offices of the Society are open until ten o'clock every evening, with the exceptions of Sundays, Thanksgiving, Christmas, and the Fourth of July. Whether the evening openings shall also be maintained during the months of July and August will remain a matter of experiment, which the use made of the library hereafter will be allowed to determine.

Respectfully submitted
By the Library Committee.

As appendix to the Library Report was presented the following transcript, altered before publication to state the condition of the finances of the Library Association at the date of its annual meeting at the end of November, 1890 :

SUMMARY OF REPORT OF BOARD OF TRUSTEES OF THE MECHANICAL ENGINEERS' LIBRARY ASSOCIATION.

The Trustees of the Mechanical Engineers' Library Association received the title from the Trustees of the New York Academy of Medicine to the house and lot, No. 12 West Thirty-first Street, at noon on the 8th of May, 1890. The price for the building and lot, with library shelves and a certain amount of furniture, was fixed at $60,000, of which $33,000 was left by the former owners upon a first mortgage, and the balance, $27,000, was paid in cash, contributed for this purpose by the friends of the library movement. This money was raised by the sale of bonds, covered by a second mortgage, held in the interest of the bondholders by the "Title Guaranty and Trust Company," of New York.

In order to adapt the house for the uses to which the trustees designed it, there had to be considerable outlay made in the way of equipment and decoration, including the working over of the plumbing of the house, painting, papering, and kalsomining, from the fourth floor to the basement. The arrangements which the trustees had made to lease the fourth floor and basement to another organization fell through without great disappointment to the trustees, inasmuch as it left these desirable parts of the house free for other uses; and the only tenants of the library building are now the American Society of Mechanical Engineers, and the American Institute of Electrical Engineers, and certain individuals to whom furnished apartments are let, which they are occupying as bachelor quarters.

A description of the house and its arrangements has been already published, and need not be repeated.

The summary of the receipts and expenditures of the trustees from May to November, 1890, is appended below:

Receipts.

Loan on 1st Mortgage New York Academy of Medicine.		$33,000 00
Loans by Bonds covered by Second Mortgage		31,800 00
Rent of Offices		1,643 31
Rent of Rooms		109 62
Sinking Fund		5 00
Bills Payable		1,000 00
Fellowship Fund		45 00
Equipment Account		218 90
		$67,821 83

Expenditures.

House and Lot, No. 12 West 31st Street:		
Purchase Money Mortgage		$33,000 00
Cash		27,000 00
Equipment Account:		
Furniture	$1,066 41	
Decoration and Plumbing	4,331 69	
Repairs to old Work	53 15	
		5,451 25

Brought forward	$65,451 25
Contingencies, Searches, Fees, etc.	1,435 21
Interest on Mortgage, July 1	215 60
Insurance, three years	48 00
Lighting, Electricity, and Gas	144 11
Fuel	159 00
Janitor's Salary	221 00
Janitor's Supplies	105 36
Laundry	19 56
Balance, November 28, 1890	22 74
	$67,821 86

The Society Committee on "Standard Flanges" reported progress, as did the Committee on "Uniform Methods of Test," but had no formal report to make.

The Committee on "Standards" reported by a telegraphic despatch from its chairman, James W. See, the usual Congressional delays.

The Tellers to count the ballot for the officers presented the following report, which was accepted and ordered on file:

The Tellers of election to count the ballots for officers of the Society for the ensuing year present the following report :

November 11, 1890.

Whole number of Ballots cast		465
President	R. W. Hunt	463
Vice-President	S. W. Baldwin	459
"	J. F. Pankhurst	462
"	A. Gordon	467
Treasurer	W. H. Wiley	465
Managers	A. Fletcher	459
"	W. R. Warner	461
"	C. Sellers, Jr.	466

Respectfully submitted,

J. H. Webster, } *Tellers.*
G. H. Smith. }

Geo. H. Barrus, Chairman of Committee on "Standard Methods of Conducting Duty Trials of Pumping Engines," read a discussion of that report by R. H. Thurston, F. W. Dean, A. F. Hall, William Kent, John R. Freeman, A. M. Wellington, John E. Codman, and his own reply thereto, after which additional matter was presented by Messrs. Jacobus, Denton, F. M. Wheeler, W. M. Barr, and J. T. Hawkins.

Upon the question of the acceptance of the report, it was explained that the Society's policy in accepting such technical reports and proposed standards was to direct their acceptance and incorporation in the Transactions as the committee's mature opinion, but that the Society did not thereby make itself responsible, nor did it formally adopt the standards recommended. The report was, on motion, accepted and ordered on the record.

Notice of amendment to Article 31 of the Rules, by D. K. Nicholson, was presented as follows:

TO THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, RICHMOND MEETING, *November*, 1890:

Notice is hereby given, according to the rules, that a motion will be made at the annual meeting in 1891 to amend Art. 31 of the Rules to read as follows: At the regular meeting preceding the annual meeting a nominating committee of five members, not officers of the Society, shall be appointed, and this committee shall, at least thirty days before the annual meeting, send to the secretary the names of *at least two* nominees for each and every office falling vacant under the rules. In addition to such regularly appointed committee, any five members or associates, etc., as per printed rules.

Very respectfully,
DAVID KIRK NICHOLSON,
Steelton, Pa.

November 5, 1890.

This called for no action of the Society at this time, but was merely read under the rules.

The secretary read the report, in the abstract, of the committee appointed by a convention of representatives from the leading engineering societies, in reference to the establishment of an Engineering Headquarters, and of holding an international congress in Chicago during the Columbian Exposition in that city. The complete report is as follows:

REPORT OF CONVENTION HELD IN CHICAGO, OCTOBER 14, 15, 1890, TO CONSIDER THE ESTABLISHMENT OF AN ENGINEERING HEADQUARTERS AND THE HOLDING OF AN INTERNATIONAL ENGINEERING CONGRESS DURING THE WORLD'S COLUMBIAN EXPOSITION, 1893.

The convention met at ten A.M. in the rooms of the Western Society of Engineers, 78 La Salle Street, Chicago, Tuesday, October 14, 1890. The societies represented and the delegates present were as follows:

THE AMERICAN SOCIETY OF CIVIL ENGINEERS:
Wm. P. Shinn (president), C. L. Strobel, A. E. Hunt.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS:
Wm. Forsyth, Jesse M. Smith.

THE AMERICAN INSTITUTE OF MINING ENGINEERS:
Wm. P. Shinn, A. E. Hunt.

CANADIAN SOCIETY OF CIVIL ENGINEERS:
J. D. Barnett, O. Chanute.

THE AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS:
E. M. Izard.

THE ENGINEERS' CLUB OF PHILADELPHIA:
H. W. Spangler, Wilfred T. Lewis, E. V. d'Invilliers.

THE CIVIL ENGINEERS' CLUB OF ST. LOUIS:
J. B. Johnson, E. D. Meier, Robt. E. McMath.

CIVIL ENGINEERS' CLUB OF ST. PAUL:
L. W. Rundlett, W. W. Curtis, S. D. Mason (president).

WISCONSIN ELECTRIC CLUB:
Warren S. Johnson.

ENGINEERING ASSOCIATION OF THE SOUTHWEST:
E. L. Corthell.

CIVIL ENGINEERS' CLUB OF CLEVELAND:
Wm. T. Blunt, John Eisenmann.

ENGINEERS' CLUB OF MINNEAPOLIS:
Wm. A. Pike, F. W. Cappelen.

THE SOCIETY OF CIVIL ENGINEERS, PARIS, FRANCE:
E. L. Corthell.

THE ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA:
A. E. Hunt.

THE WESTERN SOCIETY OF ENGINEERS:
O. Chanute, D. J. Whittemore, E. L. Corthell, C. L. Strobel.

Mr. E. L. Corthell explained the object of the meeting, and the following letters were read by the secretary:

OFFICE OF THE DIRECTOR-GENERAL, WORLD'S COLUMBIAN
EXPOSITION, PULLMAN BUILDING,
CHICAGO, ILL., U. S. A., *October* 9, 1890.

MR. J. W. WESTON, SECRETARY WESTERN SOCIETY OF ENGINEERS, 78 La Salle Street, Chicago.

DEAR SIR: Having been informed by Mr. E. L. Corthell that it is proposed to hold an International Engineering Congress in Chicago in 1893, and that it is desirable that this congress be held under the auspices of the World's Columbian Exposition, I wish to say that the holding of this congress meets with my hearty approval, and that I will further its interest so far as I may be able to do so.

Yours very truly,
[Signed] GEO. R. DAVIS,
Director-General.

SECRETARY'S OFFICE OF THE WORLD'S COLUMBIAN EXPOSITION,
CHICAGO, *October* 11, 1890.

JOHN W. WESTON, ESQ., SECRETARY WESTERN SOCIETY OF ENGINEERS, 78 La Salle Street, Chicago.

MY DEAR SIR: I am informed by Mr. E. L. Corthell, chairman of a committee of your society, on a proposed International Engineering Congress, to be held in this city in 1893, and am now more fully informed by you of the object of this congress. You desire to know if the Directory of the World's Columbian Exposition approves of this congress being held, and of its being held under the auspices of the Exposition, your society making arrangements for the assembling of the congress, and providing for such expense as may be incurred, the Exposition, however, to furnish such suitable building or hall as it may have, or control, for the use of the congress.

The matter was laid before the Directory last evening at its regular meeting, and your wishes in the premises explained.

I am requested to inform you that the Directory heartily approves of your purpose, and will do what it can properly to make the congress a thorough success.

I will be glad to confer with you at any time in that behalf.

With great respect, I have the honor to be,

[Signed] BENJ. BUTTERWORTH,
Secretary.

SOCIETY OF CIVIL ENGINEERS, PARIS, FRANCE.
PARIS, *September* 22, 1890.

MR. PRESIDENT: In reply to yours announcing the convention to be called by the Western Society of Engineers, Chicago, October 14 next, to consider an International Engineering Congress to be held in 1893, at the celebration of the 400th anniversary of the discovery of America, I regret to say your invitation to us arrived too late to advise our colleagues in France in time to enable any of them to report in your city on the date named.

In fact, our society has been on vacation since August, and will not meet again until October.

But we thoroughly believe that our society should be represented at your convention, and we have written to our corresponding member in Chicago, Mr. E. L. Corthell, to accept the office of delegate from the Society of Civil Engineers of France, on that occasion. We shall then be very certain of our representation.

With most sincere wishes for the full success of your important enterprise, and with fraternal good will,

Yours truly,

V. CONTAMIN,
President.

To L. E. COOLEY,
President Western Society of Engineers, Chicago, U. S. A.

After organization and discussion, a committee was appointed to formulate a plan and to report to the convention, which adjourned to meet again next morning.

October 15. The following report was submitted:

CHICAGO, ILL., *October* 15, 1890.

TO THE CHAIRMAN OF THE CONVENTION OF DELEGATES FROM ENGINEERING SOCIETIES OF THE UNITED STATES AND CANADA.

DEAR SIR: Your committee on plan for establishing and maintaining a joint Engineering Headquarters in Chicago in 1893, during the World's Columbian

Exposition, and for holding an International Engineering Congress at some time during the Exposition, beg leave to report:

It finds itself unable to present at this time more than a brief outline plan.

The proposition advanced by the committee of the Western Society of Engineers to this convention yesterday embodies our views, with some changes which we have made in the plan herewith submitted:

First: Engineering Headquarters.

In view of the existence in this country of several large engineering societies of high rank which will desire the use of headquarters for their own members, and for the entertainment of foreign visitors, and the inconvenience and expense which would result from the maintenance of separate establishments, we think it very desirable that all the engineering societies of recognized standing in the United States and Canada be requested to unite in establishing and maintaining a joint Engineering Headquarters during the continuance of the Exposition.

The Exposition management will probably furnish space free of charge within the Exposition buildings, but it may be deemed advisable to provide additional quarters outside; the headquarters to be a rendezvous for all the members of the engineering societies in this country, and their use to be freely tendered to all foreign engineers.

It is expected that the staff shall consist of a joint secretary and two or more assistants, some of whom shall speak the principal European languages. The staff to be charged more especially with:

(*a*) To give information concerning the location of various engineering exhibits within the Exposition.

(*b*) To give visiting and foreign engineers information about points of engineering interest outside of the Exposition, and to aid their investigations in other ways.

(*c*) To give visiting and foreign engineers introduction to those whom they may desire to meet, and to promote social intercourse.

(*d*) To keep a record of the addresses of visitors, and to invite them to the International Engineering Congress hereinafter outlined.

It is estimated that the expense will amount to about $10,000. This, it is suggested, may be met by an assessment of one dollar per member on each engineering society of this country which shall join this proposed association, and also by voluntary contributions. The details to be hereafter adjusted.

It is evident that this plan will be far more economical than that of maintaining separate headquarters by the several societies.

Second: Engineering Congress.

At some time, to be hereafter designated, during the Columbian Exposition, it is proposed to hold within the Exposition, in a building which the management thereof proposes to furnish, an International Engineering Congress open to engineers of all nations. This congress to last six days and to be conducted in the English language.

The opening session of welcome and organization to be a joint session, and if warranted by the attendance and the number of papers offered, the congress then to be divided into sections to consider and discuss the various branches of Civil, Mechanical, Mining, Metallurgical, Electrical, Military, and Naval Engineering.

A chairman and secretary for each section to be designated in advance, and the sessions to be so timed that papers and discussions on allied subjects shall not occur simultaneously so as to preclude those interested from attending several sections.

The congress to terminate with another joint session.

All papers, so far as practicable, to be furnished in advance, to be carefully examined by the proper committees under rules to be hereafter laid down, and, if found acceptable, to be printed for distribution in advance to the members of the congress, at which they are to be chiefly read by title so as to admit of immediate discussion.

Intending contributors to be requested to confine their papers, so far as possible, to such new and recent constructions, machines, processes, methods, experiments, and investigations, including proposed standards of tests and measurement, as are of engineering importance. Papers on purely speculative subjects should not be received.

A small fee (say $2.00) to be paid by members attending the congress, to defray its expenses. The papers and discussions to be subsequently printed and furnished to such members as may so request at a stipulated price.

A Permanent Committee to be chosen in advance, to organize the above proposed headquarters and congress.

Respectfully submitted,

E. L. CORTHELL,
O. CHANUTE,
JESSE M. SMITH,
D. J. WHITTEMORE,
C. L. STROBEL,
W. W. CURTIS,
J. B. JOHNSON.

The report was unanimously adopted in the following resolution :

Resolved, That the report of the committee on an International Congress and joint Headquarters be accepted, and that we report the same to our respective societies, with the recommendation that action in approval or in disapproval of the same be taken within the next two months, and that we desire the present committee to be continued with power to carry on the correspondence and organization until its successor is appointed.

In furtherance of the plan adopted the following resolution was passed by the convention :

Resolved, That it is the sense of this convention that the general permanent committee on International Congress and Engineering Headquarters be composed of one member from each of the societies which shall join in the plan, except that the American Society of Civil Engineers, the American Society of Mechanical Engineers, the American Institute of Mining Engineers, the American Institute of Electrical Engineers, and the Canadian Society of Civil Engineers may each appoint two members, and the Western Society of Engineers may appoint three members of such committee.

The following resolutions were also passed :

Resolved, That the secretary be instructed to prepare minutes of the proceedings of this convention and the resolutions adopted, and that he, as soon as possible, have the same printed and sent to each delegate, and the secretary of each of the societies represented.

That the Executive Committee of the convention be empowered to call the first meeting of the delegates provided for in the resolution adopted, at such time as they may see proper after January 1, 1891.

JOHN W. WESTON,
Secretary.

Jesse M. Smith, the representative of the Society in the Convention, spoke in explanation of its action as follows:

Mr. Jesse M. Smith.—As a member of the committee which was sent to the conference in Chicago, I wish to make a statement. It was understood from the beginning that nothing that was done at the conference should in any wise be binding on any of the societies there represented; that the meeting was simply one for conference, and was held for the purpose of talking over what might possibly be done with a view to a closer relation of the various engineering societies during the World's Fair; and to discuss the possibility of an Engineering Congress being held during the Exhibition and in the Exhibition building. The questions were very fully discussed, as already stated in the report, and the recommendations which have been printed, and read to you by Professor Hutton, were very fully indorsed. It was proposed that a permanent committee, the delegates to which would be appointed by the different societies, would take the place of the committee chosen by the conference which met at Chicago, and that the delegates which were sent by the different societies should be sent with power fully to represent their respective society and to do whatever seemed best for the interests of their society. I have received from Mr. Chanute, who was president of the convention, a statement of the number of societies which would probably be asked to join in the movement. There are the four national American societies, the Society of Civil Engineers of Canada, and sixteen other local societies—that is, having their headquarters in the different prominent cities of the country, New York, Philadelphia, Boston, Chicago, etc. Of these sixteen, the Western Society of Engineers, which sent out the invitation for the conference, is counted as one. It was the sentiment of the conference which was held in Chicago, that, as the Chicago Society is to bear the brunt of the hard work to be done, it seemed no more than fair that it should have a larger representation than the other local societies.

The President.—What is the membership of the Western Society, Mr. Smith?

Mr. Smith.—The Western Society of Engineers is put down on the list sent me by Mr. Chanute as having 318 members; the American Society of Civil Engineers as having 1,013; our own society as having 840, which is evidently a mistake, for it already numbered at that time over 1,000, and at the present time it numbers over 1,200 members. The American Institute of Mining Engineers is put down on the list as having 1,636; the Canadian Society, 265. The local clubs vary in membership from 482, in the Philadelphia club, to as low as 32, in the Civil Engineers of St. Paul. The representation as set forth in this report would stand as follows: The five national societies, two members each—ten members; the fifteen clubs, fifteen members; and the Western Society, three members; making a total of 28 members on the permanent committee. Of course, the permanent committee would divide itself into working committees as it thought proper.

On motion, the report was approved by the Society, and under its acceptance the Council will appoint its permanent committee to represent the organization.

Henry L. Binsse presented the following letter in reference to the issue by the Society of a technical handbook. The proposition received little discussion, and was finally laid on the table.

NEWARK, N. J., *November* 6, 1890.

F. R. HUTTON, ESQ., SECRETARY AMERICAN SOCIETY OF MECHANICAL ENGINEERS, NEW YORK CITY.

DEAR SIR: A year or so ago my attention was called to a German engineers' manual by the remark of a Swedish engineer that it excelled every other book on the subject. An examination of the work disclosed that it differed very widely in its plan from all of our well-known and excellent hand-books.

It is published by the German engineers' society called Hütte, being edited by a committee chosen from the society, and the discussion of each topic has been intrusted to one or more experts in each field of engineering. Twenty-nine names are mentioned in the present edition of those who have added new subject-matter to the work, from which may be estimated the large number of men who have taken part in the work. It stands to reason that the work of so many minds, each one excelling in its special study, should be more complete, more correct, and far more useful than a hand-book written by a single author, no matter how talented and capable he may be. I have thought that the book might be made available to American engineers by a translation; but a short inspection showed that it would have to be reëdited and many chapters rewritten in order to make the book a useful one for this country. The value of a book of this kind needs neither argument nor proof. The book has run through fourteen editions in Germany, and it seems probable to me that a work on a similar plan should meet with an equal success here. I suggest, therefore, that the American Society of Mechanical Engineers might undertake for the United States what the Germans have done for their country, and compile a similar manual, restricting the field, however, to Mechanical Engineering only. It is thought that such a work would not only be a great aid to the advancement of the profession, but that it would add also to the good name of the Society.

It is clear that our own hand-books have outlived a portion of their usefulness, for I am assured that a well-known engineer, a member of this and other societies, has been engaged by a publishing firm to compile a new hand-book, and he is busy with that work at present.

The first and perhaps the strongest objection to the plan is the practical one that our engineers are very busy men, and that it would require from them a sacrifice of time. Have, then, the German engineers a greater love for their profession than we have? Is their professional feeling stronger and are they more willing to sacrifice a part of their time than their American professional brethren? It must be borne in mind also that while the book would represent a great deal of labor when taken altogether, the amount of work of each contributor would not be very large.

It seems to me that the heaviest burden would be borne by the Committee.

I do not wish to wander beyond the limit which I set for myself, which was merely to present the suggestion for the consideration of our members. So I conclude with the earnest wish that this suggestion may bear a useful fruit for the Society.

Very respectfully yours,

HENRY BINSSE.

I append certain details and the table of contents.

The German hand-book is divided into two parts; the first consisting of eight chapters, 771 pages, devoted to mathematics, mechanics, heat, strength of materials, strains in girders and similar structures, machine construction, and motors.

The second part, 573 pages, treats of surveying, railways, architecture, ship-building, iron metallurgy, mill-work, and electricity, to which several tables are added. It will be seen that this field is very large, and that the second part has nothing to do with mechanical engineering, if we except a few chapters like those upon the locomotive and wire-rope transmission. Therefore, the proposed work would be only one-half of the size of the German hand-book. The American manual ought to have a great advantage over the German one by this concentration.

In conclusion, I offer the following table of contents of Part I. from the German hand-book, so that you may form an idea of the scope of the book and its possibilities for usefulness.

FIRST DIVISION, MATHEMATICS, 115 PAGES.

I. Tables.
II. Arithmetic.
III. Trigonometry.
IV. Calculus.
V. Analytics.
VI. Kinematics.
VII. Mensuration.
VIII. Axonometic Projections.

Second Division, Mechanics, 101 Pages.

I. Statics.
II. Dynamics.
III. Hydrostatics.
IV. Hydrodynamics.
V. Aerostatics.
VI Aerodynamics.

Third Division, Heat, 24 Pages.

I. General formula and tables.
II. Mechanical theory of heat.

Fourth Division, Theory of Strains, 74 Pages.

I. General laws and constants.
II. Tension and compression.
III. Shearing.
IV. Bending.
V. Torsion.
VI. Compound strains.
VII. Strength of vessel walls, and flat bodies.
VIII. Springs.

Fifth Division, Statics of Construction, 71 Pages.

I. Calculation for bridge and roof constructions.
II. Earth pressure.
III. Arches.
IV. Choice of safe coefficients.
V. Safe loads for bridges and roofs.

Sixth Division, Parts of Machinery, 132 Pages.

I. Connections, keys, screws, and rivets.
II. Machine parts of rotations, tooth and friction gearing, belt and rope driving, pivots, journals, and rollers, couplings and bearings.
III. Ropes, belts, chains, together with drums and rolls.
IV. Brakes.
V. Machine parts for receiving and conducting fluids ; pump and press cylinders, pipes, and valves.
VI. Pistons, piston-rods, and stuffing-boxes.
VII. Crank motion.
VIII. Regulating machinery, fly-wheels, centrifugal governors.

Seventh Division, Tools, 74 Pages.

I. Machine tools : iron-working tools, wood-working tools.
II. Lifting machinery : pulley blocks, windlasses, cranes, power cranes.
III. Hydraulic lifting machinery : hand power, machine power, hydraulic cranes, direct working plunger lifts.
IV. Hoists : winches, and power hoists.
V. Lifting machinery for fluids : water engines, pumps, and lifting pumps.
VI. Ventilating machinery : fans and blowers.

EIGHTH DIVISION, POWER MACHINERY, 148 PAGES.

I. Animal motors.
II. Water Motors : vertical water wheels, turbines, and turbine construction.
III. Steam engines : calculations, valve motions, and particulars concerning.
IV. Boilers : general data, principal systems and particulars concerning.
Laws : principles and directions for boiler and steam-engine trials.

The size and type of the German manual is about the same as that of Trautwine.

A paper by W. H. Adams, member elected at this meeting, was taken up by special vote of the Society, in advance of other papers, in view of the interest of the paper in connection with the excursion of that afternoon. This paper was entitled "An Engineering Problem at Richmond, Va.," but elicited no discussion.

Two other papers were read before adjournment, by Frank Van Vleck, of San Diego, Cal., entitled "Light Cable Road Construction," and by Professor Thurston, entitled "Authorities on Steam Jackets." The discussion of this latter paper was adjourned until the evening session, the afternoon being devoted to an excursion upon the James River.

THIRD SESSION, WEDNESDAY, NOVEMBER 12, 8 P.M.

Professor Thurston's paper on "Steam Jackets" was discussed by Professor Denton and Mr. Scott A. Smith.

Professor Thurston's paper on "Chimney Draught" was discussed by Professor Denton.

Mr. T. R. Almond read a paper entitled "A Novel Form of Flexible Tubing," and Professor Carpenter's paper on "Heat Transmission through Cast-iron Plates" was discussed by Professor Denton and Mr. W. M. McFarland.

Three papers by Professor Wood, "Some Properties of Ammonia," "Theoretical Investigation of the Efficiency of Vapor Engines," and "Mechanical and Physical Properties of Sulphur Dioxide," and one by Professor Jacobus, "Experimental Determination of the Latent Heat of Ammonia," were presented, but elicited no discussion.

James McBride read a paper on "Automatic Regulation of Injection Water," which was discussed by Messrs. Wheeler, Webster, Grimm, and Engel.

At the close of these papers, the "Topical Question" was taken up: "Is there any reason why corrosion should be more active in one place rather than another inside of a steam-drum properly piped to connect several boilers in a battery?"

This was discussed with blackboard illustrations by Messrs. McBride, Stirling, Babbitt, Nason, Grimm, Engel, Barr, Hawkins, E. F. C. Davis, F. M. Wheeler, Scott A. Smith, G. R. Henderson, and J. W. McElroy.

FOURTH SESSION, THURSDAY, NOVEMBER 13.

The fourth and concluding session for papers was called to order at 9.30 A.M., in the Assembly Hall of Exchange Hotel.

A paper on "Hydraulic Hoisting Plant for Brooklyn Sugar Refinery" was read by L. G. Engel, and that on "Hydraulic Travelling Cranes" received discussion by Messrs. Engel, D. G. Moore, and Huston.

A paper on "Rope Driving," by C. W. Hunt, was discussed by Messrs. Samuel Webber, T. S. Miller, Scott A. Smith, Grimm, and Denton.

John H. Cooper's paper on "Accident-Preventing Devices for Machinery" was discussed by J. L. Gobeille and Oberlin Smith.

Ambrose Swasy's paper on "New Process of Generating and Cutting Teeth of Spur Wheels" was illustrated by samples, and was discussed by Messrs. Kimball, Oberlin Smith, Hawkins, Denton, McFarland, and Tompkins.

A paper by G. W. Bissell, entitled "Interesting Experiments with Lubricants," was discussed by Messrs. Thurston and Denton.

Professor Denton presented two papers: "Performance of Seventy-five-ton Refrigerating Machine," and "Special Experiments with Lubricants," and that by W. A. Bole, on "Single-Acting Compound Engines," received no discussion.

At the close of these papers, the following resolutions were presented and carried unanimously with acclamation, whereupon the president announced the meeting adjourned:

Resolved, That the thanks of the Society be tendered to his Honor J. Taylor Ellyson, mayor of the city of Richmond, for the noble and full-hearted welcome to the hospitality tendered by him in behalf of the citizens and for the freedom of the city so generously given. Our closer acquaintance with our friends of the South has not failed to impress us with the fact that they have entered upon an era of manufacturing prosperity which they richly deserve, and which they are well qualified to carry forward for the betterment of their own city and State, and the country at large, and which prosperity must be due largely to the knowledge, skill, and industry of the American engineers.

Resolved, That the thanks of the Society, and especially of the visiting ladies, are hereby tendered his Excellency Governor Philip McKinney, and his wife, not only for the opportunity of making a most pleasing acquaintance with themselves, as well as other prominent citizens, but for the opportunity of inspecting a mansion so full of historic memories, clustering around the home of a long line of governors of a State that may proudly call herself the Mother of Presidents.

Resolved, That the American Society of Mechanical Engineers desires to convey the thanks of its members and their ladies to Col. C. T. E. Burgwyn and to Mr. Lewis D. Cutshaw for a delightful excursion on the historic James River, and for the opportunity of examining the admirable river improvements under their charge. May the completion of this great work make real the dream of the beautiful city of Richmond of becoming a seaport of the first order. May the sight of Fort Darling and the monitors resting peacefully side by side remind us forcibly that whereas we were for a time divided we are now united more closely than ever before, in one strong patriotic and prosperous nation.

Resolved, That conspicuous among the numerous attentions showered upon us during our visit to Richmond are those by Messrs. P. H. Mayo and brother, and Allen & Ginter, who not only kindly opened to us their vast establishments, in which we were entertained by the melodies so peculiar to the sunny South, but who took especial pains that we should bring away with us mementos which shall hereafter serve as pleasant reminders of an interesting and novel experience in the line of manufacturing nowhere else revealed.

Resolved, That while our visit to this city has been replete with new and interesting sights, we cannot but express ourselves as under special obligation to the Tredegar Iron Company and the Richmond Locomotive Works.

It is in such establishments and amid such industries that the mechanical engineer feels himself quite at home. Our visit to the works of both companies has an added pleasure for the reason that both have a history reaching over many years, and both have contributed during that time much to the importance and value of the iron industries of the South.

The lady guests of the Society are under many obligations to Mr. F. W. Burke and Messrs. A. Hoen & Company for the beautiful and tasteful souvenirs so pleasing to the eye and tickling to the palate, which the gentlemen members were permitted to admire as works of art and fine examples of the many and growing industries of the city of Richmond.

Resolved, That to the Chesapeake & Ohio, Atlantic & Danville, Norfolk & Western, and Richmond, Fredericksburg & Potomac Railways the American Society of Mechanical Engineers desires to render its hearty thanks for the courtesy shown to its members, and for the generous tender of a special train and free transportation to the many places of interest in the vicinity of Richmond.

Resolved, That our Society were greatly pleased for the courteous attention of Mr. Dellie Sutherland in directing their conveyance about the city of Richmond, and desire to render him many thanks.

Resolved, That to Mr. Scott A. Carrington, proprietor of the Exchange and the Ballard hotels, the members of the American Society of Mechanical Engineers express themselves as much pleased with the courtesies extended to them during the meeting in Richmond, and especially for the use of the hall in which the meetings are held.

Resolved, That the American Society is thankful to the Virginia Electric Light & Power Company for the opportunity given its members of visiting the electric stations, and for the return of their watches properly demagnetized.

Resolved, That the society is especially indebted to Mr. T. W. Morgan Draper, our fellow-member here, for the elaborate entertainment proposed for us at Norfolk and vicinity, which cannot fail to be most enjoyable and a notable feature of our Richmond meeting.

Resolved, That the Press of Richmond has placed the American Society of Mechanical Engineers under many obligations for its kindly words of welcome to the city of Richmond, and for its highly intelligent reports of the proceedings of the meetings of the Society.

Resolved, That the American Society of Mechanical Engineers kindly thank the street railway companies of Richmond for the free use of their cars by their members, which has greatly assisted them in seeing the many beauties and places of historic interest of this hospitable city.

Resolved, That the assembled engineers, while unable to thank individually the various firms who have opened their doors to us and invited our attention to their several places of industry, especially desire to thank the Richmond & Danville Railway Company for having done so, and we ask that our inability to name the long line of other persons to whom we are indebted will not be by them understood as indicating on our part a want of appreciation of their courtesies, but rather an indication of our dearth of words to properly express our thanks to them.

Resolved, That the thanks of the Society be tendered to Newport News Dry Dock & Ship Building Company for the interest they have taken to make this first visit of the Society to the South one of pleasure and enjoyment, and the fact that we cannot enumerate the various methods they have adopted to bring this about is the only reason we do not mention them more in detail.

Resolved, That knowing how important and essential to the success of the local committees who have had in charge the entertainment of the American Society of Mechanical Engineers in the various cities we have visited is the coöperation of the citizens thereof, it gives pleasure to ask that our warmest thanks may be accepted by the Citizens' Committee of the Chamber of Commerce, who by their hearty coöperation have honored themselves and the body they so well represent by their efforts to make our visit to the city of Richmond one of pleasure, and our going away one of regret.

Resolved, That the thanks of this Society are due to the local committee of arrangements—Messrs. Davis, Raynal, Archer, Brooks, Burgwyn, Delaney, Miller, Greenwood, Sherrill, and Simpkin, for their taste and appropriateness in providing this beautiful assemblage hall, and particularly for their untiring energy in arranging and providing for the comfort and pleasure of the members in attendance.

Resolved, That we recognize in all the efforts made by the committee for our entertainment, a certain potential emanating from the ladies of Richmond.

Resolved, That we shall leave this city bearing with us a most happy remembrance of the hospitality extended us everywhere.

Excursion Days.

On the afternoon of Wednesday, an excursion barge, convoyed by two tugs, conveyed the party down the James River to visit the historic "Drewry's Bluff." A luncheon was served on the upper deck of the barge, and a stop was made at the United States monitors anchored in the James River. Not a little interest was elicited when it became known that several of the members of the party were connected with these monitors, either in their construction, or in engagements during their history.

Thursday.

Carriages conveyed the members and their ladies, at the close of the fourth session, to the historic Tredegar Iron Works, where they were received by the veteran General Anderson, and conducted to a luncheon served in one of the rooms of the works. In the luncheon apartment was pointed out a drill-press which had been used for the purpose of drilling cartridge shells for the use of the Confederate army. At the close of the luncheon, a touching address was made by General Anderson, to which the president responded, and one or two others who were informally called upon.

After visiting the Tredegar Iron Works in its various departments, the party was conveyed to the Richmond L. & M. Works, where the machinery for the United States battle-ship "Texas" was in process of construction.

At the close of this visit the drivers were instructed to convey those filling the carriages to points of interest in Richmond, including Hollywood Cemetery, the Lee Statue, the old St. John's Church, the Jefferson Davis and Lee mansions, and the residence portion of the city. The ladies had also been previously taken to certain of the cigarette factories, and had been honored by special souvenir gifts from the committee who escorted them.

On the evening of Thursday a social reception was tendered to the members and their ladies by the residents of Richmond, in the Richmond theatre. The seats had been floored over, and the building specially draped, and dancing and a handsome supper completed the evening.

On Friday, a special train on the C. and O. R.R. conveyed the party to Newport News, where they were to accept the hospitalities of the Newport News Dry Dock & Ship Building Company.

A stop was made by the way at old Providence Forge, where matters of historic interest were exhibited, and where the visitors saw the curious cypress growth which manifests itself in the waters of the Dismal Swamp.

After the inspection at Newport News of the new shops and the dry-dock of the company, the party were conveyed to the Warwick Hotel, where the ladies had already been enjoying a part of their luncheon, and a most enjoyable collation was spread for them.

A speech of recognition was made by the president, and the response was by Horace See, consulting engineer for the company.

Leaving the hotel, the party were guided to the steamer wharf, where boats of the Atlantic & Danville Co., under the general charge of Mr. T. W. M. Draper, conveyed the party across the historic waters of the bay to Norfolk and Portsmouth. After a stop at the Portsmouth navy yard, the boats brought the tourists across the Elizabeth River to Norfolk, where a part of them took a special train on the N. & W. Ry., back to Richmond, to make connection by north-bound evening trains, and the rest, after visiting a cotton compress, were carried across to the Hygeia Hotel at Old Point Comfort, from whence they returned at their own convenience.

The railroad company furnished special trains on Friday evening, northward, from Richmond to Washington, for the party who came from Norfolk desiring to make north-bound connections.

CCCCXV.*

PRESIDENT'S ANNUAL ADDRESS.

THE ENGINEER AS A SCHOLAR AND A GENTLEMAN.

BY OBERLIN SMITH, BRIDGETON, N. J.
(President 1889-90.)

FAR back among the ages, in times beyond the ken of History's written page, the young world invented the *Engineer*, as the creator of its coming civilization. He it was who established synthetic methods, and sewed together fig-leaves into a mantle which was a prototype of our textile fabrics, and, in analogous metal patchwork, our steam-boilers and ship-hulls of a later age—one large piece made from many small ones. He it was who, with sticks and puddled clay, established the first order of architecture—Adamesque, if we may so call it. He it was who, before he happened to think of a Pullman car on a steel rail, built the roads and rude wagons which made the dawn of commercial and social life possible. He built the bridges which brought tribes and nations into communion, and helped them to reduce their uncomfortable excess of population by making machines with which they could kill each other.

Throughout the earth, in all ages, the *Engineer* has wrested from nature her well-kept secrets, and has made his non-engineering friends comfortable by showing them how to deal with the material world around them. But for him, as we now know him, practising his art in its present stage of development, we should be set back a century, without railways, or telegraphs, or steam-power manufactories. There would be no electric-lights, nor telephones, nor electric-bells; no sewing-machines, nor gas-fixtures, nor modern plumbing. Our farmers would work with the sickle and the flail; our sailors, as of old, would keep us tossing months, instead of days, upon the sea. Following time logically backward, and robbing each age of its ministering angel, with his

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

acquired knowledge reënforcing the accumulated experience of his predecessors, we would soon arrive at the blackness of social darkness. We can therefore say that were it not for the *Engineer* in history, our fashionable society would probably all be modelled upon that of Terra del Fuego, where an entire wardrobe consists of a piece of fur, held upon the *windward* side of its wearer. Our roofs would be the caves and trees; our food, shellfish and fruits and nuts—good enough dinner courses in their way, but not followed by champagne; our roads would be but foot-trodden paths; our bridges fallen logs; our weapons' stones and clubs.

So much for a cursory negative view of engineering in the past. A positive exposition, in a concrete form, even if for a limited period of time, might occupy too many pages should I follow what has been a frequent custom in this Society, and an obligatory duty in one at least of our American sister societies; namely, to make the annual address a *resumé* of the important engineering news of the world for the preceding year. Tempting to an essayist, and interesting in itself, as this field may be, I venture to leave it to be harvested by our members with their individual sickles—or should I say self-binding automatic reapers?—that I may touch upon a theme which, though not less important, is less often brought to your attention. This I feel the more willing to do, from the fact that I have more than once been requested to make public certain views which I have at various times strongly expressed in private conversation.

The term *Engineer*, the subject of my title (and of the foregoing brief historical sketch), should, to my mind, include in the person described thereby all the attributes implied in the two nouns which follow. If it always had been thus inclusive, then that higher professional standard would have been attained which our clients in the world outside demand, as well as our own interest and happiness, and this essay would have remained unwritten. Assuming that the great majority of the men who are styled, and who style themselves, "engineers" are really worthy of the title—that in some one or more of the numerous departments of their calling they know how to get the better of Dame Nature, so to speak, by designing and constructing a good road, or bridge, or railway, or canal; by locating and digging a good mine, and knowing what comes out of it; by planning and building a good machine, without the scrap-heap's credit-entry showing more avoirdupois than the bill of lading;

by taming the lightning as a gentle beast of burden in its modern harness of copper and silk and iron; by creating beautiful buildings not of the order of the ephemera; the following questions arise:

Does this aggregate mass of engineers attain to as high a professional standard as, all things considered, our modern civilization would lead us to expect? Does this body of men, who, without question, are of vastly more importance to the world than those of any other one profession or trade, stand as high in the estimation of their fellow-men as their important position would seem to demand? Is their craft (one which, if properly practised, requires as much learning as do the crafts of law or medicine) thoroughly recognized as one of the learned professions?

To these queries we engineers cannot, unqualifiedly, give an affirmative reply. In the first place, we do not in all cases make high enough and absolute enough our standard of qualifications for admission to our ranks. In the second place, we do not have our forces systematically organized into a mighty army, with unbroken front, which would compel, to a proper degree, the admiration and respect of the non-engineering world for the profession as a whole. We have, on the contrary, been fighting too much upon the guerilla principle, and have too often shown the world brilliant dashes by individuals, unsupported by the great body of their fellow-fighters. In a retrospective glance through history, we may perhaps trace some of the causes which have prevented engineering from being definitely organized as a learned profession in early times. Among these causes was possibly the fact that the men on whom alone, among the intellectual classes, devolved the bulk of the hard work of the world's advancing civilization, the engineers and architects, were too busy even to cultivate each other's acquaintance, to say nothing of that of the lazy kings and knights and priests, who were the leaders of influential society. In times of war these kings and knights—lazy no longer—called upon their men of practical science to build their roads and forts and towers, their ballistas and their catapults. Thus arose the military engineer. He was far too important a man to have his eyes put out, or to be walled-up alive in one of his own buttresses, after the completion of his first valuable piece of work, as had been the pleasant experience of some of his civil brethren. He grew to rank with other high officials of the army which he helped to keep in existence, and organized his work after the methods of his fellow-soldiers. Hence the systematic education and training, the *esprit du corps* and high professional status, of the body of men who should in some respects be the model for the now larger body of their lineal descendants, the civil engineers.

I here use the term " civil engineer " in its general, rather than its restricted, sense ; and it seems to me that we Americans should follow the practice of our European brethren in giving to the word " civil " its proper and original meaning—simply " non-military." It is bad enough, in these days of friendship and earnest coöperation between our government engineers in both Army and Navy, with the much larger body of those in civil life, to have the general distinction between military and civil. The classification is not a logical or scientific one, as much of the work in these different branches is identical. A military engineer, in these times of wonderfully rapid mechanical evolution, must be a good deal of a machinist and electrician and aeronaut, as well as a digger of ditches and builder of forts. Even in the last mentioned work (the one thing which distinguishes him from his civil brother, and which the latter is supposed to know nothing about), his knowledge is becoming very uncertain ; for the conventional science of fortification, with its visage grim of brick and granite, seems to be crumbling into *débris* and ashes, from which the young phœnix of mechanical engineering shall spring with a shining countenance, bearing in its lineaments the similitude of nickel-steel.

From this general view of the case, it would seem that the profession as a whole should designate its members by the simple word " engineer." The public would in time follow this example, but, meanwhile, persistent and organized effort should be made to discourage the " Americanism " of using the word to describe the driver of a locomotive or the engineman of a factory ; nay, even the clodhopper who stuffs straw into the fire-door of an agricultural engine, smears lard upon its feverish journals, and hangs his boots and jacket upon the safety-valve, for a maximum test of the elastic limit of the boiler shell.

Allowing the distinction between military and civil work to be expressed only when necessary, it seems to me that the normal use of adjectives as prefixes to our general name should be simply for classification into specialties of practice, as topographical, mining, metallurgical, mechanical, electrical, hydraulic, railway, bridging, architectural, sanitary, etc. These terms are written in a somewhat natural order of progression from nature to art; but are not, of course, in strictly logical form and sequence. This and the preceding paragraphs may be somewhat in the way of a digression from my subject, but are in sympathy with the general idea intended to be expressed of fixing a definite status for the man (or woman, if she be so minded) who shall be called an engineer. Having briefly and sketchily traced his past history and present position, let us, following the *motif* of my title, see whether the term "engineer" includes those of "scholar" and "gentleman." If such is not wholly the case, how far should it do so, and how may it be made to?

As a matter of fact, the modern engineer, if he be worthy of the name, must be a scholar as regards many important branches of knowledge. To have become this, he must possess a trained intellect and must have been through a course, whether in college, or office, or shop, in which he has fulfilled the most important condition of all scholarship, by learning how to learn. The particular branch of learning into which his natural talent, his inclination, and the necessities of his chosen profession have led him, will have required as much study as if he had, for a specialty, chosen Sanscrit, or archæology, or astronomy, or Spanish literature.

In stating the case thus, I do not wish to disparage classical learning. If a young man about preparing for any branch of engineering has time and money enough to take a classical course in addition to the scientific course, which is absolutely essential (if not at college, then at home, or somewhere else), so much the better. The delay of a year or two in starting upon his practical life-work will be well paid for by the increment of mental culture obtained, and by the additional opportunity for class friendships, in after life, with scholarly men, who are not running exactly in his own grooves of thought. If, however, he can by no possibility give to the schools all the time necessary for both courses, let him go through the scientific course thoroughly, remembering the masterly epigram once uttered upon an occasion like this, by my talented predecessor in this chair, Prof. Sweet, which, as I remember, ran thus: "'Tis better to know *what* wants to be done, and *how* to do it, than to know what *has been* done, and *who* did it." Surely, no comparison between the study of practical science and of history in literature could be more crisply, yet more powerfully, formulated.

Our embryo engineer should not, however, take sides so strongly in favor of pure science as to ignore entirely the claims of polite literature. Too many of our young men who are faithful students and earnest workers, but who are too poor to take a full college course, or who, yet more unfortunately, can take no course at all, beyond the common school or academy, are apt to imbibe a contempt for *belles-lettres*, and even, in some cases, for the shade of Lindley Murray himself. They wish to be intensely engineers, and are willing to be nothing more, ignoring social life and other pleasures in their zeal for their chosen work.

To such young men the advice cannot be too strongly given: Do not limit your future happiness and that of your friends and associates by becoming mental hermits—one-sided, unsymmetrical characters, with ideas running in a single groove. Not only for your social happiness, but for your professional advancement, for your worldly prospects in wealth and reputation, make yourselves fit to appear as educated men of the world, not in the bad sense of being familiar with its vices and ready to sneer at its homely virtues, but in the larger sense of being ready to meet men anywhere, of any degree, upon their own ground, familiar with their methods, and acquainted with their ways. For all this you need not be able to instruct a learned Rabbi in deciphering Hebrew inscriptions; nor a Harvard professor in extracting Greek roots; nor even an astronomer royal, regarding the width and straightness of the bands on Mars; they could not tell you the area of an anchorage-plate in your suspension bridge, or the best diameters for the piston-rods in your latest triple-compound engine. You should all four, however, be ready to meet on common ground at your club, or in each other's drawing-rooms, and be not wholly at sea should discussion arise about Shakespeare's iconoclasts; or the most-talked-of article in the last *Nineteenth Century* or *Forum;* or as to *how* American was last week's *American Order of Architecture*, as exemplified in some fearfully and wonderfully made new public building. For all this, it is not absolutely necessary that you should be able to translate even a page of Homer into flowing English rhymes.

Continuing the imperative mood, the advice to our hypothetical young man would be more definitely formulated: Before you begin engineering, ground yourself with a thorough English (if England or America be the home of your birth or adoption) common-school or academical education, learning as much of the classics and of modern languages as time and circumstances may permit. For your professional training, enter the best technical school, college, or university available, the larger and more fully equipped it is, the better. Adapt your personal course of study especially to the branch of engineering which you intend to follow, bringing in as much of the classics as may come, without hampering your science. If your time for languages, dead or alive, is limited, choose German, French, Latin, Greek, in the order named, unless you need Spanish, Italian, etc., for local reasons. If the curriculum of your school does not include ample practical work in field, shop, mine, or laboratory (according as your future work may lie), take care to have had enough such practice, either before, during, or after your school course, as to amount to three or four solid years of such work, before you announce yourself as a practising engineer. In addition to this, do not neglect your commercial education, if possible spending such time in store, bank, or counting-house as will give you a general idea of accounts and commercial law. In these days an engineer must be a man of affairs, and must be something of a merchant and lawyer, as well as a scientist. With all this, keep in mind, as before indicated, the great importance of writing and speaking your own language correctly, and of attaining at least a cursory knowledge of general literature, and something of the shibboleths of cultivated society.

So much for the engineer as a scholar. Answering a question propounded earlier in this essay, I will express the opinion that the best modern engineering courses of study do include such scholarship; and that the great majority of studious men who have been fortunate enough to graduate therein, with their culture supplemented by a few years of active professional life, whether as manufacturers, consulting engineers, or what not, may lay claim to the true scholarship, which consists of a general knowledge of the world at large, combined with such particular knowledge of, and experience in, their chosen vocation, as to make it a success. That minority who have not reached the ideal just pictured, though perhaps more or less eminent technically, are either men who have fought their way up through early disadvantages, or men who, having had the advantages, have a natural distaste for the refinements and amenities of intellectual and social life. If eminent, they are so in spite of this neglect, not because of it. Happily, our modern technical schools, a course in which is now considered almost essential as a preparation for any branch of engineering, are creating a new and higher standard than did or could exist even a score or two of years ago, especially in the case of mechanical engineering. Many of us now only in middle life can remember the times when this branch of the profession had hardly begun to be a science, and but very crudely was an art. Not so very long since our machinery was designed, empirically, by the machinists (or white-smiths, as the English termed them), who were to build and operate it. A mere shop experience was considered sufficient education for these designers, and the resulting machines showed their fatherhood. Their chief pride was to be "practical," and they vied with each other in scoffing at theory and at science. Unfortunately, some of these men are alive yet, and are still at work. Mechanical engineering as a science has, however, sprung up like a young giant ready for the fray. Its influence has pervaded all other divisions of the profession, until the "civil," the mining, and the electrical engineer are largely dependent upon its methods and its men. The scholarship which, but comparatively a few years ago, was monopolized by the older branches of the engineering family, has become necessary to all; and the time will doubtless soon come when nobody will attempt to practise without having enjoyed the rigid training and culture of the schools, or, at any rate, such private training as will pass their examinations.

Having seen that an engineer who is worthy of the name generally is, and always should be, to a greater or less degree, a scholar, we come to the more delicate and difficult question suggested by the title of this address: Is he, or should he be, a gentleman, in the highest and most perfect sense of the word? So embarrassing is this matter that I feel loath to speak of it, and do so only with the hope of adding my mite of influence in favor of the good work, which is already in progress, of elevating the tone and standard of the *personnel* of our noble profession to the highest possible degree, intellectually, morally, physically, and socially.

To the second portion of the query propounded in the last paragraph, we shall all without doubt unhesitatingly say yes! No standard of gentility, no patent of nobility, can be too high for a profession which leads the civilization of the world, and which is probably destined, in the not too far off future, to mark out pathways of material and moral advancement in the life of the race, which are beyond all our present conceptions. Taking a lower and merely selfish view of the case, there can be no question about the great advantage which will accrue to the engineer, in common with the architect, the lawyer, or the physician, from the ability to meet his wealthy and cultivated clients upon their own level; to excel them, if anything, in the intelligent appreciation of their mutual affairs, and in the amenities of social intercourse. This social aspect of professional work, and its great importance in furthering his success in life, not only in the way of pleasure, but of profit and reputation, is too often ignored by big-brained young men who are full of scientific zeal, but who have not learned sufficient practical respect for the ways of the world.

Answering last the first portion of our query, we may all congratulate ourselves that engineers as a body rank high as gentlemen the world over. There are, of course, exceptions, as there are in all vocations, whether with priest or doctor, counsellor or gowned judge. Happily, the tendency of an engineering education is to refine, rather than coarsen ; and this is, I think, the case with the study of science in any of its forms. As we have seen that the very name of engineer is apt to include and carry with it a fair degree of scholarship, so this scholarship carries with it and includes a certain inherent gentility, just as does scholarship in other and quite different branches of learning. Engineering has, however, the advantage enjoyed by other purely scientific studies; it commands an absorbing interest in its devotees which entices away from frivolity and dissipation, and the dealing with pure truth and with the resistless logic of nature leads to veracity and accuracy of character and speech. Who, that has mingled freely in engineering circles, has not noticed the comparatively high moral tone and solidity of character prevalent in their *personnel?* Temperance in all things is in the very nature of the engineer's daily line of thought. He is too much accustomed to trace carefully every effect to its cause, and to modify his causes that his desired ends may be accomplished, to fall an easy prey to dissipation of any sort.

So firmly do I believe in the elevating tendencies of our profession that I might well have added to my title the words " and a moralist." Not only do the studies of the votaries of science tend toward scholarship and gentility, but toward morality; and these attributes act and re-act upon each other. Science is truly the handmaid of religion; "her ways are ways of pleasantness;" she seeketh but the things that are true, "that are lovely, and of good report."

Assuming and believing that the status of engineers as a body is good, and that on the whole we rank before the world with other learned professions, the question arises just as it does with one of our "perfected" machines): "How can we make it better?" How can we outrank all the others, and make ourselves fit to stand before the world so that it will acknowledge us as the leaders of its advancing civilization and large factors in it, as openly doing what we really have been doing *sub rosa* through all the passing centuries? How shall we make it consider us, when in properly organized form, a body so wise and powerful as to be fit advisers not only in matters of applied physics, but in social science, in commercial economics, and perhaps even in politics itself (if that subtle profession is not killed out by that time)?

And why not such added functions? They fall within the line of thought which I have projected—a line reaching from before the dawn of history to a bright era in the future, when the affairs of men shall be run on the engineer's time-table; that is, by the rules of common sense. This, being interpreted, doth mean simply that we shall try to win nature's gifts by using nature's laws, not by controverting them.

Answering the questions above propounded, as to how we can in all ways elevate our professional standard, I will suggest that the best way is to have a standard, and then to elevate it. To do this we must have such general organization as will give concerted action, and then command general respect. Furthermore, this standard should have some sort of a legal status, at any rate as regards its minimum limitations. Just as firmly as I believe that no doctor, nor architect, nor lawyer, nor chemist, should be allowed to invite fees from the public, without first passing an examination and receiving a degree from some reputable school, which, in its turn, should be subject to examination and regulation by the general government as much as should a national bank, so do I verily believe that the practising engineer should be authorized to be such by some higher and more uniformly acting power than his own choice. Not only do we want such regulation for the protection of the public, but more particularly for our own protection, that we may not suffer from the charlatans and quacks who infest our ranks, with symbols ending with an E appended to their names. We may not yet be ready for any such system of unification, but it seems to me that we should *build* toward something of the kind in the future, as the surest way of fixing our status in the eyes of the world as belonging to one of the definite and recognized learned professions. Should such a system once be established, there would be no more chance of a reversion to the present systemless style than there is of our American national banks changing back to the old-fashioned "wild-cat" banks of forty years ago.

In the meantime, whilst we as a body are educating ourselves up to, and preparing for, some scheme looking toward more unity, power, and usefulness, how shall the average tone of the profession be improved by the cultivation of the individual? Here the advice regarding the Scotchman in the story may be followed: "Catch him young." The rising generation of engineers must be trained to higher ideals than has been any preceding one, and this is largely the work of the schools.

We have, in this country, many excellent technical colleges. Generally speaking, their standard of scholarship is high, their methods thorough. Their chief fault is, perhaps, lack of uniformity with each other, in certain things where standard methods of study and experiment would be desirable; and, more important yet by far, some standard regarding the conferring of degrees. The evils due to these sins of omission may perhaps be remedied by some system of official conference between their respective faculties, some intellectual "pooling of their issues," so to speak. Not only would this be productive of uniformity where uniformity might be an advantage, but it would improve the average character of the schools themselves, and tone up the weaker ones by exciting a spirit of more active emulation.

We have assumed that our coming engineers, many of whom are now in our colleges and "institutes" training for the development of brain and eye and hand, will there find all needful helps to the scholarship which is to prepare them for their calling. They will there be trained, also, to some extent, in manners and personal habits, both by precept and example, especially those of them who, by reason of poverty or geographical isolation, have not had the early advantages enjoyed by some of their more

Chesterfieldian fellow-students. Recurring once more to the second natural attribute of an engineer mentioned in the title, we may ask: Do these schools establish and maintain the high standard which they should do in regard to good breeding, elegant manners, tasteful costume, and those graces and accomplishments which, as before intimated, will so much aid any professional man in gaining the desirable prizes of life? The answer must be, that our civil schools do not, to the extent they might, provide this culture as yet.

In this respect they should follow some of the methods of governmental military and naval schools, whose curriculum includes not only a rigid technical and professional course of study, mingled with enough of classical and general literature to cultivate the mental graces, and a superabundance of military drill and other *systematic* physical culture for strength and grace of body, but also instruction and practise in dancing, in dress, and in the nameless other social arts which make for politeness and refinement. Nor do we see that attention paid to these matters in any way cuts short the more severe culture which is the chief aim of education. To disprove this we need but think of the long list of names, which it would be invidious to specify, among the regularly educated army and navy men of this country and Europe who have become distinguished in science and in general engineering.

We cannot take all our young engineers from cultivated homes and teach them only technics. Many are blessed with such homes, but many others must come from orphan asylum, from factory, from distant farm. We are all, in a certain sense, children of the people. Class distinctions are gradually disappearing not only in America but in Europe and the Islands of the Sea. Notably is this the case in what we may almost term our sister republic of England. The ultimate social distinction will simply be between gentlemen and non-gentlemen. Only the men with sufficient "voltage" of brain-power will survive in the struggle for a properly standardized engineering education. If they happen to need it, why should they not incidentally receive, as they pass through the mill, the polish of gentility?

It is an observed fact, that many a mere "cub," drawn from his lair in backwoods or mine by some benevolent congressman, and entered at West Point or Annapolis, has, if showing brains enough to go through at all, been turned out a scholar and a gentleman.

He, throughout his after life, mingles with ladies and gentlemen, is one with his fellow-officers, for nowhere is *esprit du corps* stronger than in army and navy circles. Let us do likewise with any engineering cubs that we may catch. Let our schools imitate successful methods wherever found. Let them throw such a mantle of refinement over each of these young men as not only to conceal, but to absorb his cubbishness. Make him not alone truthful, temperate, scientific, skilful, business-like, scholarly; but healthy, erect, graceful, socially accomplished, possessing "manner," as well as manners. Give him a love for moral and physical cleanness—morally a cleanness shining out from soul and mind, and physically one which shall be manifested not only upon his face and upon his shoes, but shall reach even to the tips of his fingers. Give him a horror of careless pronunciation and orthography, but such a practical respect for etiquette and fashion as will cause him to use either at will, as an expedient tool, while, perhaps, contemning it theoretically. Teach him that there is raiment fit for the morning, and raiment meet for the evening; or, at any rate, that he may find chances to make some pleasant hostess happier by assuming such a proposition to be true, whether it be strictly logical or not. Search the effects of a youth thus trained for personal and well-used implements of his calling, and expect to find not only tape-line and calipers, but a thumbed-up volume of Shakespeare or Emerson; not only a two-foot rule and an "aneroid," but a card-case and the accessories of a dainty toilet.

The general and social culture which our schools should provide for these young men cannot be a matter of precept only, but must be one of example also. There must be a high standard among professors and instructors as well as among students —not only in scholarship, but in manners and morals. While speaking again of morals, as among the attributes of a gentleman, I fain would utter an earnest plea for the nurture and cultivation of that knightly sense of personal honor which is regarded all too lightly in the money-getting strife and turmoil of our modern commercial life, but which possibly we may find more prevalent in military than in civil society. Such honor is above all bribe-taking, and should be sacredly held as a part of the capital of every engineer. It should keep him (if for his reputation's sake only) infinitely above the consideration of such loot, for example, as commission upon material, the sale of which might be affected by his professional opinion thereupon. It should lead engineering influence to reform our wretched system of expert testimony, wherein two reputable scientific men are paid by interested clients to draw two opposite sets of conclusions from the same premises, instead of being paid by the courts to tell the truth, as should be the case, and will be—when we engineers make it so. This high honor, and the gentle courtesy by which it should be manifested, are but the "applied science" of that *Golden Rule* which was preached by the Greatest of Gentlemen, in Galilee, eighteen centuries ago. It merely causes us to treat all men and all women as we ourselves would be served by them. Shall not our schools include it, more definitely than now, among the things which it is good to learn?

Catch we thus young the rising generation, thus lead them and guide them toward the highest possible ideals of morality, science, work, scholarship, and gentlemanhood, meanwhile organizing our forces by standardizing our methods, and by federating our societies, great and small, for unity of action in matters where unity may seem desirable (but retaining their autonomy for local and departmental objects), and the early half of the twentieth century will see and recognize a noble guild, girdling the earth both intellectually and materially, whose power and influence will lead mankind forward, yet more than in the long past, toward all that makes for prosperity, for purity, for pleasure, and for peace.

CCCCXVI.*

LIGHT CABLE ROAD CONSTRUCTION.

BY FRANK VAN VLECK, LOS ANGELES, CAL.

(Member of the Society.)

THE main objection urged against cable roads for small cities has been the almost prohibitive cost of construction. Advocates of electric traction have even asserted that cable roads could not be built as a single-track system, thus making the cable road a luxury only for the metropolis. In the far West there are a few good examples of single-track cable road work, and the object of this paper is to present another case where a single-track cable road has been put under construction—a road in which the main aim has been to combine the elements of a thoroughly good plant with such economy in cost as would warrant the erection of such a system in a town of between thirty and forty thousand inhabitants.

In these days, when it is almost universally recognized that for the purposes of intra-urban traffic the emancipation of the street-car horse is at hand, it is asked and demanded, which, then, shall be our steed for our new rapid transit, electricity or the cable? Electric railways, under the enterprising energy of special corporations formed to develop them, have made enormous strides, and their success has been well merited; for, in truth, it must be said their attainments, both mechanical and electrical, have been almost phenomenal. Yet there are some things the electric railroads cannot do, or cannot do, as at present constructed; notwithstanding some amusing claims of the electricians that certain grades can be ascended which the steam-railroad men know to be impossible of ascent under all conditions of track and weather. The writer, although connected with the development of the cable system, cannot but conclude that the day of usefulness for the cable on the level is forever gone, and that the electric road stands the champion of the field. When the ascent of steep hills is to be undertaken by

---

* Presented at the Richmond Meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

any street road, then the advantages of the cable over any other means of locomotion are apparent. Representatives of the electrical companies have claimed, and even attempted, grades ranging up as high as 16 per cent. for their motors, and have repeatedly asserted that, when using the return current by the rail, the tractive effort is increased by the electric adhesion between rail and wheel. The amount and character of this adhesion has not as yet been as fully explored as the demands of tramway engineers would wish, and with the electrical engineers themselves this feature has not been made as much use of as the merits of the discovery would seem to warrant. Therefore, in the present condition of the system, it would appear unwise to assume that any electric-motor car could have any greater capabilities of ascending hills, or for traction, than steam-motors with same conditions of weight, wheel base, and track.

Most of the cities of our country are located on the level, and as such are eminently adapted for electric railway work, while often other cities have the misfortune to be built on hills. He would indeed be a bold projector who would claim the possibility of ascending some of the San Francisco hills with any form of traction locomotive, some hill streets in that city being entirely given over to the cable roads, as no vehicle can with safety ascend or descend. The inclines of either the Hoboken Elevated or the Brooklyn Bridge would also be a difficult ascent under all loads and wet tracks.

Space has been taken for allusion to the relative merits of these two systems, as the consideration of them has been an important feature in the San Diego Cable Railroad, about to be described.

This San Diego franchise was first operated as an electric road, and as such continued in operation for about a year. Deficiencies of construction, improper design, inability to ascend grades with any reasonable speed, or at all, and large leakages of electricity and consequent loss of power, compelled the company to cease operation at an early date.

The electric road's franchise and equipment were then purchased by a company purposing the adoption of the cable system. This new cable road has just been put into operation, and its description constitutes the subject-matter of this sketch.

Before breaking ground the officers of the road had already determined for the engineer that the road was to be a single-track system, and was to be constructed with thoroughness, yet at a cost which was to be materially less than that considered moderate for other roads. For it must be understood that the city of San Diego is of such size that neither heavy traffic nor receipts could be expected for some time to come. It therefore may be observed in the following descriptions that, while many of the best elements in cable engineering are employed, the designs also show that the items of cost have in each case been pared down to lowest figures. In many cases new departures from usual practice were adopted for economy, yet, when such new designs have been completed and installed, they have been found to be more satisfactory than the older forms.

It will be seen that the construction of a single-track system produces a complexity of design which is practically a simple matter to the double-track system—a case where the apparently simple is yet the most complicated.

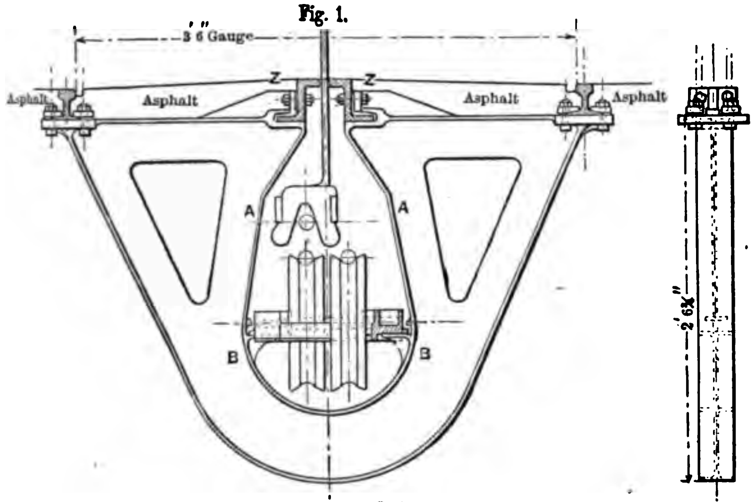
The double-track roads have but one cable moving in their conduits, and moving invariably in the same direction. The underground passage of grips and the movements of surface cars are therefore positive and in the same direction. But in the single-track system, although it may be the essence of simplicity in the streets, yet underground its ways are devious. Two cables demand attention, each hurrying in opposite directions, yet side by side. The grip likewise must be capable of holding to its proper advancing cable and not to the other. Curves present a maze of difficulties. For the uninitiated can see that two cables, each running in opposite directions, cannot pass around the same curve-pulley at the same time.

PERMANENT WAY.

The backbone and foundation of a cable system is the road substructure, the form of the iron yoke binding all the rails and shaping the conduit, and the concrete foundation. The influence of a fine California climate had not a small share in the determination of the iron which forms the yoke, for in this part of California frost in the ground is a contingency which engineers never consider.

This yoke is shown in Fig. 1. The positions of track-rails and slot-rails are as shown. The form of the central conduit was determined by these considerations: Two cables must traverse the entire length of the conduit. They must not be so close together as to chafe each other in oscillating sideways, nor so close as to touch the rims of the sheaves carrying the opposite cable; nor, again, must they be so far apart as to swing against the concrete sides of the conduit, or so close to those sides as to render difficult the passage of the grip, which must embrace the cable from the sides. The sizes and position of the sheaves carrying the cable also had their claims. Drainage had to be provided below all, in order to allow free passage of water and dirt below both cables and sheaves—for it must be said of Southern California, with somewhat of the reputation for being

Fig. 1.



STANDARD YOKE.
Scale, ¼"—1 foot.

during most months in the year a semi-arid region, yet it does have the accompaniment of rain sometimes during the winter months, and when it does so rain vast volumes of water fall in short spaces of time. In such cases cable roads may run like small sewers, and not to make provision for water would be ruinous to the cable; for the grit thus carried about and into the strands of the cable doubtless is a prominent factor in decreasing its life. Finally, the cross-section of the grip must determine the main form of the conduit, for this grip must pass every point of the road and must be capable of taking the rope only at the proper level, and taking the correct moving cable according to the position in which the car is headed.

Therefore, in a road of this kind it was necessary that, even before any other mechanical details were determined upon, the grip itself should be designed, or, at least, the form of its cross-section unalterably fixed.

The form of conduit thus derived will be observed to depart from the usual practice of double-track roads, in that the grip necessitates an angle in the side of the conduit where ordinarily is but a straight side, as seen at *A*, Fig. 1. In the matter of the gauge between rails the directors of the road simply consulted expediency, nor did they hunt up any absurd regulations, if such existed, requiring street lines to conform to some old rustic's idea that street tramways should always be made to conform to the supposed gauge of all the local wagons and carriages. Many cities have enacted such ordinances, with the thought that it was the right and privilege of all vehicles to have the pleasure of riding in the "flats" of the street rails. Thus in various cities of the country do we see gauges running from six feet to three, and even less. It may be that the "battle of the gauges" is yet unfought for tramway work. It almost appears that the standard 4′ 8½″ has not a proper *raison d'être* for this class of construction, which is scarcely comparable with steam railway work, and rarely exchanges traffic with such roads. Therefore this San Diego road, following the example of numerous other cable systems, promptly decided for the 3′ 6″ gauge, and would even have decided for six inches less and reaped further benefit of still diminished cost of construction, had they thought that public sentiment would look with favor upon such a narrow gauge in a city already compassed by the wider standard gauge. The outside width of the yoke being thus determined by the gauge, the yoke as completed is somewhat of a V form with the bottom rounded. The conduit or tunnel extends to within six inches of the bottom. From the figure it will be seen that the tendency of the yoke to close is resisted mainly by the metal in the cross-section at the bottom of the conduit; this, therefore, was given as much iron as could be conveniently done in a yoke of so light a weight—150 pounds.

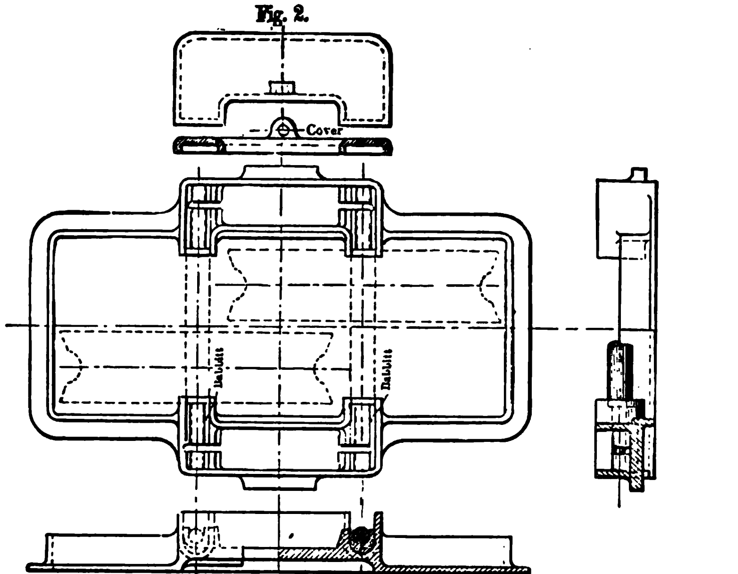
The rail used is of twenty-five pounds section, and although light for cable road service, yet, as large quantities were left as an heirloom by the former electric road, it was decided to use it for whatever might be its term of life, and then, if the traffic so demanded, lay a heavier rail. T rails can never be regarded as the best tramway section to lay in a city street, yet where the paving outside is close to the head of the rail, and the paving inside is at the height of the top of the rail, with but a flange-groove between rail and paving, the objections to the T section almost disappear. This was easily arranged, as the paving between rails on the entire road is of asphalt, or what is more specifically termed, at the Pacific, bituminous lime rock.

The slot-rail (*Z*, Fig. 1) is a special section rolled for this road, the desire here being to secure one of the lightest possible forms, one which would be inexpensive to adjust, and one which should avoid what with many other roads has been inevitable—the use of paving-plates. What are known in cable construction as paving-plates are strips of cast or wrought iron extending from yoke to yoke, near the slot-rail, to prevent the concrete from being pushed from its sharp angle into the conduit or tunnel.

This matter of paving-plates alone is one which may affect the cost of a mile a thousand dollars or more.

A few cable roads, on the ground of pinching economy, have attempted to save a considerable amount by the abandonment of concrete as a binding medium to hold yokes and roadway together. Although this saving may be material in first cost, yet the expenses of general repairs and the hosts of nuisances of bad alignment and closing slots render it unsuited to what aspires to be a satisfactory system. Therefore it was early decided to use concrete throughout this new system, but not in such large cross-section as would require a large outlay. The external bounding of the concrete cross-section follows the outline of the yoke, except that at the bottom the concrete is run three inches below the yoke for the purpose of forming a better foundation, and a bond between the contiguous blocks of concrete between yokes. This concrete was formed with the proportions usual to many roads, three parts sand, six of broken stone, and one part of best imported English Portland cement. The mixing was in all cases done on the road, beside the work, and at once thrown into position between the yokes in the trench, and then thoroughly tamped. Yokes were spaced at four feet centre to centre, and at every tenth yoke was placed a "pulley yoke," this being such a one as is shown in Fig. 1. The plain yokes do not have cast on their conduit curve the two brackets shown at *B*, *B*. In other single-track cable systems it has heretofore been the custom to place a pulley or rope sheave on one yoke, and on the next yoke another pulley for carrying the opposite-going cable. This requires two yokes to be prepared for the reception of the pulley-boxes, and then on each yoke two boxes are to be attached, usually by bolting, and brought into proper alignment. To make each of these two pulleys with their four boxes accessible from the street required either two manholes through the paving or one long and expensive manhole spanning from yoke to yoke.

Fig. 2.



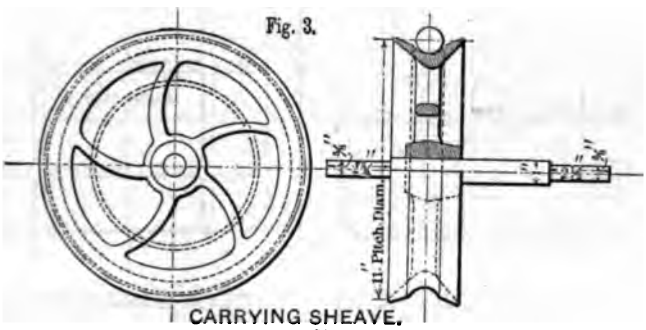
SHEAVE FRAME.
Scale ⅛

By the system here employed for the first, both pulley shafts are brought close together, and can therefore be carried in a single frame (see Fig. 2). This frame passes around and carries both pulleys. Having them both in the same frame, whenever the frame is adjusted to proper position then are all four bearings in correct line. This frame is simply passed in over the brackets attached to the yokes until it comes to its mid-position, when a depression in the frame drops over the bracket; if, then, a thin, wedge-shaped piece is driven in on each side, the frame is rigidly held to place and can neither back out nor move forward.

Another advantage of this frame-carrying device is that the oil-box on one side embraces both journals; thus, one oiling oils the two shaft ends. It also affords a large receptacle for containing the oil or oil-waste, or "dope." The whole thing is also readily accessible from one small manhole in the street.

The small carrying sheaves (Fig. 3) are ten inches in diameter with a three-inch face, and are mounted on cold rolled steel shafts about twelve inches long. The sheaves are of cast-iron throughout. To the uninitiated in cable work it appears strange that cable engineers persist in letting their splendid steel cables wear themselves out by passing along and around wheels whose surface is hard cast-iron, while theory would seem to dictate the invariable practice of using a softer material for this rope to

Fig. 3.

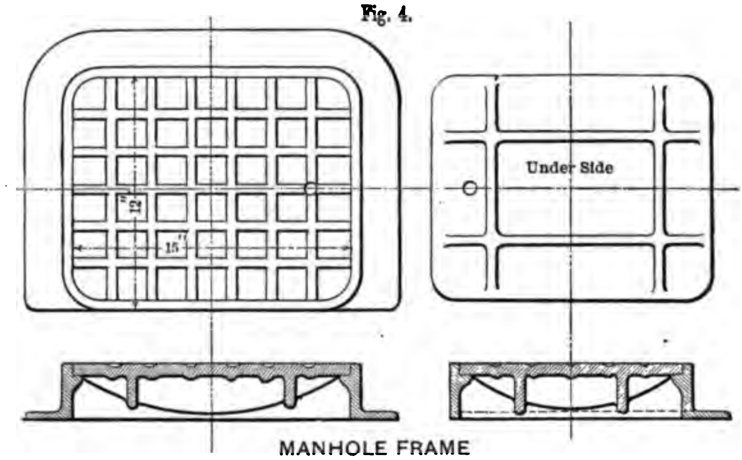


CARRYING SHEAVE.
Scale ½ Size

pass about on. Some roads do, indeed, use sheaves with grooves filled with Babbitt or wood. Whether such use has lengthened the life of the cable appears to be doubtful, while its manifest disadvantages have been often too evident. The great disadvantages are greater wear and consequent frequent renewal of the softer material and the greater difficulty experienced by the soft wheels taking the "lay of the rope." This latter is an effect which all cable men seek to avoid. When a cable is considered in its best condition, its depressions between contiguous strands ire uniformly filled with tar or pitch, so that the appearance of the cable is not unlike a well-tarred gas-pipe. In this condition at resists better the wearing action of sand or grit, which may chance in the conduit, and is far smoother in its action in passing through the jaws of the grip. When a wheel or sheave takes the "lay of the rope" it has formed in its groove the matrix of the cable or the opposite of the space between strands. This matrix gradually increases its resemblance to the space between strands, with the consequence that it pushes all the tar from between the strands, thus setting back the cable to its original naked condition. When such a wheel or sheave is found it is promptly taken from the road and thrown away, or the groove, if the stock in the rim permits, is re-turned.

The manhole through the paving whereby the oiler has the opportunity to perform his duties is perhaps smaller in this road than any other. As this is a detail of constant repetition

Fig. 4.

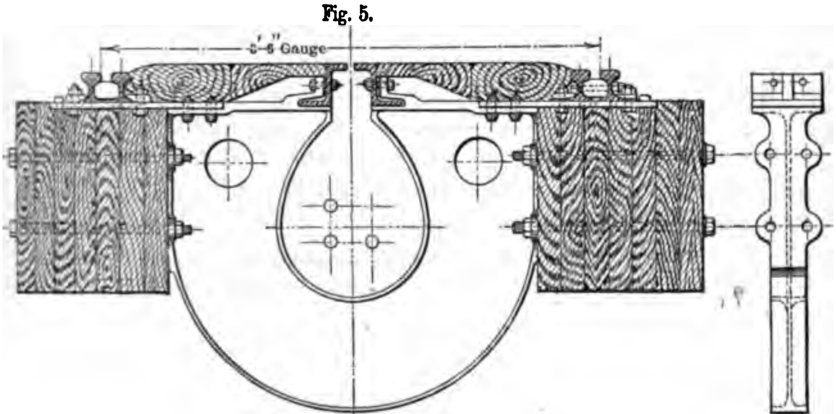


MANHOLE FRAME
FOR PERMANENT WAY.
Scale, 1/6 Size.

on the road, it is evident that five or ten dollars saved in its construction means a comfortable saving at the end of the road. The size is only 12" × 14," yet a small man or youth can pass through it and along the conduit of the road. During the progress of construction on many roads it is desirable or necessary that a man should pass into the conduit for the purpose of bolting up parts or attaching sheave supports, but on this road all can be done from outside, even to the attaching of the pulley-frames, for neither do these nor other internal devices require any work of bolting. Practically, then, the main and only use for these manholes is for oil-holes. The manholes and frames are shown in Fig. 4, and although they are but made of cast iron, yet they are so braced that they give maximum strength. In fact, not one of them has yet been broken by heavily loaded wagons, by fire-engines, or road rollers.

The line of the road in two places crosses deep gullies, in one of which the roadbed is some seventy feet above the bottom of the ravine. In both of these cases high wooden bridges have been built. As it was possible for vehicle traffic to come upon these bridges, it was decided not to leave the space between rails open, and accordingly the problem was presented—to carry the slot-rails in the centre and to carry any load which might come between rails, and to preserve accurately the gauge distances between all rails; and, finally, to have the yoke so formed that it might be afterward filled with concrete, should the ravine be

Fig. 5.



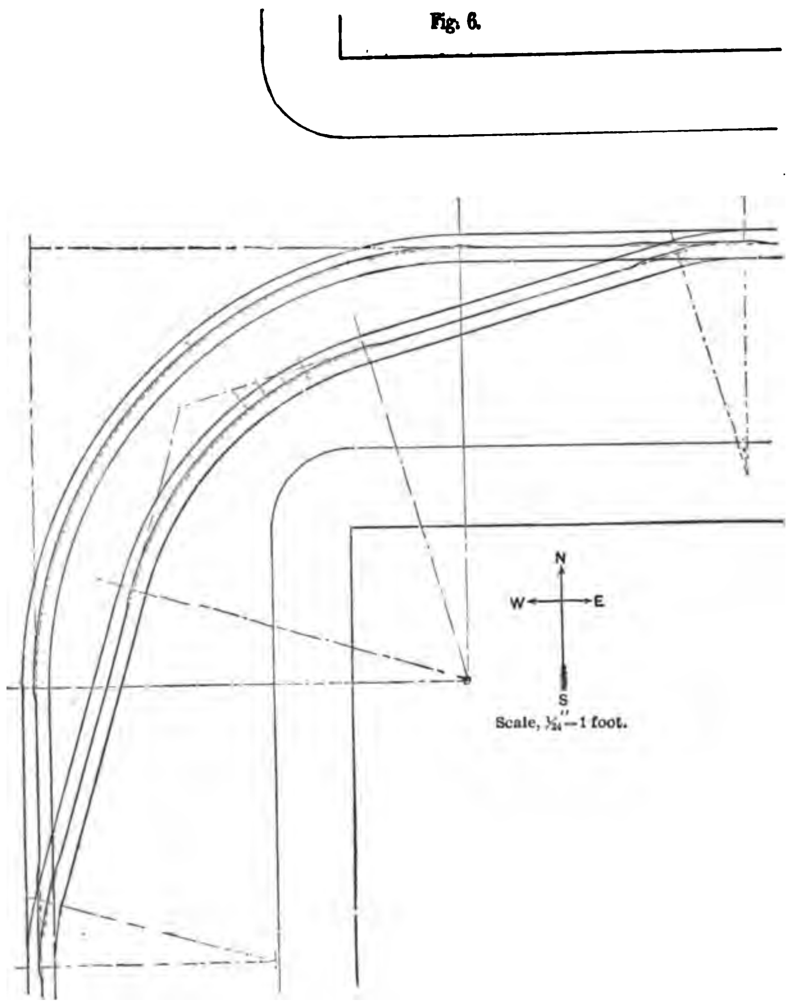
BRIDGE YOKE.
Scale, ¾" = 1 foot.

filled and graded. The yoke devised for this purpose is shown in Fig. 5.

This yoke was given a little more metal through the central section in order to resist closure, should heavy loads come upon the slot-rails, and in this case it was the more desirable as no concrete was here used to bind or to strengthen the yokes. The ravines over which these bridges pass will probably be filled and graded to the proper street-level later, when these yokes can be arranged for the usual track work by attaching a triangular bracket to each side of the yoke and under the rails, and then filling in with concrete in the usual manner.

Scarcely any street railway-system can be built without curves, and these to the cable system are perhaps a necessary yet an unmitigated nuisance. Costly to a degree which make stockholders sigh, they are a source of endless care and anxiety to the management; and not content with that, these crooked tracks still demand further extravagance by eating the life from

Fig. 6.

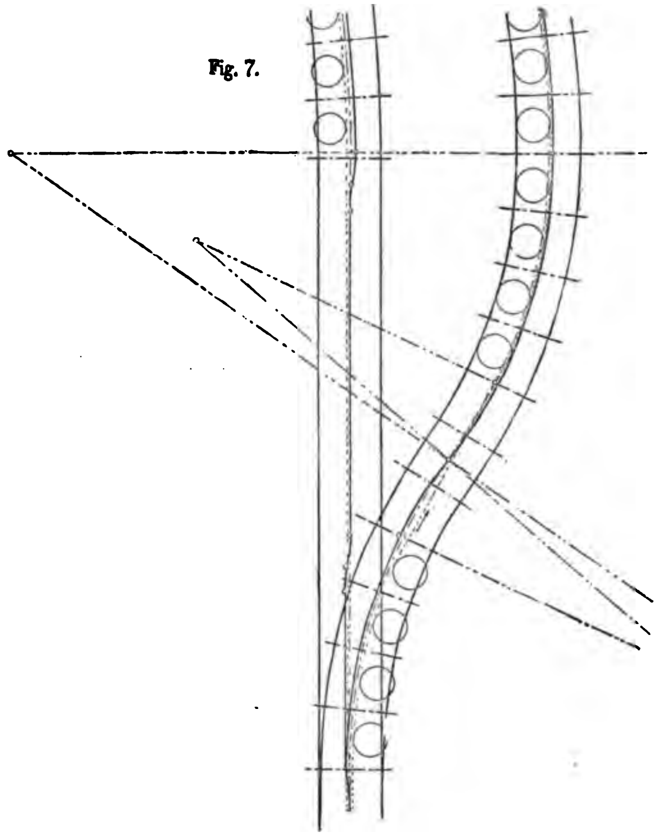


the cable. Curves are bad enough for the double-track systems, but with a single track the difficulties increase. For how shall these two cables, each moving in opposite directions, be taken around the horizontal curve-pulleys? For neither can both ropes be carried on the same sheave—for no wheel revolves in two directions at once—nor can alternate pulleys be used for either rope. Practically, there are but two methods for accomplishing this feat. The method first made use of was to carry one rope at a higher level than the other, and have it pass over curve-pulleys above the pulleys carrying the opposite-moving rope. This required that the gripman, when having the lower rope, must invariably release his rope from the grip, and then let his car "float" over the curve either by momentum or gravity. Usually these curves were located on a grade in order specially to make use of gravity to carry the car around. In these cases, when the gripman did not release and drop his rope, the grip, from its construction, carried the rope into the wrong pulleys, and trouble ensued at once. To add to the difficulties, after passing the curve provision must also be made for picking the rope up, for grips as usually constructed have not the power to go down after the cable. The other solution of the problem is to separate the ropes and to construct the curves as double tracks. This was necessary in the road described, as no curve was on an incline which could be depended upon to float the car over by gravity. As a single-track road requires a certain number of passing places or turnouts, these double-track curves were so arranged on the line as to serve this same purpose of passing places. Fig. 6 shows in diagram the manner in which one of these curve turnouts was arranged.

This is a right-angle curve, where the turnout is located on the inside of the curve. It will be noticed that the inner curve is not included by the same angle as the outer one. This was caused by the fact that this arrangement would give a short piece of straight track between the ends of the inner curve and the switches, thus doing away with a number of curve-pulleys and their corresponding disadvantages. The turnouts, whether arranged as thus indicated for the curves or whether constructed at the side of the straight track, invariably have passing through them the moving cable to insure that the car will always be able to move in its proper direction. The entrance to a turnout on the straight is shown in Fig. 7. This diagram also indicates the manner in which the slot is deflected from the centre line in order to insure that the grip will not strike the curve-pulleys. At no point on the road between termini, except at power station, does the grip have to let go the cable. Thus at all points, whether on curves, turnouts, or switches, the car can always retain its rope. This releases the gripman from a vast amount of responsibility and insures greater safety for the passenger traffic.

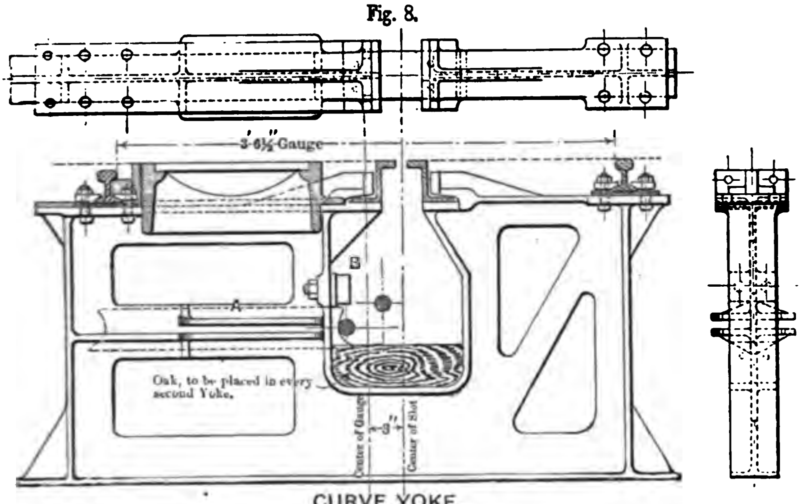
The form of yokes for the curves were designed with many differences from the yoke required for the straight, as the curve demands features of strength which the other need not possess.

Fig. 7.



The form of curve-yoke is shown in Fig. 8. The main outline is made in the form of a square, for the purpose of strength, and for holding a greater body of concrete than such a depth of yoke would if made in the V form. For if concrete in sufficient mass has not been put in a curve, the curve has been known to be pulled either up or out of shape by the heavy tension of the cable. The conduit, it will be noticed, is shallower than the straight yoke; this is due to the fact that each alternate curve-yoke is fitted with a wooden block whose top surface is but a short distance from the cable. As this impedes the flow of water through the conduit, it was decided to form the channel-way for water under the frames for the pulleys, where it might have unobstructed flow, and which space of necessity is left unfilled with concrete, for the action of the curve-pulley. The yokes as thus constructed are strong, and when concreted up form a rigid construction capable of withstanding any of the strains to which curves may be subjected. The frame which carries the pulley is attached between two brackets cast on the yoke at *A*. The grip is prevented from being drawn against

Fig. 8.



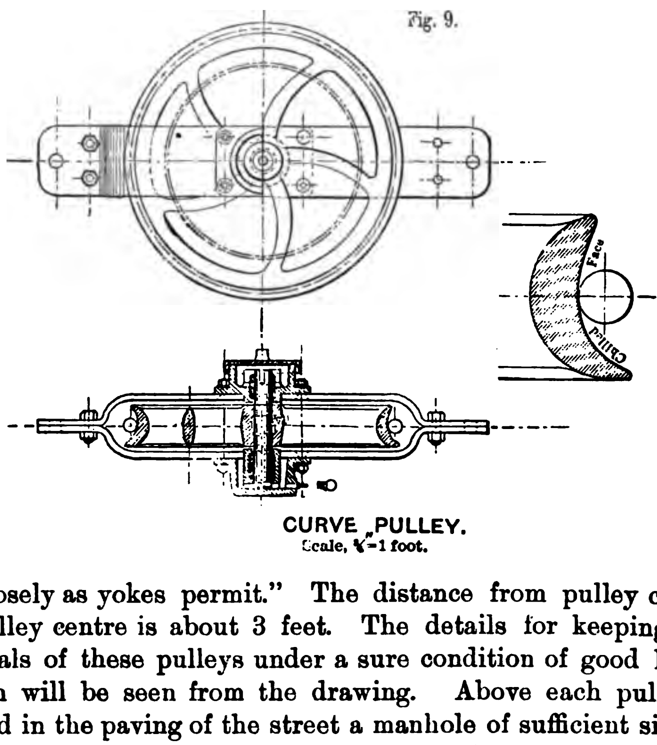
CURVE YOKE.
Scale, ¾" = 1 foot.

the pulleys by the grip-guard at *B*, against which the grip slides in its entire passage about the curve. The grip, previous to its entering the curve, is deflected three inches to the outer side, by a deflection made in the track-slot to that amount. This permits the cable to run directly from the tangent to the first curve-pulley in its own proper line, while the grip by this means is carried to one side enough to clear the pulleys, and the strain imposed upon the cable by this bending is taken by the grip-guard mentioned.

The curve-pulleys are carried between each yoke by the special frame of wrought iron shown in Fig. 9. These pulleys are of cast iron with chilled rims, the width of rim being but

$3\frac{1}{2}$ inches. This narrowness of rim is made possible by the fact that the rope is lifted but 3 inches by the passage of the grip, and, consequently, after the grip has passed the rope falls to its mid-position on the sheave. Even should the rope come higher it could not remain above the rim of the sheave, due to the presence of the grip-guard. As these sheaves are so small in width it is deemed cheaper to renew the whole sheave when worn than to use a larger pulley with a renewable tire. The diameter of the wheel is $22\frac{1}{8}$ inches on the pitch circle, thus striving to attain what the present best cable practice deems most desirable, "as large curve pulleys as possible, and spaced

Fig. 9.



CURVE PULLEY.
Scale, ¼"=1 foot.

as closely as yokes permit." The distance from pulley centre to pulley centre is about 3 feet. The details for keeping the journals of these pulleys under a sure condition of good lubrication will be seen from the drawing. Above each pulley is placed in the paving of the street a manhole of sufficient size to permit of the removal of the pulley and its frame complete.

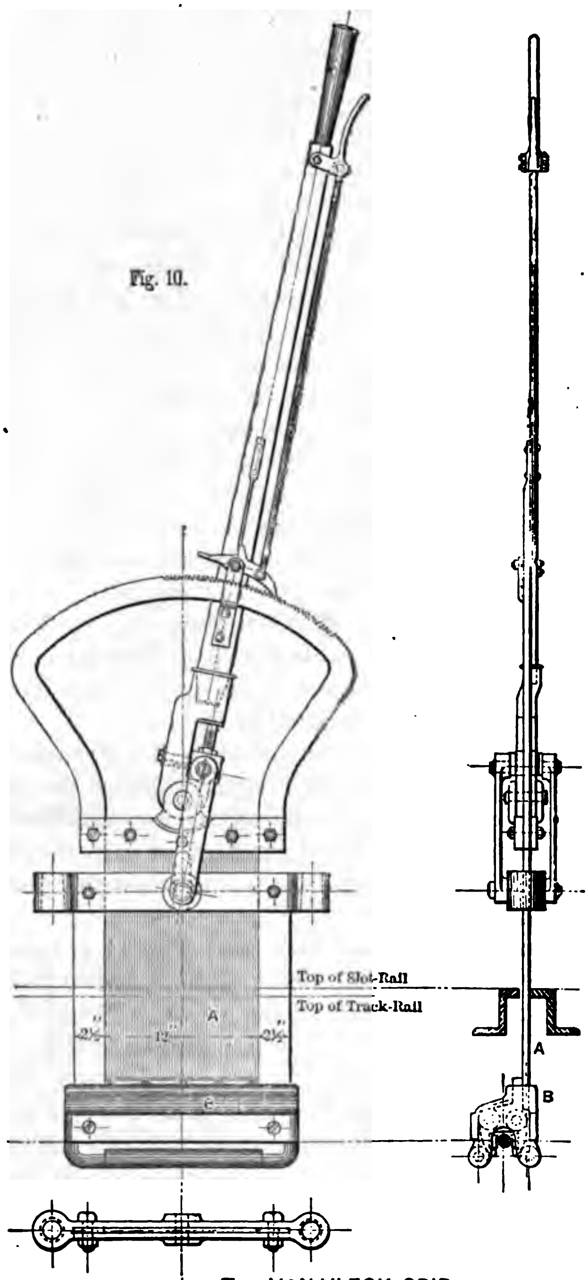
THE GRIP.

Perhaps the one thing about which the whole design of a cable road hinges, from roadway to rolling-stock, from curves to turnouts, is the form of the grip. The grip early decided upon was a modification of the "bottom grip," now in most general use in

San Francisco—a form which has been known as the San Francisco grip; but as so many of the grip constructions have emanated there and have received the same appellation, it is now no longer a designation of any definiteness. Single-track roads have heretofore used the grip known as the "side grip." But for its manifest disadvantages, it would have been adopted in this new work. It does, however, have one feature recommending it to single-track work—the ability to grasp the cable at the side—for, necessarily, where two cables are in the same conduit, each must be at the side of the centre line. The bad features of the grip for this new plant were that it lifts the cable too high from the normal position which it occupies on the carrying sheaves, thus requiring a deeper conduit; and its general unmanageableness when the cable has to be dropped and picked up. For in case the gripman by carelessness neglects to let go the cable at the proper time, the cable must either be torn or the grip demolished, and there is no escape from the alternative. At certain points on the road, as at power stations and turn-tables, the cable must plunge down, and for the grip to "hang on" requires that something must give way. The "bottom grip" offers the two advantages that it lifts the cable but little from its normal position, and that in cases of neglect it can spring open itself before doing great damage either to itself or the cable.

It was found on investigation that these San Francisco bottom grips had never been constructed on the side, for the purpose of taking the cable, as would be necessary in single-track work; accordingly, the writer designed the form of grip shown in Fig. 10, which is possibly the first attempt to construct this grip as a side and bottom grip.

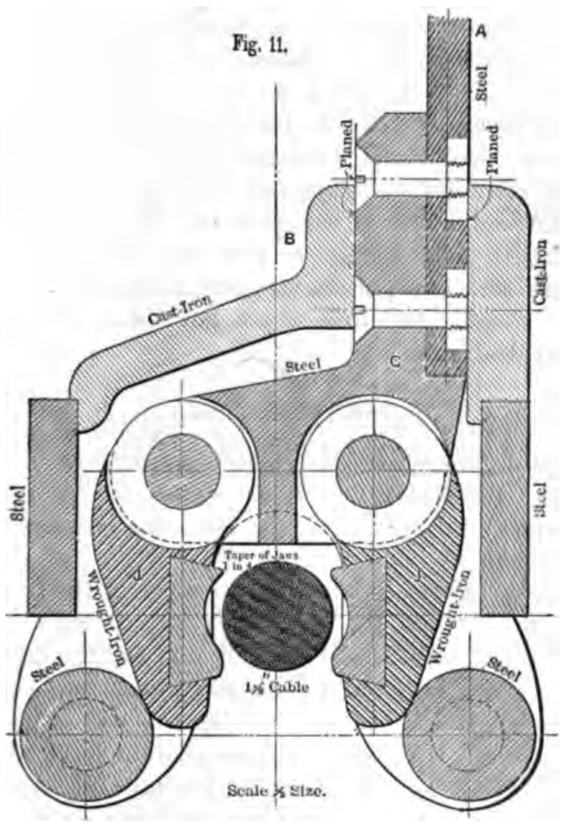
Fig. 10 shows the general view, and Fig. 11 a cross-section through the grip-jaws. The shank plate *A*, below the slot-rail, passes through the upper part of the grip-box *B*, and carries the hinge-piece *C*. This hinge-piece has hinged to it the two jaws *J*, *J*: these jaws constitute the vice for gripping the cable. The special surface which grasps the cable is an inserted piece, usually of cast steel, dovetailed in the inner side of each jaw. The softer metals are often employed for this purpose, but, as the wear is so rapid and the renewals so frequent, the expense counts up to such an item that it is considered as good practice to put in hardened jaws and assume the possibility of a trifling greater wear upon the cable. The pressure upon the jaws caus-



THE VAN VLECK GRIP.
Scale, ¾" = 1 foot.

ing them to tighten upon the cable is produced by pulling upward on the outer side of the jaw a steel roller *E*, carried in the bottom of the grip-box. This grip-box is fastened to the two side shanks of the grip, and when the gripman pulls up his lever to the position for tightening, this movement causes the two side grip shanks to be drawn upward, thus by means of the rollers causing the closure of the jaws upon the cable.

Fig. 11.



This subject of grips is perhaps one of the most important ones in cable road management, and, like many other important details, is treated differently by different companies or engineers. Each road is found to be using its own device. The relative merits and demerits of each have caused considerable discussion, and yet offer a fruitful field for engineers to balance practical considerations against theorized advantages. Thus, for example, may be drawn out the case of the ingenious device of Colonel

Paine, on the Brooklyn Bridge, contrasted with the direct rubbing grips of some other cities.

The rolling-stock for this road presents a few new features, but such perhaps as, in detail, would only be interesting to tramway men.

To those of the East it may be well to mention that the cars are equipped entirely with trucks, as it is found that a much larger and more convenient car can be built than is possible for one which is simply to ride on a comparatively inflexible wheel base of four wheels. As the wheel base of the trucks is only four feet, it is seen that curves of extremely small radii can be turned with ease. The height of car body is no more than is usual with the standard Eastern build, yet the trucks are given a free angle for swinging and the wheels do not necessitate ugly covered holes in the car floor. This is made possible by using smaller wheels than the four-wheeled cars. As neither draught considerations nor tractive adhesion need be regarded in cable work, wheels, therefore, of smaller diameter are not only possible but desirable.

THE POWER PLANT.

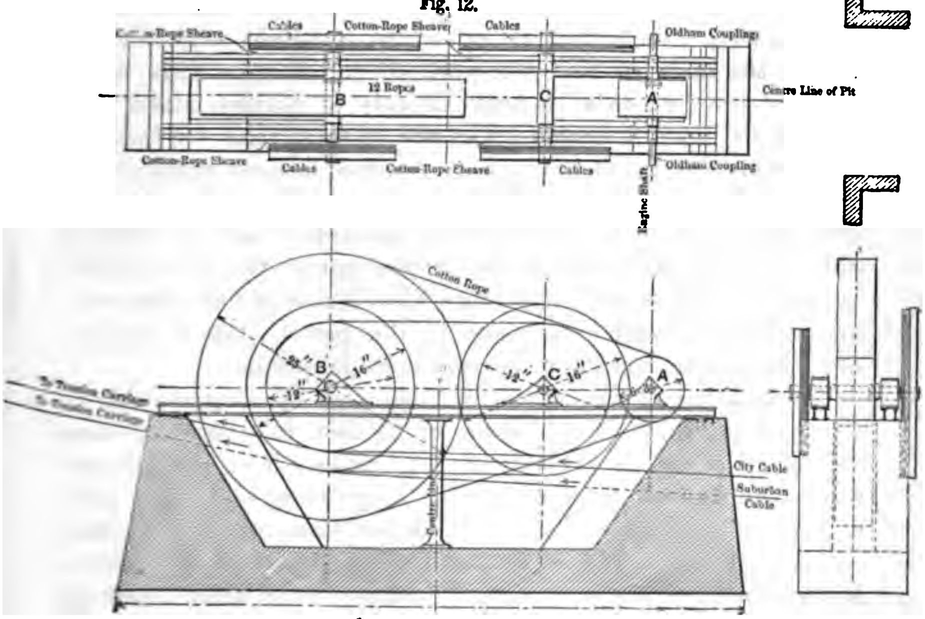
The limitations set by the directors were not unusual—a power house and equipment to be supplied at mininum cost, yet as designed throughout to be not only satisfactory, but, if possible, ornate.

The engines at once contracted for were of the Corliss type, one being 18 × 42 and the other 22 × 42. Compound engines would have been considered were it not for the fact that the expense of the engine plant was already heavy, due to the necessity for two separate engines, where but one was in use with the other in reserve. High-speed engines might have been desirable for their well-known advantages, but in cable work they are practically inadmissible, as the reduction in speed must be considerable, in order to bring the speed down to the comparatively slow speed of the cable. Even with the slower-going Corliss, in the case before us, the reduction in velocity ratio between engine-shaft and cable-winder was as four to one.

The steam-plant of the power station consists of four steel shell boilers, 62″ × 15′, arranged in batteries of two. Tubes are 4″ in diameter. The smoke flue is taken over the tops of the boilers to the rear wall of the boiler room, and thence to the stack, which is located on a higher level in the car-yard of the power house. The boilers are capable of furnishing steam at 100 pounds, and not more than two or three boilers are expected to be in use at the same time, thus affording a reserve for alternation in case of the break-down of any one of them.

The disposition of engines is such that either one can be coupled to the intermediate transmission shaft. This inter-

Fig. 12.



GENERAL PLAN OF WINDERS.
Scale, $\frac{1}{16}$"—1 foot.

mediate shaft is shown at *A* in Fig. 12. This figure also shows the general skeleton on which all the details of the winding machinery have been built. Of the two principal methods of driving cable-winding machinery, one by gearing, and the other by belting, the latter has been adopted as being the one nearest fulfilling the local requirements. The belting from the engine-shaft to the main winding-drum is effected by means of cotton ropes. This system of power transmission, although so much despised by some engineers, yet for cable work offers decided advantages. Foremost of all, it is noiseless, which is more than can be said of most cable plants furnished with cog-wheel gearing, but in justice to later builders of gears it must be said that some of the recent large-cast gears are models of quietness and perfection. Cotton-rope transmission also has the one greater feature of ease in renewing any part or parts without disturbing the action of the remainder. Any rope can be cut out, or a fresh one turned in, while the machinery is in motion. The foundation plan for the winding machinery is perhaps rather new in this class of work. The main foundation consists of a mass of concrete, represented by the sectioned portions of the drawing. Each end is raised to a height sufficient to receive the ends of the iron-work of the frame. This main frame of the "winders" is built entirely of heaviest sized sections of 12″ steel I beams. The use of steel thus for foundation beams was found not only to conduce to most satisfactory economy, but to a lightness of structure which would be impossible with cast iron, and rendered possible a form of design which brought all parts of the mechanism to view at a glance —a feature of no small importance when directors had requested that all mechanical movements of the power station should, whenever convenient, be made visible to the public.

The diameter of the cotton-rope drum on the engine-shaft is 6′ 3″. The large driven drum is 25 feet in diameter, thus reducing the engine revolutions four times. The shaft *B*, on which the 25-foot drum is carried, is prolonged at each side beyond its bearings, and carries a 12-foot diameter cable sheave on one end and a 16-foot diameter cable sheave on the other. Midway between the engine-shaft and the large drum shaft is placed the shaft *C*, carrying the cable "idlers."

The method of pulling the cables can now be seen. The line marked "city cable" is at the right hand of the drawing, and its motion is in the direction indicated by the arrow. This cable is drawn up on the cable sheave in its first groove, the cable passes but half a revolution about this sheave, and is carried off at the top to the idler sheave on shaft *C*. The centre line of the groove in the idler wheel is half the groove pitch behind the groove in the other sheave, as will be seen in a following drawing. After passing half a revolution around the idler it again passes to the other sheave. This constitutes one wrap, and on roads performing ordinary service from three to four wraps are considered necessary. After the last wrap the cable

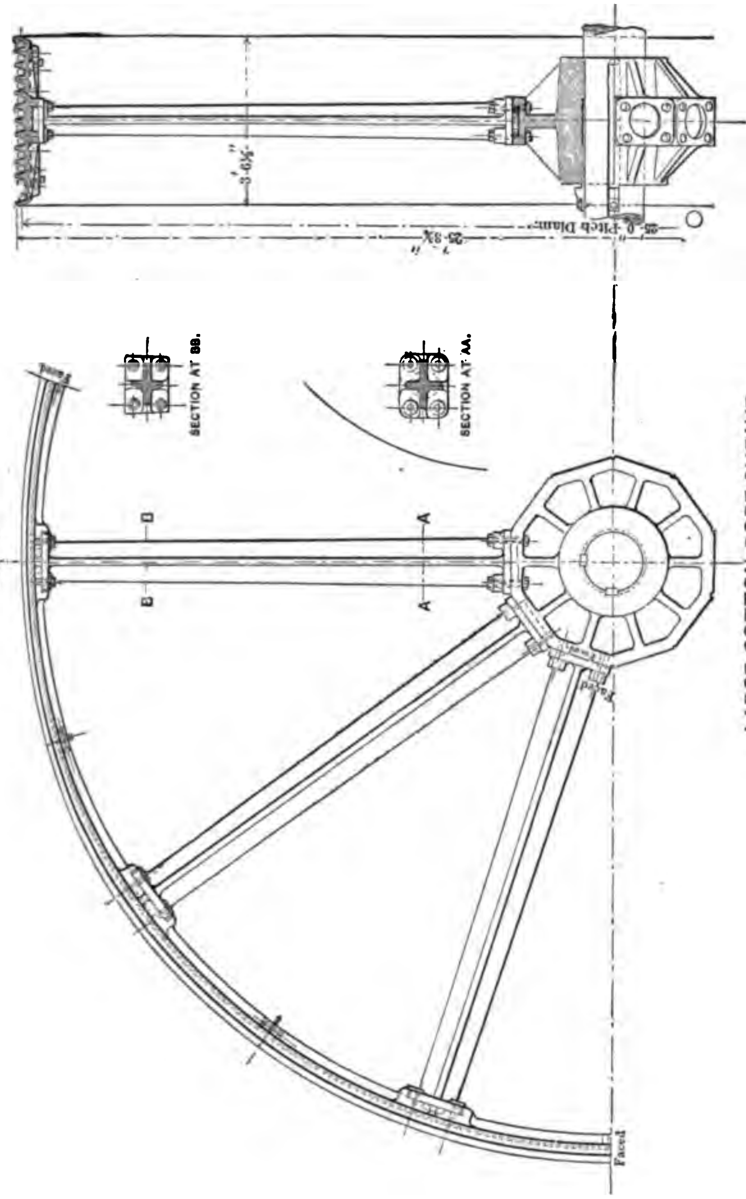


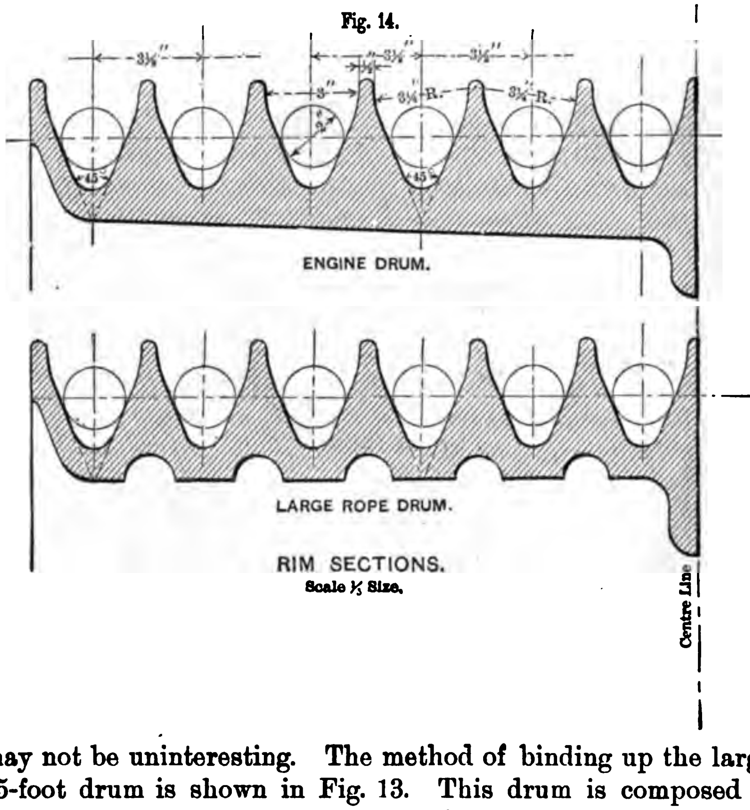
Fig. 13.

LARGE COTTON-ROPE SHEAVE.
Scale; ¼″ = 1 foot.

passes to the tension carriage, and thence is returned to the road. The city cable, it will be noticed, is wound about sheaves which are 12 feet in diameter, the speed of the engines being so proportioned as to give this cable the uniform speed of eight miles per hour. The "suburban cable," on the other side of the foundation, passes to sheaves on the same shafts which are 16 feet in diameter. Thus, with the same shaft and engine speed it will be observed that a speed of twelve miles per hour is given to the cable. As horse-cars average but six to seven miles per hour, this twelve-mile speed is far better rapid transit.

Some of the details of these large wheels and forms of grooves

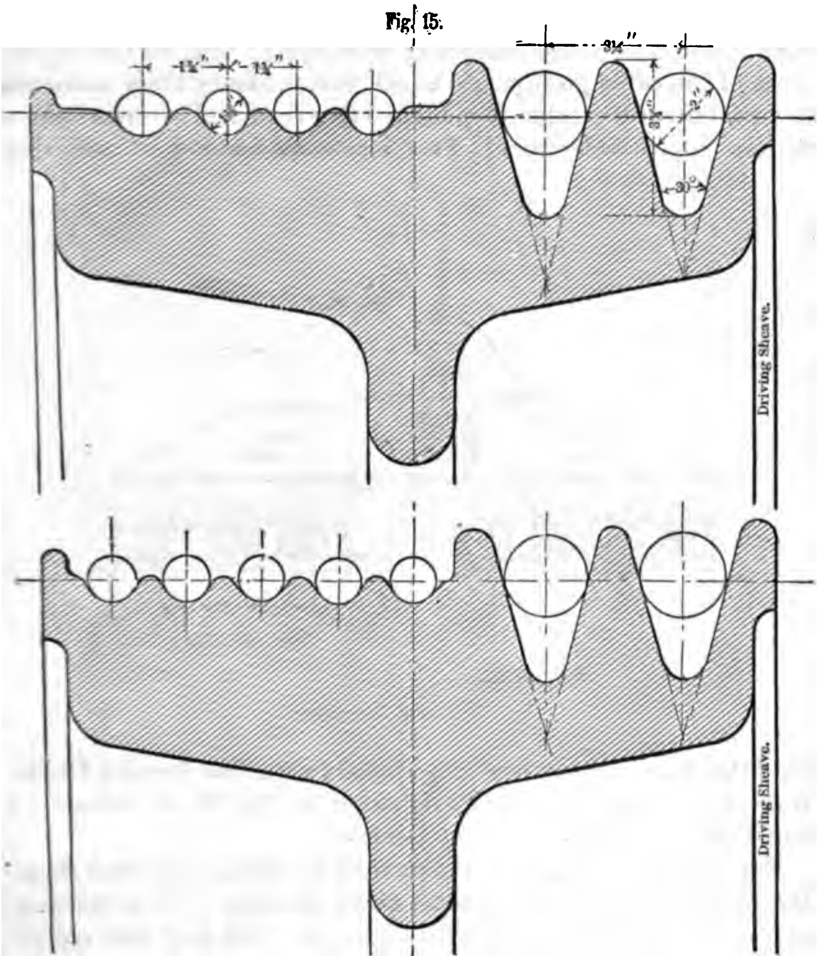
Fig. 14.



RIM SECTIONS.
Scale ⅓ Size.

may not be uninteresting. The method of binding up the large 25-foot drum is shown in Fig. 13. This drum is composed of ten separate segments and ten spokes bolted together. Recourse was again had here to the exercise of economy, and all parts were proportioned as lightly as strength and stiffness would warrant. A surprising amount of metal was saved without sensible decrease of strength by simply grooving out the backs of the segments to correspond to the ridges formed by the rope grooves on the front. The form and angle of the cotton-rope grooves will be seen on reference to Fig. 14. The form of grooves carrying the cable, or the grooves on which is thrust the entire work of pulling the cable, is represented in Fig. 15. The driving sheave is shown with five grooves and the driven sheave with four. This difference is necessitated by the driving sheave having the cable touch

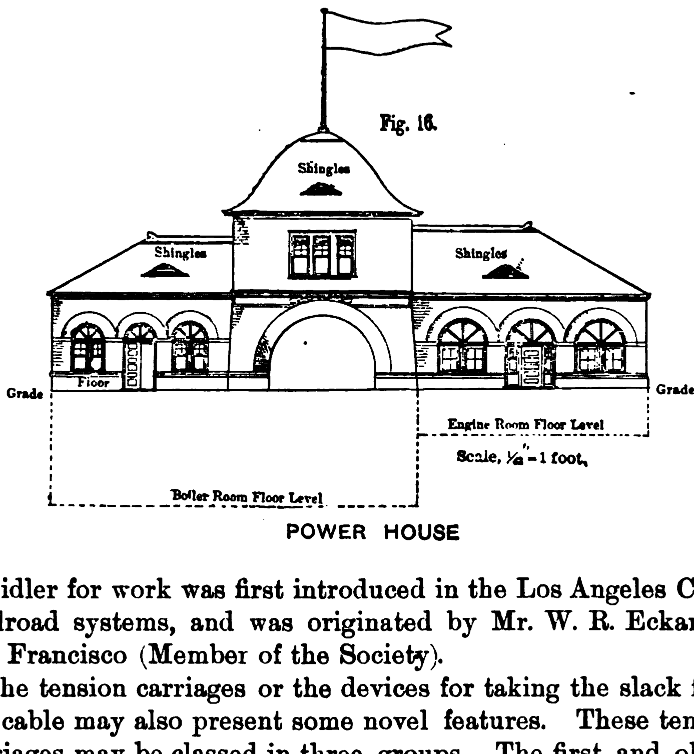
Fig. 15.



TIRE SECTIONS OF WINDERS.
Scale ½ Size.

it in the last groove before the cable passes to the tension carriage. It will also be observed that both driving and driven cable sheaves are provided with larger and deeper grooves for carrying two cotton ropes. The driven sheave has here been termed an idler, but, properly speaking, it is not an idler, and to relieve it from its reputation of idleness provision has been made by means of these grooves and cotton-rope transmission for it to do its share of driving effort. Two cotton ropes pass from the driving sheave directly to the driven sheave, and, as the latter is one-fourth of an inch smaller than the former, it causes in the cotton rope a tendency to hurry the driven sheave more than it would do if the diameters were equal; and, as the diameters of the cable grooves are equal, this tendency then amounts to a pull upon the cable, hence the idler is caused to perform a certain appreciable work. This ingenious method of utilizing

Fig. 16.



POWER HOUSE

the idler for work was first introduced in the Los Angeles Cable Railroad systems, and was originated by Mr. W. R. Eckart, of San Francisco (Member of the Society).

The tension carriages or the devices for taking the slack from the cable may also present some novel features. These tension carriages may be classed in three groups. The first and oldest form, and the one now considered most unsatisfactory, was an arrangement whereby the tension carriage, carrying its own or added weight, hung in a loop of the cable, so that its movements took place in an inclined plane. The second class of tension devices consists of the carriage placed in the loop, but travelling on a horizontal tramway, the weight, hanging in a pit, being attached to the carriage by cables or chains; the great disadvantage

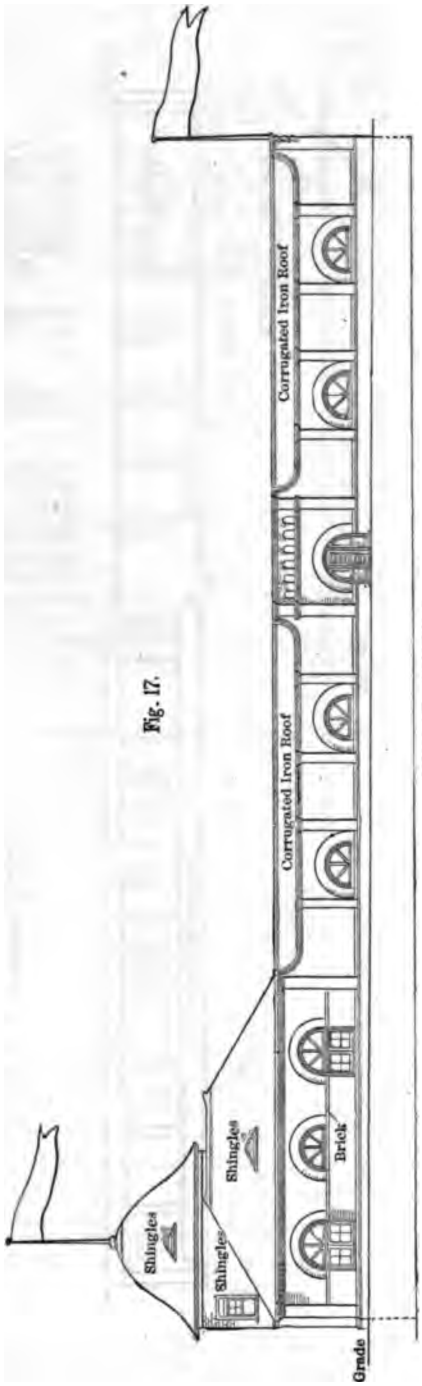


Fig. 17.

POWER HOUSE.
Scale, $\frac{1}{32}''$=1 foot.

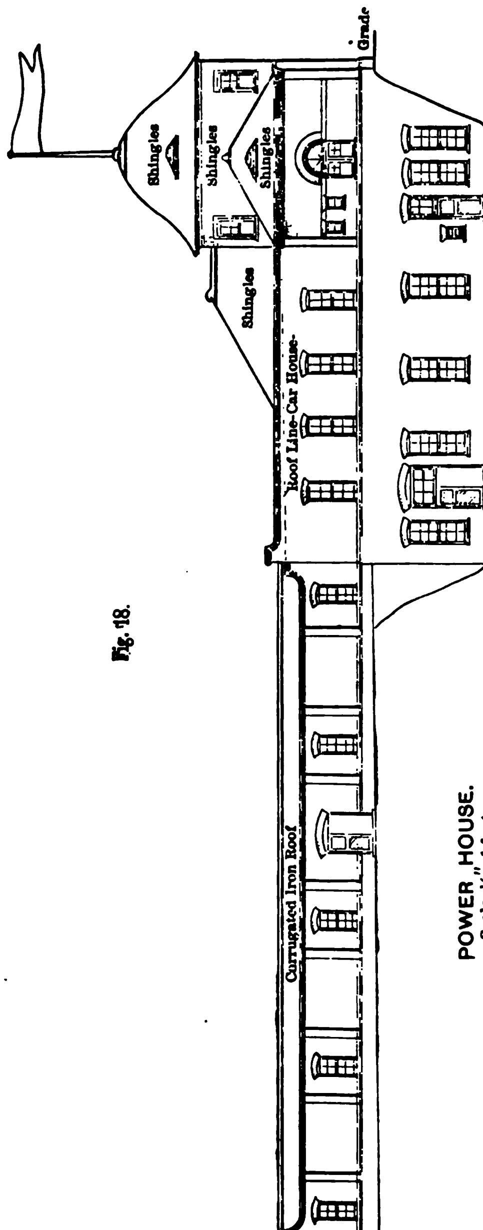
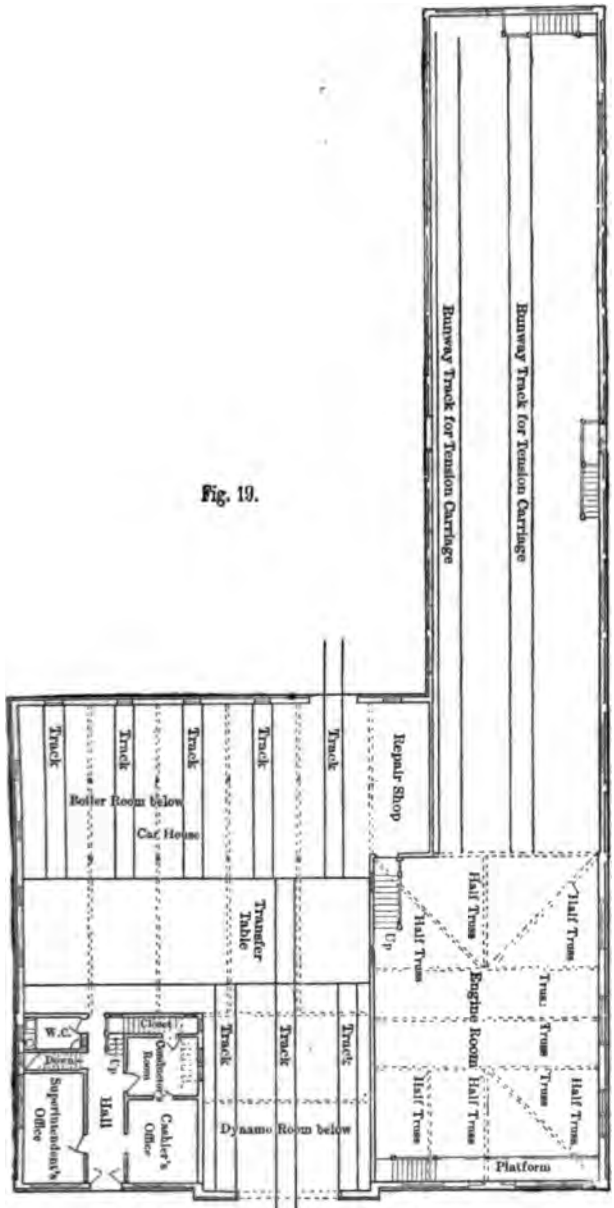


Fig. 18.

POWER HOUSE.
Scale, 1/16" = 1 foot.

Fig. 19.



FIRST FLOOR PLAN
Scale, 1/32"=1 foot.

of both of these systems being that, should the cable suddenly give way (which is a rare but still possible occurrence), the weight and carriage instantly rush to the end of their respective ways to their own demolition.

The third class of tension carriage consists of a double carriage, one moving upon the other. The upper carriage holds the cable sheave. The weight is attached by chain to this carriage, but this chain passes over pulleys on the under carriage. The weight is therefore carried by the carriages themselves. Usually this weight hangs suspended under the carriage while the carriage itself travels along a runway or trestle. In the San Diego pattern, which is adapted from the last form, the weight hangs within the under carriage; thus the tension carriageway need only be an ordinary track laid upon the surface of the floor or ground.

The architectural appearance of the power station is seen in the general views as shown. Although the building was by no means expensive, yet the desire was to have it of such tasteful appearance as would not render it an eyesore to the residence part of the city in which it was located. The architect, Mr. Wm. H. Hebbard, having had valuable experience in designing the architectural details of the three ornamental and most satisfactory power stations of the Los Angeles Cable Railway Company, came to this San Diego work with a complete knowledge of what such a building demanded, devoted to boilers and engines, to rolling-stock and repair shops. Views of this power building are shown in the diagrams here given.

DISCUSSION.

Prof. Jas. E. Denton.—I would like to know what the grades are which this road can handle and is handling?

Mr. Frank Van Vleck.—In reply to the question of Professor Denton, it may be said that cable roads may be operated over any grade, even to a vertical, and attention may be called to the fact that ordinary passenger elevators are but vertical cable railways. The grades on the road in question are but eight per cent. on the steepest incline, with a number of short grades of but six per cent.

CCCCXVII.*

CHIMNEY DRAUGHT—FACTS AND THEORIES.

BY ROBERT H. THURSTON, ITHACA, N. Y.
(Member of the Society and Past President.)

THE theory of chimney draught and the question whether there exists a temperature of maximum delivery of gas, as indicated by the theory of Péclet and Rankine, have been frequently discussed, and, so far as the mathematical treatment of the Rankine equations is concerned, may be considered as fully settled. It has seemed to the writer that it may be a good time to import a few facts, and to ascertain by direct experiment to what extent the purely mathematical treatment accords with such facts; whether the premises on which it is based are those of practice, and whether the conclusion reached has any value in the practical work of the designing and constructing engineer. He therefore planned an investigation which seemed to him likely to give a correct solution of the problem from the standpoint of the engineer, and to settle at least the question whether, in ordinary practice, there exists a temperature of maximum delivery within the range customarily observed in temperatures of chimneys as attached to the steam-boiler. This investigation was made during the spring of 1890, and the following is an account of it and its results. The work was done and all observations made by Messrs. W. S. Monroe and E. C. Fisher, who are to be credited with many ingenious devices eliminating obstacles or directly contributing to the attainment of the simple results sought, and to whom the writer is under great obligations.

The stack employed was 20 inches in diameter and 32 feet in height, circular in section, composed of a single thickness of boiler plate, lined throughout with fire-brick. According to the usual computations, it should be sufficient to produce a draught of about 0.3 inch of water and a rate of combustion of not far from twelve pounds of good anthracite coal on the square foot of a grate eight times its own area, each hour. A grate was con-

* Presented at the Richmond Meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

structed especially for this work, however, and, to eliminate all uncertainty as to resistances, was built so as to be placed directly at the foot of the stack and to be removable at pleasure. The instruments employed were standardized Bulkley pyrometers in the stack, an accurate anemometer to measure the rate of flow of the air into the fire, and a very finely divided water-pressure gauge, made for the writer by the Hartford Steam Boiler Insurance Co. These instruments were standardized before and after use. For exceptionally high temperatures a Pouillet pyrometer was employed.

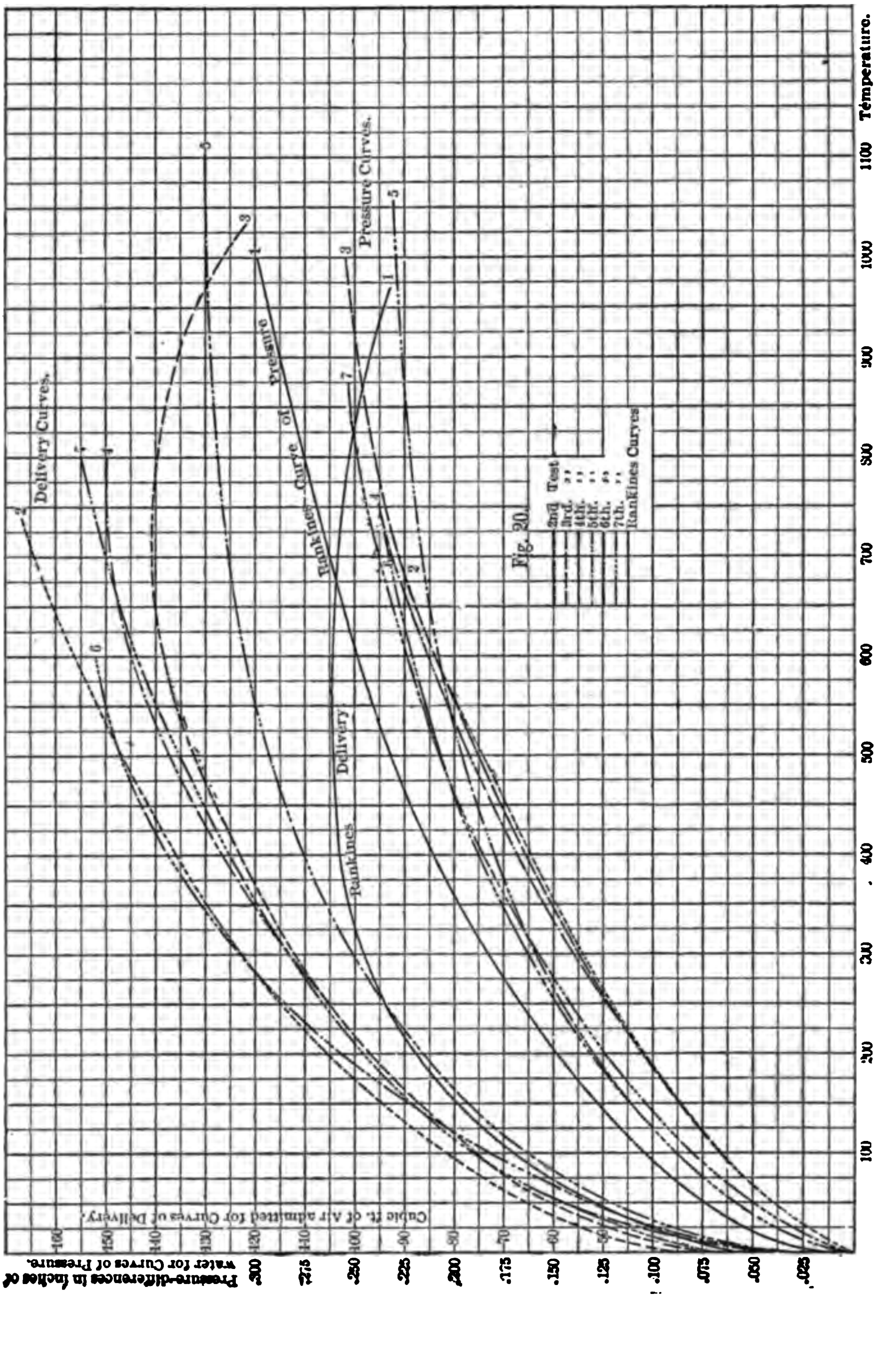
Preliminary tests were made to discover any conditions which might lead to uncertainty or embarrassment; and the work was only begun formally when it was found practicable to eliminate such obstacles to success. Heavy fires of coke were first used, the purpose being to secure, if possible, a uniform and manageable variation of chimney temperatures. It was found that lighter fires and the best of coke were requisite. It was found impracticable to keep uniform resistances of fuel-bed by attempting to fire regularly, as a very slight variation in the condition of the fire made an enormous difference in the resistance to flow of the gases passing through it and the stack. It was as difficult to secure perfect uniformity of thickness of a burning fire as of condition of the fire itself. It was finally discovered that the best method of securing uniform resistance was to allow a heavy fire to burn completely out, leaving a bed of ash, which remains unchanged, substantially, throughout the period of temperature and air measurement, and gives invariable resistance during that time. The temperatures of the stack ran down rather rapidly at first, then more slowly after the fire had burned out, and always so steadily and smoothly that it was practicable to secure perfectly satisfactory measurements. In some instances, however, another system was tried: The fire was entirely removed and the resistance was adjusted and kept constant by applying a closed box at the base of the stack, openings in which permitted the desired quantity of air to draw through, the only variable being then, as required, the falling temperature of the stack. Where the fire was used, or the ash-bed employed, the condition of the grate could be examined by an opening in the side of the stack.

During each test five-minute runs were made with the anemometer while taking the temperature and water-pressure readings, and observations were made at frequent intervals. Ordinarily the temperature in the stack fell rapidly to 500° Fahr., and then it required a number of hours to bring it down to 100°. The data thus secured are tabulated and are given in the appendix to this paper. Their study will probably settle any question which may arise relating to the subject thus investigated. Some discrepancies will be discovered, due to error in judgment as to the time of expiration of the fire, but none which can affect the general conclusions or the value of the data for the purposes of the engineer. The best method of examination, however, as in all such work, is that in which graphical representation is made of the data, and the curves thus produced are studied to ascertain the law of variation of the variable quantities. Such curves are reproduced in the accompanying plate (Fig. 20).

Examining the plate, it is seen that the curves of pressure differences and of delivery of gas range up to maxima in some cases, and to temperatures, above those of the maxima, from 800° to 1,100° Fahr. All represent work done in the manner above described, the same process being followed in all, except in the second, fifth, and sixth trials. In the second and sixth it was attempted to secure a constant resistance with "dumped" fires, the resistance being maintained by the use of a closed box at the base of the stack. The only defect of these records is the low temperatures which were naturally obtained. The fifth test was made especially to reach high temperatures; but it was less successful than had been hoped, though the pyrometer reached a maximum of 2,200° Fahr., and maximum delivery at 1,150°.

The first of these graphically recorded facts to attract attention is that of the Rankine curves (1), falling between the two groups which exhibit, the one the pressure differences, the other the volumes and weights of air, as measured by the anemometer. A second interesting fact is the variation of the temperature of maximum delivery, not only as between Rankine and actual, but among the various sets of data illustrated by the several curves derived by experiment. The lowest pressure curve (5), obtained by the purely artificial resistance of a throttled ingress of air, gives the highest temperature figures but the lowest delivery, and a maximum which, instead of being found at the same point as that of Rankine, or with any other case taken, is evidently far outside them all. Here a low delivery is given by high temperatures and low pressure differences. The obvious explana-

CHIMNEY-DRAUGHT PRESSURE-DIFFERENCES AND DELIVERY.

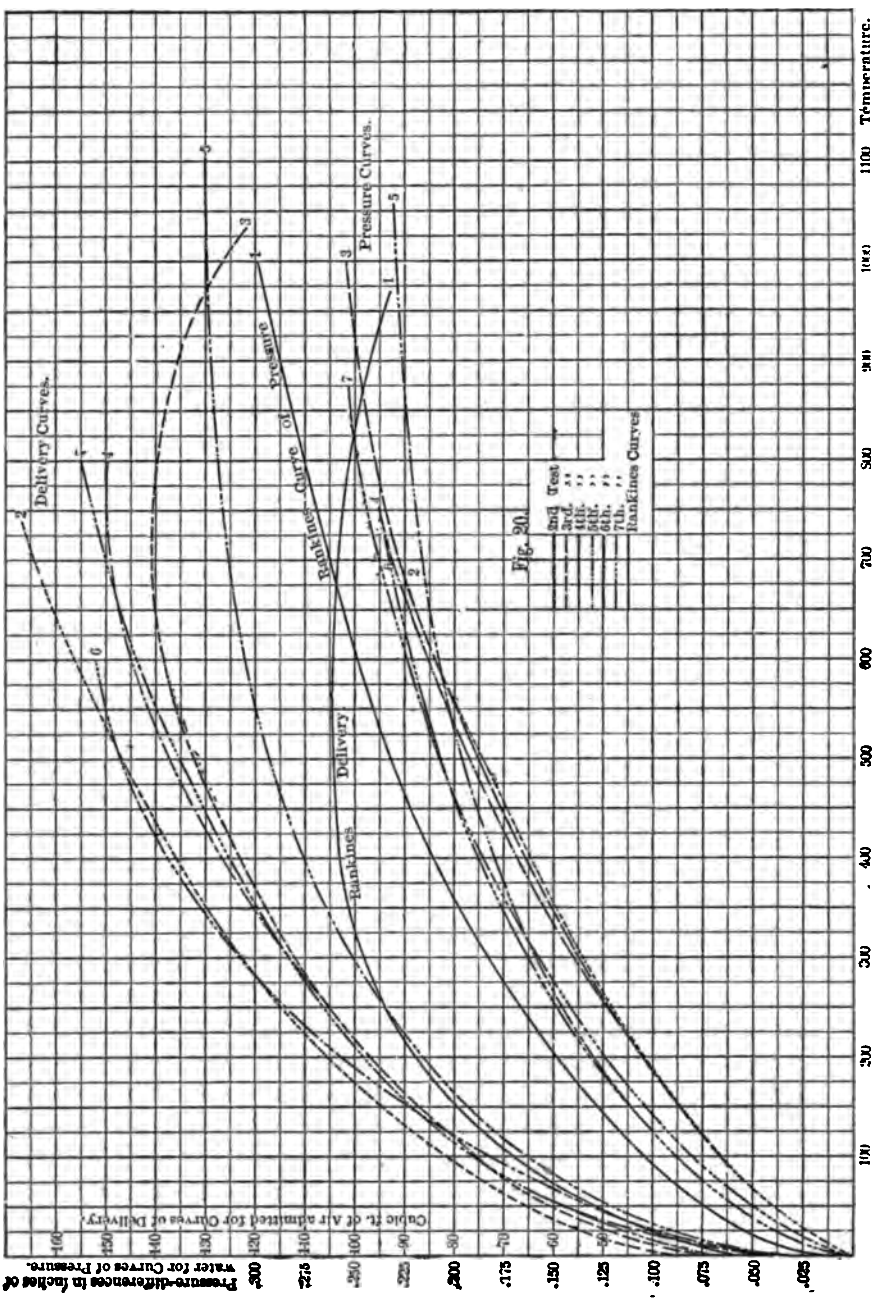


tion is that the friction and inertia resistances were low. In (2) and (3) the pressure differences are substantially the same, but the delivery very much greater in the former, which has no maximum within the range attained, while the latter passes a well-defined maximum at about 700°. There is a general resemblance between both sets of curves, as obtained by experiment, and the corresponding curves deduced from Rankine; but the differences among themselves and between the mean of either group and the typical Rankine curve would indicate that the maximum indicated by the analysis, as ordinarily given, is a variable quantity and dependent upon the variable conditions of draught and combustion. The curves here given are curves of volumes of air passed through the fire; but the division of these volumes by the density of the gas, as determined by temperature, will give the weights discharged, and, if desired, curves of weights may be thus obtained of a similar nature.

The essential facts are here given, however, quite as well. These are: (1) That the resistances and pressure differences in the chimney are lower than those computed from the temperature variations by the usual method; (2) that the delivery is greater, with a given resistance, than is thus computed; (3) that the temperature of maximum delivery is a variable quantity; (4) that the chimney-temperature of maximum delivery is far above that ordinarily taken as that of good practice in the operation of the steam-boiler.

These facts are evident at a glance from the forms of the curves and their location relatively to one another, and to the Rankine delivery curve. The reasons of these facts, and of the discrepancy between the deductions from analysis, as commonly made, and the actual operation of the boiler and chimney, are not difficult of determination. It is at least here seen that the interpretation of the algebraic expressions of accepted theory require modification. In general, it would seem that the delivery of any given chimney under stated conditions of temperature and resistance is probably greater than would be computed, using the formulas of Péclet and Rankine, those expressions being thus found to be in error on the right side, so far as these experiments give fair basis for judgment. Within the usual range of temperatures of chimney in steam-boiler practice, the excess would seem to be from 25% to 50%, being largest at the highest temperatures of flue. The maximum delivery takes place ap-

CHIMNEY-DRAUGHT PRESSURE-DIFFERENCES AND DELIVERY.



tion is that the friction and inertia resistances were low. In (2) and (3) the pressure differences are substantially the same, but the delivery very much greater in the former, which has no maximum within the range attained, while the latter passes a well-defined maximum at about 700°. There is a general resemblance between both sets of curves, as obtained by experiment, and the corresponding curves deduced from Rankine; but the differences among themselves and between the mean of either group and the typical Rankine curve would indicate that the maximum indicated by the analysis, as ordinarily given, is a variable quantity and dependent upon the variable conditions of draught and combustion. The curves here given are curves of volumes of air passed through the fire; but the division of these volumes by the density of the gas, as determined by temperature, will give the weights discharged, and, if desired, curves of weights may be thus obtained of a similar nature.

The essential facts are here given, however, quite as well. These are: (1) That the resistances and pressure differences in the chimney are lower than those computed from the temperature variations by the usual method; (2) that the delivery is greater, with a given resistance, than is thus computed; (3) that the temperature of maximum delivery is a variable quantity; (4) that the chimney-temperature of maximum delivery is far above that ordinarily taken as that of good practice in the operation of the steam-boiler.

These facts are evident at a glance from the forms of the curves and their location relatively to one another, and to the Rankine delivery curve. The reasons of these facts, and of the discrepancy between the deductions from analysis, as commonly made, and the actual operation of the boiler and chimney, are not difficult of determination. It is at least here seen that the interpretation of the algebraic expressions of accepted theory require modification. In general, it would seem that the delivery of any given chimney under stated conditions of temperature and resistance is probably greater than would be computed, using the formulas of Péclet and Rankine, those expressions being thus found to be in error on the right side, so far as these experiments give fair basis for judgment. Within the usual range of temperatures of chimney in steam-boiler practice, the excess would seem to be from 25% to 50%, being largest at the highest temperatures of flue. The maximum delivery takes place apparently, in cases of which these tests may be taken as fairly representative, at higher temperatures than are given by the theory. In case 2 the stack was left entirely open and free; but the maximum is not reached at the highest point, notwithstanding the fact that here the resistance was purely chimney friction. The same is true of curves 4 and 7; and it seems only approximated in cases 3 and 5 and 6 at somewhere about 700° in the lowest instances, and beyond 1,100° Fahr. in No. 5, the highest. In No. 7 the maximum had not even been brought into sight at a temperature of 800°; but, as in case 2, the curve was still rising steadily at the last observations, and its outline, so far as traced, would appear to indicate a maximum far beyond the red heat. No. 3 is the only curve of the whole collection which follows very closely the general shape of the Rankine curve and attains its maximum, clearly and definitely, well within the range of temperature obtained; and even this gives that maximum far above that of the Rankine curve, at above 700° instead of below 600°. These differences are probably due to the comparatively large resistance due to friction of flues in the cases on which that analysis and its constants are based in large part if not wholly.

The formula is, however, obviously inaccurate in form. The head is assumed to be proportional to the square of the velocity of the gases in the chimney-shaft; and it is further assumed that the resistances in the chimney due to friction, that due the inertia of the mass of air set in motion, and that due to the resistances at the grate and in the boiler, are all similarly variable. This would seem to be an entirely false set of assumptions. It seems to the writer certain that the rates of flow of the gases in the chimney, in cases of heavy fires and large grate resistance, are so slow, and the resistances within the bed of fuel so great, with respect the one to the other, that the rates of flow must be dependent practically upon the pressure-head and to but an insignificant extent upon the friction in the chimney-shaft. In such cases, it is evident that the rate of introduction of air must be simply proportional, or nearly so, to the square-root of the head, or of the temperature-differences, and that no maximum delivery temperature can exist. Where the temperature similarly varies in a shaft in which the resistance of the chimney friction is, if such a case be possible, the main source of obstruction to flow, it can be seen, from the method of analysis pursued by Rankine, that it may happen that the density may diminish, at high velocities, so much more rapidly than the rate of flow can increase that the maximum may appear. In such cases it will evidently be found, at least approximately, where Rankine places it.

In the formula of Rankine the coefficient G, taken as a measure of the fuel-bed resistance and assumed to be a constant, is probably a variable of wide range of value. It is obvious that in the series of experiments here recorded its value was never very nearly that obtained in the work of Péclet, whose grate resistances were probably much larger than any of these here given. To what extent this is true may be ascertained, where such resistances have been measured, by comparing the water pressure with those here recorded. Until the character and magnitudes of this method of resistance have been ascertained, no exact analysis is possible. Meantime, the well-known results of the experiments of Clark and Isherwood and numerous later investigators have given the designing engineer enough of fact and sufficient variety of condition to enable him to proportion his chimneys and stacks satisfactorily. He has never used the theoretical method and evidently is not likely to do so. His methods are empirical, and must remain so for the present, probably.

It needs no experiment to show that it is impossible that there should exist, in ordinary steam-boiler practice, a definite and constant temperature of maximum weight-delivery of gases from any chimney, such as has been inferred from the mathematical treatment here referred to. The simple fact that it is well known that in hundreds of cases—in thousands, probably—temperatures exceeding that stated temperature of maximum draught have been observed with "natural" draught. Since the temperature of chimney in any boiler plant is a direct function of the quantity of coal burned on the grate; since the quantity so burned depends, in turn, on the draught; and since maximum temperature and maximum draught and maximum combustion all find a common maximum synchronously, it is obvious that, were the proposition attributed to Péclet and Rankine correct, no chimney operating by natural draught could ever exhibit a higher temperature than that of this maximum—600° Fahr., approximately. Passing this figure, the delivery would decrease; the quantity of air supplied the fire would be reduced;

the diminished draught would give cooler gases; and the temperature of chimney would at once fall to some lower point; 600° Fahr. being thus the highest attainable temperature with natural draught. The discovery of any one case in which the boiler chimney is delivering gas at a higher temperature is thus sufficient to prove the fallacy of the proposition.

The error follows so naturally, however, from the analysis ordinarily accepted that it is less remarkable that authors have accepted it and trusted to further investigation to reconcile this apparent conflict of deduction. It appears first in Péclet, but it was dropped out of the last edition, published after his death, and has only held its place, probably, on the authority of that greatest of engineer-authors, Rankine, who adopted it in his work substantially as first asserted by Péclet.

The first direct experiments of which the writer is aware, proving the fallacy of this proposition of a definite temperature of maximum delivery of gas, were those of Engineer-in-Chief B. F. Isherwood, U.S.N., made at the Brooklyn Navy Yard in 1866, and published in a report to the Navy Department "On Experiments with the Horizontal Fire-tube and the Vertical Water-tube Boilers." The boiler used for these special experiments was a Martin water-tube boiler of the type then common in the naval service, and which, as the writer learned by extended experience, gave most excellent results economically, but were somewhat troublesome when the tubes began to give way; and they have, partly for the latter reason, now largely gone out of use. The tube-box of this boiler contained 34 rows of vertical tubes, set lengthwise in the box. These rows of tubes were cut out, progressively, in such manner as to permit a more and more rapid draught, until only 13 were left and the experiment terminated. The following is a tabulated exhibit of the results:

INTENSITY OF DRAUGHT AND OF COMBUSTION.

No. rows of tubes in use.	31	28	25	22	19	16	13
Lbs. coal in 48 hrs.	21934	23555	24455	27240	26078	29736	30533
Lbs. water "from and at 212°," per lb. "combustible" part of fuel	12.55	12.63	12.05	11.77	11.69	11.20	10.69
Temperature, chimney, Fahr.	416°	480°	491°	600°	700°	700°	770°

The figures above 600° are approximate, and sufficiently accurate for present purposes. The fires were kept about 9 inches thick, and the fuel was ordinary anthracite, such as was customarily used in marine boilers. The volume of air was about double the theoretical minimum required for perfect combustion.

It is obvious from the above that more air must have been sent up the chimney at 700° and 770° than at 600° Fahr., since more coal was burned by 10%. The temperature of furnace being taken at a common figure for such cases, as determined by frequent measurements in Mr. Isherwood's many experiments, at 1,800° Fahr., the loss at the chimney, comparing, for example, the last experiments with the first, is nearly proportional to the reported diminution of evaporative efficiency, showing the combustion to be substantially as good at one rate as at another.

The conditions involved are so numerous and so variable in practice that the writer seriously doubts if any analysis can be made which will be sufficiently simple and convenient, or so accurate, as to come into general use in the solution of this problem. The facts above presented are given as merely an initial step in this research, and only repeated experiments, under all the representative conditions of common practice, can finally determine just how far algebraic treatment can be made useful in representing the law and the facts. Meantime, the following seems to the writer a more satisfactory and probably more nearly exact treatment than that which has hitherto been accepted:

The working conditions affecting draught in chimneys are obviously something like the following: Assume an ordinary boiler and furnace, having, we will say, 20 square feet of grate surface, 600 square feet of heating surface, a chimney 60 feet high, with a flue of 3 square feet area and set close to the boiler. The cross section of flues, the "calorimeter," as it is sometimes though incorrectly called, may be taken as 4 square feet; the tubes being 15 feet long, and the boiler an ordinary "cylindrical tubular," set in the usual manner, the gases going back from the furnace to the rear, thence through the tubes to the front, and then up chimney. Anthracite coal, "nut" size, is burned at the rate of 12 lbs. per square foot of grate per hour, evaporating a total of 2,160 lbs. of water into dry steam each hour, and thus rating the boiler at 72 H.P., on the usual conventional basis.

The process of draught is essentially this: Air to the amount of about 18 lbs. per pound of fuel burned, or 4,320 lbs. per hour as a total, enters the ash pit and passes through the fuel. In traversing the 6 inches thickness of the coal on the grate its temperature is raised from that of the external air to 2,500° Fahr., and its volume increased 6 times nearly, or to about 1,400 cubic feet per pound of fuel, from a total volume per hour of 57,600 cubic feet to about 346,000. This immense mass of air drifts through the boiler flues and tubes and up the chimney, its temperature gradually falling, in decreasing progression, within the boiler to about 500°, which temperature it substantially retains as it passes through the chimney and into the atmosphere. In passing through the fuel-bed its temperature is raised in proportion to the increase of its volume, and while within the porous mass of coal. This sudden expansion, occurring, as it does, within a distance of 6 inches and less than half a second, means the acceleration of its velocity from about 1 foot below to about 6 feet per second above the grate, or at the rate of 10 feet per second, multiplied by the ratio of reduction of section of current in the mass of fuel, the maximum being, perhaps, many times greater. The resistance to this rapid acceleration, added to the frictional resistances of the enormous extent of air-passages in the mass of fuel, and of the flue and tubes and chimney, constitutes the total resistance to be overcome by the difference of head of the hot gas and the external air, reckoned from the grate level to the chimney-top. The cooling of the gases in the boiler reduces the volume flowing at the chimney to about 100,000 cubic feet per hour, and its rate of flow is thus made, finally, only about 9 feet per second. The energy of acceleration is thus largely regained, but the loss due to this acceleration, by friction within the fire, is not compensated as a resistance; though it has been utilized, in a certain sense, by its production of equivalent heat, and resulting elevation of temperature of fire and of gaseous products of combustion.

Of all the resistances to flow, to overcome which a chimney-draught is required, that of the fuel-bed is obviously enormously the greatest. Anthracite coal on the grate weighs about 55 lbs. to the cubic foot; but its specific gravity is about 1.5, or 90 lbs. per cubic foot. It thus contains not far from 40% of its own volume in air-spaces and air-passages, mainly in closed and nearly enclosed spaces. The passages are narrow, tortuous, and of comparatively small total section. What may be taken as the mean value for this reduced section is not ascertained; but experiment shows that the "head of water" corresponding to the case in hand, and measuring the resistance to flow through the fuel-bed, is not far from 0.3 inch, or equivalent, in this case, to a head of about 20 feet of cold air at the temperature of entrance, 60° Fahr., or 40 feet, approximately, of hot air in the chimney. The frictional resistance of the chimney is not far from

$$h = \frac{f l}{m} \frac{v^2}{2g};$$

l being the length of flue from boiler to chimney-top, m the mean hydraulic depth, and $f = 0.012$, a constant coefficient. This gives for the head measured in the cold air less than 1.5 feet, or 3 feet in the chimney, and about one-fourteenth the resistance at the grate. The loss of head in the boiler-flues, similarly computed, is considerably less than the loss in the chimney. Of the total resistance, therefore, the main portion is at the grate and in the fuel-bed.

It is now obvious that the principal resistance being at the fuel on the grate, the rate of flow of the gas through the apparatus and up the chimney will be determined mainly by the rate at which air may be driven through that mass. The velocity in the chimney will be proportional to a function, not of the head due the temperature-differences in the chimney and outer air, but nearly a direct function of the rate of flow of air into and through the furnace as produced by that head. It could only be at exceptionally high velocities of flow in the chimney-flue that its resistance there could become the controlling element in determining the total flow; and such relative velocities could not be obtained by natural draught, ordinarily, but must be found, if at all, in cases of forced and extraordinarily heavy draught, or of very thin or very badly worked fires. The experiments of Clark and of Isherwood, and many others, show the rate of combustion to be substantially a direct function of the head, varying with variation of height of chimney; for example, as the square root of that height. It is readily seen that, this being the fact, the heads being produced, at constant height of chimney, by variation of temperature-differences, the quantity of air supplied and of fuel burned will be similarly a direct function of head, and without an algebraic maximum. But when the velocities in the fire and in the flues are directly proportioned, or when the friction of chimney-flue becomes the controlling quantity, as assumed in the hitherto accepted analysis, a maximum must occur; and our records and curves of results show that we have at times (Nos. 3 and 5) probably approximated to that condition.*

To obtain a satisfactory and probably substantially correct expression for the heads producing chimney-draught, in any usual case, we need only reconstruct the Péclet theory in the following manner:

The total resistance is the sum of that due the acceleration of the air flowing from the chimney-top, the friction in the chimney-flue, the resistance in the fuel-bed, and those in the boiler and ash pit. It may be measured either in head of cold air or of hot gases in the chimney-flue; the one is equal to the product of the other into the ratio of temperatures. We will adopt the usual method and measure it in head of hot gas.

Péclet and later writers take this head as equal to

$$\frac{u^2}{2g}\left(1 + G + \frac{fl}{m}\right) = \left(0.96\,\frac{t_1 - t_2}{t^2}\right) H = h;$$

in which u is the velocity of flow through the chimney; G measures the resistance at the grate; f is the coefficient of friction of gas flowing over flue surfaces; l is the length of flue traversed; m is the "hydraulic mean depth" of flues; H is the height of chimney; and t_1 and t_2 are the absolute temperatures of chimney-flue and of external air.†

Of these several terms in the first member of the equation, the first is the resistance due to acceleration; the second and third the friction resistances in furnace and flue. The first and last are obviously direct functions of the same quantity, u^2. Their

* This is by no means certain, however, as it is not improbable that the apparent maximum in Nos. 3 and 5 may be due, in part, if not wholly, to a decreasing resistance at the grate, caused by an undetected consumption of fuel supposed to have been completely burned before the record was begun. The fact that, in No. 2, in which the grate was removed and the stack left open, all resistances being those of chimney-friction, no maximum was found, is particularly significant.

† Rankine, *Steam Engine*, 1859.

values are expressed with sufficient accuracy for our present purposes by the usual measures, $f = 0.012$ being the coefficient of friction found by Péclet. For square or circular flues m is one-fourth the diameter, or the side. The second term is, however, not so evidently a constant factor of energy of gas-flow in the chimney. It is easy to see that it is probably variable in all cases, and that its value varies approximately inversely as the absolute temperature of chimney.

In all cases of good practice the air entering the fire is raised to the same maximum temperature, whatever its volume or weight. Practically complete combustion can be secured at any usual rate of chimney-draught. This being the case, the maximum temperature of the fire is substantially the same for all ordinary good practice, and its variation is dependent not upon the chimney temperature, but on the proportion of air admitted to dilute the products of combustion. Assuming that the cases considered thus represent good practice, or the same quality of practice in this respect, *the movement of the air through the fuel-bed and grate is effected at temperatures absolutely independent of the chimney temperatures*, and, in the assumed cases, constant. The resistance at the grate is proportional to the mean square of the actually occurring velocities at the thus fixed invariable temperatures, and the value of G thus becomes $\frac{a^2 t^2}{t_1^2}$; in which a is a constant easily determinable for any given case, and usually not far from 2; t is the absolute temperature representing the mean within the fire, which temperature is usually, in common practice, probably not far from 2,000° on the absolute Fahrenheit scale, or, approximately, 1,500° on the common scale, and sensibly constant.

The weight of air discharged from the system is measured, at any point in its path, by the product of its velocity and density. Since u is the velocity of discharge at the chimney-top, and as t_1 is taken as its temperature at that point, the weight discharged must be proportional to $u \times \frac{1}{t_1}$; thence

$$\frac{u}{t_1} = \sqrt{\left[2\ gH\left(0.96\ t_1 - t_2\right) \div \left[I + \frac{a^2 t_3^2}{t_1^2} + \frac{fl}{m}\right] t_1 \sqrt{t_2}\right]}$$

in which, if G be taken as constant, a maximum is found when

$t_1 = 2.1\ t_2$, nearly. If in any case, also, the value of G may be neglected beside those of the two other terms, the same result follows. In the actual case the value of G is variable, and the maximum may be found, but at higher temperature than has been previously supposed. It is obvious that where the frictional resistance in the flue is small, as when the flue section is comparatively large, or the velocity of upward flow is small, the maximum is found at a high temperature. The same is true when the resistance of the fuel is large, as when the thickness of the fire is great or the fuel is exceptionally fine or of mixed sizes, while the opposite conditions of rapid flow in the stack, or of very thin fires, produce a maximum efficiency of draught at temperatures more nearly approaching that given by the commonly accepted analysis.

It is thus seen that, if we may adopt the theory here suggested in modification of Péclet and Rankine, both the facts of experiment and the deductions of our theory confirm the proposition that the maximum draught, in the sense in which the term is here used, ordinarily, if not invariably, occurs at a temperature far above that found in good steam-boiler practice, and above that previously supposed; that maximum rising as the resistance at the grate increases and as the resistance in the stack decreases. It is probably rarely found at temperatures less than 700° or 800°, or very probably 1,000° Fahr. (538° C.)

In the example above taken it has been seen that the resistance at the grate is above 90% of the total. In a series of examples reported by Clark this difference is often shown to be enormously in excess of this proportion.* In examples of what may be taken to be common and good practice the rate of flow in the flue is found to be from one-half to one-seventeenth the acceleration due the head, which means that the resistances in the boiler approached three-fourths the total, as a minimum, and attained above 99% in some cases. The usual proportion seems to be about one-eighth the rate of acceleration, or over 98% of total resistance at the boiler. The proportion of the resistance at the fuel-bed to the total is a trifle lower, and it may be stated without much doubt that the resistances in ordinary steam-boiler practice are principally—probably at least 95%—at the grate; that this determines the rates of flow; and that no maximum draught is likely to be approached at any temperatures occur-

* *Steam Engine*, vol. i. chap. xxxiv.

ring in the chimney, the actual maxima falling more commonly at points on the scale between 800° and 2,000° Fahr.

An incidental, but none the less interesting and important, outcome of this investigation is obtained from a comparison of the pressure curves for the several cases given. It is seen that they vary with the quantity of air passing through the stack, and that we have an independent curve for each run, corresponding to each delivery curve. The pressures depend, therefore, on other than temperature-differences; otherwise, were they dependent, as Rankine assumes, on those differences alone, we should have but one curve for the whole series, and that identical with the Rankine curve, which is shown as deduced for the stack here employed. The Rankine curve is higher than the actual curves by a quantity which is substantially the measure of the chimney-friction.

While the facts here revealed as derived from direct appeal to nature in this matter may aid us in our endeavor to learn the true philosophy of the phenomena of chimney-draught, the writer would hardly venture to claim that they will prove a better guide in the design of chimneys than that experience which is now so ample in this field of engineering work. As he has elsewhere stated, current successful practice must be the ultimate and unquestionable guide in all standard and usual practice;* but where exceptional conditions must be met, where no precedent exists, the construction of an exact theory becomes important.

* *Manual of Steam Boilers.* New York: J. Wiley & Sons. Revised edition, 1890.

APPENDIX.

LOG OF TEST TRIALS.—DATA.

FIRST TEST.

MONDAY, *April* 21, 1890.

1st Run : Time, 10.30–10.35. Anemometer, 89351
80856

8495

Cubic feet of air per minute, 126.5.

TEMPERATURE.		PRESSURE.	
Stack.	Room.	Left.	Right.
940	66	1.412	1.154
955	68	1.413	1.150
960	68	1.414	1.151
965	68	1.410	1.157
960	—		
—	68	1.412	1.153
965		1.153	
Corrected,			
1101		.259 Difference.	

1101 — 68 = 1003°.

2d Run : Time, 10.40–10.45¼. Anemometer, 80856
72988

For 5¼ min. = 7868
" 5 " = 7500

Cubic feet of air per minute, 112.11.

TEMPERATURE.		PRESSURE.	
Stack.	Room.	Left.	Right.
980	68	1.410	1.153
990	68	1.412	1.153
990	68	1.412	1.153
980	68		
970	—	1.411	1.153
—	68	1.153	
982			
Corrected,		.258 Difference.	
1131			

1131— 68 = 1063°.

3d Run: Time, 10.51–10.56. Anemometer, 72984
65743

7241

Cubic feet of air per minute, 108.55.

TEMPERATURE.		PRESSURE.	
Stack.	Room.	Left	Right.
850	68	1.401	1.165
840	68	1.410	1.158
830	68	1.410	1.157
830	—		
830	68	1.407	1.160
—		1.160	
836			
Corrected		.247 Difference.	
965			

965 — 68 = 897°.

4th Run: Time, 10.57–11.02. Anemometer, 65743
59129

6614

Cubic feet of air per minute, 105.82.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
815	68	1.412	1.160
815	69	1.410	1.164
810	—	———	———
805	69	1.412	1.162
795		1.162	
808		.250 Difference.	
Corrected, 934			

$934-69=865^{\circ}$.

5th Run: Time, 11.05–11.10. Anemometer, 59129
50278

8841

Cubic feet of air per minute, 131.41.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
760	69	1.410	1.161
750	69	1.409	1.165
740	—	———	———
720	69	1.4095	1.163
710		1.163	
736		.2465 Difference.	
Corrected, 853			

$853-69=784^{\circ}$.

6th Run: Time, 11.55½–12.00½. Anemometer, 950280
940758

9522

Cubic feet of air per minute, 139.73.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
765	73	1.415	1.164
730	74	1.417	1.160
735	—	1.416	1.160
745	74	———	———
740		1.416	1.161
		1.161	
743			
Corrected, 860		.254 Difference.	

$860-74=786^{\circ}$.

7th Run: Time, 12.09–12.14. Anemometer, 940758
931022

9736

Cubic feet of air per minute, 144.21.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
680	72	1.408	1.175
675	72	1.408	1.176
660	73.5	1.415	1.175
650	—	1.410	1.165
645	73	——	——
635		1.410	1.173
—		1.173	
657		——	
Corrected,		..237 Difference.	
763			

763 — 73 = 690°.

8th Run : Time, 12.17–12.22. Anemometer, 931022
921146
———
9876

Cubic feet of air per minute, 146.23.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
615	72	1.415	1.167
610	72	1.400	1.190
600	—	1.407	1.179
590	72	——	——
575		1.407	1.179
565		1.179	
—		——	
596		.228 Difference.	
Corrected,			
693			

693 — 72 = 621°.

9th Run : Time, 12.32–12.37. Anemometer, 921146
911710
———
9436

Cubic feet of air per minute, 139.95.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
500	72	1.400	1.190
495	72	1.395	1.192
490	72	1.397	1.190
490	—	1.390	1.195
485	72	——	——
—		1.395	1.192
492		1.192	
Corrected,		——	
575		.203 Difference.	

575 — 72 = 503°.

SECOND TEST.

1st Run: Time, 3.12–3.17. Anemometer, 911710
900467
———
11243

Cubic feet of air per minute, 165.8.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
730	76	1.452	1.208
715	76	1.445	1.212
700	—	——	——
680	76	1.448	1.210
—		1.210	
703		——	
Corrected,		.238 Difference.	
815			

815 − 76 = 739°.

2d Run : Time, 3.21–3.26. Anemometer, 900467
888621
———
10846

Cubic feet of air per minute, 160.1.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
625	77	1.438	1.220
615	78	1.440	1.216
610	78	1.443	1.212
602	—	1.432	1.225
595	78	——	——
—		1.437	1.218
611		1.218	
Corrected,		——	
710		.215 Difference.	

710 − 78 = 632°.

3d Run : Time, 3.31–3.36. Anemometer, 889618
879156
———
10462

Cubic feet of air per minute, 154.61.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
565	78	1.430	1.228
555	78	1.433	1.226
550	78	1.425	1.232
545	—	——	——
540	78	1.429	1.229
——		1.229	
552		——	
Corrected,		.200 Difference.	
643			

643 − 78 = 565°.

4th Run: Time, 4.05–4.10. Anemometer, 879150
869889
———
9260

Cubic feet of air per minute, 137.42.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
440	74	1.425	1.245
438	72	1.420	1 250
435	74	1.417	1.248
435	75	——	——
430	—	1.422	1.248
——	78	1.248	
435		——	
Corrected,		.174 Difference.	
511			

511 − 78 = 433°.

5th Run : Time, 4.25–4.30. Anemometer, 869853
860816
———
9037

Cubic feet of air per minute, 134.24.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
400	75	1.410	1.260
397	76	1.415	1.253
395	—	1.415	1.253
389	75	——	——
—		1.413	1.255
398		1.255	
Corrected,		——	
469		.158 Difference.	

469 − 75 = 394°.

6th Run : Time, 5.04¼–5.09¼. Anemometer. 860816
852449
———
8369

Cubic feet of air per minute, 124.69.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
335	70	1.400	1.270
335	70	1.400	1.272
333	—	1.407	4.267
332	70	——	——
330		1.402	1.270
—		1.270	
333		——	
Corrected,		.132 Difference.	
395			

395 − 70 = 325°.

7th Run : Time, 5.19¼–5.24¼. Anemometer, 852449
844560
———
7889

Cubic feet of air per minute, 117.81.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
275	76	1.400	1.275
275	76	1.397	1.275
280	—	——	——
280	76	1.399	1.275
—		1.275	
278		——	
Corrected,		.124 Difference.	
333			

333 − 76 = 257°.

8th Run : Time, 7.15–7.20. Anemometer, 844560
837700
———
6860

Cubic feet of air per minute, 103.1.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
205	64	1.395	1.295
201	68	1.295	
200	66	——	
200	—	.100 Difference.	
——	66		
200			
Corrected, 244			

$244 - 66 = 178°$.

9th Run: Time, 8.00–8.05. Anemometer, 837700
835225
——
2475

Cubic feet of air per minute, 40.4.

Temperature.			Pressure.	
Stack.	[Ther.]	Room.	Left.	Right.
82		58	1.480	1.365
			1.365	
			——	
			.115 Difference.	

$82 - 58 = 24°$.

THIRD TEST.

April 22, 1890.

1st Run: Time, 1.14½–1.19½. Anemometer, 0099400
91263
——
8137

Cubic feet of air per minute, 121.37.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
1000	72	1.500	1.244
980	72	1.500	1.244
975	—	1.495	1.247
960	72	1.500	1.243
940		——	——
——		1.498	1.244
971		1.244	
Corrected,		——	
1108		.254 Difference.	

$1108 - 72 = 1036°$.

2d Run: Time, 1.21–1.26. Anemometer, 91263
82963
——
8300

Cubic feet of air per minute, 123.7.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
925	72	1.498	1.244
925	72	1.500	1.245
915	—	1.495	1.244
900	72	——	——
898		1.498	1.244
——		1.244	
913		——	
Corrected,		.254 Difference.	
1052			

$1052 - 72 = 980°$.

3d Run: Time, 1.30½–1.35½. Anemometer, 82964 / 73900 = 9064

Cubic feet of air per minute, 134.61.

Temperature. Stack.	Room.	Pressure. Left.	Right.
850	74	1.495	1.247
830	76	1.493	1.250
820	—	1.490	1.253
820			
810		1.493	1.250
		1.250	
826			
Corrected, 954		.243 Difference.	

954 − 76 = 878°.

4th Run: Time, 1.40–1.45. Anemometer, 73900 / 64190 = 9710

Cubic feet of air per minute, 143.87.

Temperature. Stack.	Room.	Pressure. Left.	Right.
751	74.5	1.492	1.252
746	75	1.488	1.260
735	—		
725		1.490	1.256
		1.256	
722			
Corrected, 836		.234 Difference.	

836 − 75 = 761°.

5th Run: Time, 1.50–1.55. Anemometer, 64190 / 54700 = 9481

Cubic feet of air per minute, 140.6.

Temperature. Stack.	Room.	Pressure. Left.	Right.
670	76	1.485	1.260
665	74	1.484	1.260
660	—		
652		1.485	1.260
645		1.260	
635			
		.225 Difference.	
654			
Corrected, 759			

759 — 75 = 684°.

6th Run: Time, 1.59–2.04. Anemometer, 954709 / 945264 = 9445

Cubic feet of air per minute, 140.09.

TEMPERATURE.		PRESSURE.	
Stack.	Room.	Left.	Right.
610	78	1.485	1.260
605	74	1.488	1.265
600	—	1.470	1.268
595	74		
590		1.481	1.265
585		1.265	
597		.216 Difference.	
Corrected, 694			

694 — 74 = 620°.

7th Run: Time, 2.08–2.13. Anemometer, 945264
985975

9289

Cubic feet of air per minute, 137.81.

TEMPERATURE.		PRESSURE.	
Stack.	Room.	Left.	Right.
560	74	1.478	1.268
553	73	1.477	1.268
549	—	1.480	1.265
545			
538		1.478	1.267
535		1.267	
556		.211 Difference.	
Corrected, 648			

648 — 73 = 575°.

8th Run: Time, 2.54–2.59. Anemometer, 0100135
0091762

8378

Cubic feet of air per minute, 124.72.

TEMPERATURE.		PRESSURE.	
Stack.	Room.	Left.	Right.
415	78	1.460	1.292
410	78	1.460	1.295
408	—	1.464	1.292
405			
404		1.461	1.292
		1.293	
408			
Corrected, 480		.168 Difference.	

480 — 73 = 407°.

9th Run: Time, 3.49–3.54. Anemometer, 91762
84068

7694

Cubic feet of air per minute, 115.02.

Temperature. Stack.	Room.	Pressure. Left.	Right.
345	76	1.450	1.310
345	75	1.454	1.308
343	—		
342		1.452	1.309
		1.309	
344		.143 Difference.	
Corrected, 407			

407 — 75 = 332°.

10th Run: Time, 4.31½–4.37½. Anemometer, 84060
75234

8826 for 6 min.
7355 " 5 "

Cubic feet of air per minute, 110.18.

Temperature. Stack.	Room.	Pressure. Left.	Right.
301	74	1.450	1.320
301	74	1.320	
301	—		
301		130 Difference.	
301			
301			
Corrected, 368			

368 — 74 = 294°.

11th Run: Time, 4.51–4.56. Anemometer, 975234
968125

7109

Cubic feet of air per minute, 105.69.

Temperature. Stack.	Room.	Pressure. Left.	Right.
250	72	1.445	1.333
255	72	1.445	1.334
255	—		
255	72	1.445	1.333
255		1.333	
254		.112 Difference.	
Corrected, 305			

305 — 72 = 233°.

12th Run: Time, 9.08–9.13. Anemometer, 968125
962736

5389

Cubic feet of air per minute, 82.08.

Temperature. Stack.	Room.	Pressure. Left.	Right.
150	66	1.468	1.400
148	66	1.472	1.395
—	—		
149		1.470	1.398
Corrected, 186		1.398	
		.072 Difference.	

186 — 66 = 120°.

FOURTH TEST.

April 23, 1890.

1st Run: Time, 11.54½–11.59½. Anemometer, 960340
950400

9940

Cubic feet of air per minute, 147.16.

Temperature. Stack.	Room.	Pressure. Left.	Right.
750	68	1.623	1.380
730	68	1.620	1.380
722	—	1.616	1.385
725	68		
710		1.619	1.381
		1.381	
727			
Corrected,		.238 Difference.	
842			

842 — 68 = 774°.

2d Run: Time, 12.02–12.07. Anemometer, 950400
940200

10200

Cubic feet of air per minute, 150.88.

Temperature. Stack.	Room.	Pressure. Left.	Right.
690	70	1.617	1.384
685	69	1.616	1.386
680	—	1.615	1.384
670	70		
660		1.616	1.385
650		1.385	
672		.231 Difference.	
Corrected,			
779			

779 — 70 = 709°.

3d Run: Time, 12.09½–12.14½. Anemometer, 940195
930200

9995

Cubic feet of air per minute =147.95.

Temperature. Stack.	Room.	Pressure. Left.	Right.
625	69	1.613	1.390
615	69	1.611	1.390
610	—		
600	69	1.612	1.390
595		1.390	
609		.222 Difference.	
Corrected,			
708			

708 — 69 = 639°.

4th Run : Time, 12.18–12.23. Anemometer, 930200
920575

.9625

Cubic feet of air per minute =142.65.

Temperature. Stack.	Room.	Pressure. Left.	Right.
560	70	1.610	1.400
550	70	1.612	1.398
540	—		
535	70	1.611	1.399
530		1.399	
543		.212 Difference.	
Corrected, 633			

633 − 70 = 563°.

5th Run: Time, 12.32–12.37. Anemometer, 920500
911390

8110

Cubic feet of air per minute, 135.29.

Temperature. Stack.	Room.	Pressure. Left.	Right.
465	70	1.599	1.414
460	70	1.595	1 419
455	—		
450	70	1.597	1.417
445		1.417	
445		.180 Difference.	
Corrected, 533			

533 − 70 = 460°.

6th Run : Time, 12.56–1.01. Anemometer, 911390
901525

9865

Cubic feet of air per minute, 122.56.

Temperature. Stack.	Room.	Pressure. Left.	Right.
380	71	1.591	1.434
375	71	1.592	1.432
372	—		
370		1.592	1.433
368		1.433	
373		.159 Difference.	
Corrected, 440			

440 − 71 = 370°.

7th Run : Time, 1.10–1.15. Anemometer, 900738
892823

7915

Cubic feet of air per minute, 118.2.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
350	71	1.586	1.440
345	71	1.588	1.442
345	—		
345		1.587	1.441
340		1.441	
345		.146 Difference.	
Corrected, 408			

408 − 71 = 337°.

8th Run : Time, 2.35–2.40. Anemometer, 892825
885990
6835

Cubic feet of air per minute, 102.75.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
255	69	1.585	1.480
255	69	1.590	1.475
252	—	1.590	1.480
252	69		
251		1.588	1.478
		1.478	
253			
Corrected, 304		.110 Difference.	

304 − 69 = 235°.

9th Run : Time, 4.24–4.29. Anemometer, 885990
880339
5651

Cubic feet of air per minute, 85.82.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
205	69	1.595	1.520
205	69	1.595	1.520
205	—		
205	69	1.595	1.520
		1.520	
205			
Corrected, 205		.075 Difference.	

205 − 69 = 136°.

10th Run : Time, 8.56–9.01 P.M. Anemometer, 80340
76187
4153

Cubic feet of air per minute, 64.4.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
135	70	1.610	1.568
		1.608	1.568
		1.609	1.568
		1.568	
		.041 Difference.	

135 − 70 = 65°.

11th Run: Time, 8.31½–8.36½. Anemometer, 76187
76090

97

Cubic feet of air per minute, 18.86.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
80	58	1.637	1.633
80	58	1.640	1.633
80		1.638	1.633
		1.633	
		.005 Difference.	

80 − 58 = 22°.

FIFTH TEST.

April 24, 1890.

1st Run: Time, 1.06½–1.11½. Anemometer, 766684
759720

6964

Cubic feet of air per minute, 104.6.

Temperature of room, 62°.

Temperature of stack determined by Pouillet's pyrometer.

Weight of piece of iron, .066 lb.

Temperature of water after, 68.25
" " before, 60.20

8.05°

$$.11 \times .066(T - 68.25) = 2(68.25 - 60.2)$$
$$T = 2280°$$
$$2280 - 62 = 2220°.$$

Pressure.	
Left.	Right.
1.792	1.516
1.790	1.520
1.798	1.510
1.794	1.517
1.517	

.271 + .011 (correction) = .285 Difference.

2d Run: Time, 1.31–136. Anemometer, 759720
752450

7270

Cubic feet of air per minute, 108.97.

Temperature of room, 62°.

Temperature of stack determined by Pouillet's pyrometer.

Weight of iron ball, .065 lb.

Temperature of water after, 75.4
" " before, 68.0

7.4

$$.065 \times .11 (T - 75.4) = 24.8$$
$$T = 2140^\circ$$
$$2140 - 62 = 2078^\circ.$$

PRESSURE.

Left.	Right.
1.790	1.520
1.792	1.515
1.785	1.524
1.792	1.515
1.789	1.518
1.518	
.269 Difference.	

269 + .011 (correction) = .280.

3d Run: Time, 2.14–2.19. Anemometer, 851775
843090
8685

Cubic feet of air per minute, 129.21.
Temperature of room, 60°.
Weight of iron ball = .068 lbs.
Temperature determined by Pouillet's Pyrometer:
Temperature of water after, 62.2
" " before, 58.4
3.8

$$.11 \times .068 (T - 62.2) = 7.6$$
$$T = 1077^\circ$$
$$1077 - 60 = 1017^\circ.$$

PRESSURE.

Left.	Right.
1.778	1.544
1.775	1.550
1.774	1.550
1.776	1.548
1.548	
.228 Difference.	

.228 + .011 (correction) = .239 Difference.

4th Run: Time, 2.33–2.38. Anemometer, 843090
835025
8065

Cubic feet of air per minute = 120.35.

TEMPERATURE.		PRESSURE.	
Stack.	Room.	Left.	Right.
645	70	1.760	1.560
630	70	1.756	1.564
625	—		
633		1.758	1.562
		1.562	

.196 + .011 (correction) = .207 [Difference.

633 — 70 = 563°.

5th Run: Time, 3.39–3.45. Anemometer, 834910
826770
8140

Cubic feet of air per minute, 101.97.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
395	69	1.740	1.604
395	69	1.744	1.600
390	—		
393		1.742	1.602
		1.602	
		.140 + .011 (correction) = .151 [Difference.	

393 — 69 = 324°.

6th Run: Time, 4.27½–4.32½ Anemometer, 9826770
9820392
6378

Cubic feet of air per minute, 96.22.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
320	68	1.744	1.619
320	—	1.750	1.620
320		1.750	1.620
320		1.748	1.620
		1.620	
		.128 + .011 = .139 Difference.	

320 — 68 = 25.2°.

7th Run: Time, 9.50–9.55. Anemometer, 820400
818732
1668

Cubic feet of air per minute, 28.56.

Temperature.		Pressure.	
Stack. [Ther.]	Room.	Left.	Right.
75	63	1.732	1.734
75		1.730	1.734
75 — 63 = 12°		1.731	1.734
			1.731
			.003

—.003 + .011 = .008 (real pressure).

SIXTH TEST.

April 29, 1890, A.M.

1st Run: Time, 10.40¼–10.45¼. Anemometer, 960750
950420
10330

Cubic feet of air per minute, 152.8.

Temperature. Stack.	Room.	Pressure. Left.	Right.
680	68	1.440	1.206
670	68	1.825	1.216
625	—		
		1.483	1.211
658		1.211	
		.222 Difference.	

658 − 68 = 590°.

2d Run: Time, 10.48–10.58. Anemometer, 950420
940500

9920

Cubic feet of air per minute, 146.88.

Temperature. Stack.	Room.	Pressure. Left.	Right.
590	68	1.439	1.210
570	68	1.420	1.225
555	—	1.442	1.208
540			
		1.433	1.214
551		1.214	
		.219 Difference.	

551−68 = 483°.

3d Run: Time, 10.55-11.00. Anemometer, 940500
931150

9350

Cubic feet of air per minute = 138.72.

Temperature. Stack.	Room.	Pressure. Left.	Right.
520	68	1.410	1.232
500	68	1.413	1.228
490	—	1.410	1.235
480			
		1.411	1.232
498		1.232	
		.179 Difference.	

498 − 68 = 430°.

4th Run: Time, 11.03–11.08. Anemometer, 931150
921990

9160

Cubic feet of air per minute, 136.00.

Temperature. Stack.	Room.	Pressure. Left.	Right.
455	70	1.404	1.240
445	70	1.415	1.231
430	—	1.417	1.225
443		1.412	1.232
		1.232	
		.180 Difference.	

443 − 70 = 373°.

5th Run : Time, 11.12¼–11.17¼. Anemometer, 921990
913290

8700

Cubic feet of air per minute, 129.42.

Temperature. Stack.	Room.	Pressure. Left.	Right.
410	70	1.410	1.235
390	72	1.392	1.255
400	71	1.401	1.245
		1.245	
		.156 Difference.	

400 − 71 = 329°.

6th Run : Time, 11.33–11.38. Anemometer, 913290
905298

8000

Cubic feet of air per minute, 119.41.

Temperature. Stack.	Room.	Pressure. Left.	Right.
340	72	1.385	1.250
330		1.393	1.245
335		1.389	1.247
		1.247	
		.142 Difference.	

335 − 72 = 263°.

7th Run : Time, 11.58–12.04. Anemometer, 905294
897487

6 min. = 7807
5 " = 6505

Cubic feet of air per minute, 98.03.

Temperature. Stack.	Room.	Pressure. Left.	Right.
270	70	1.390	1.250
270		1.388	1.260
		1.395	1.254
		1.391	1.255
		1.255	
		.136 Difference.	

270 − 70 = 200°.

SEVENTH TEST.

April 29, 1890, P.M.

1st Run : Time, 1.58–2.03. Anemometer, 897486
886918

10568

Cubic feet of air per minute, 156.46.

Temperature. Stack.	Room.	Pressure. Left.	Right.
800	74	1.472	1.212
790	—	1.468	1.225
770			
785		1.470	1.218
Corrected, 910		1.218	
		.252 Difference.	

910 − 74 = 836°.

2d Run : Time, 2.08–2.13. Anemometer, 886918
878750

8168

Cubic feet of air per minute, 116.91.

Temperature. Stack.	Room.	Pressure. Left.	Right.
900	76	1.473	1.220
840	76	1.465	1.230
810	—	1.468	1.230
850		1.468	1.227
		1.227	
		.241 Difference.	

850 − 76 = 774°.

3d Run: Time, 2.15–2.20. Anemometer, 878750
868930

9820

Cubic feet of air per minute, 145.44.

Temperature. Stack.	Room.	Pressure. Left.	Right.
730	76	1.468	1.230
720	76	1.463	1.230
710	—	1.465	1.232
690			
712		1.465	1.231
		1.231	
		.234 Difference.	

712 − 76 = 636°.

4th Run: Time, 2.22½–2.27½. Anemometer, 868930
859570

9360

Cubic feet of air per minute, 138.87

Temperature. Stack.	Room.	Pressure. Left.	Right.
660	75	1.456	1.243
640	75	1.465	1.235
635	—	1.465	1.235
630		1.452	1.247
641		1.459	1.240
		1.240	
		.219 Difference.	

641 − 75 = 566°.

5th Run: Time, 2.30–2.35. Anemometer, 859570
849998

9572

Cubic feet of air per minute, 141.9.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
605	75	1.477	1.225
600	75	1.450	1.245
590	—	1.480	1.226
580			
		1.469	1.232
594		1.232	
		.237 Difference.	

594 − 75 = 519°.

6th Run: Time, 2.38–243. Anemometer, 849998
840702

9296

Cubic feet of air per minute, 137.95.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
560	64	1.455	1.250
550	—	1.480	1.254
545			
540		1.452	1.252
		1.252	
552			
		.200 Difference.	

552 − 64 = 488°.

7th Run: Time, 2.50–2.55. Anemometer, 840702
832242

8460

Cubic feet of air per minute, 125.978.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
510	73	1.445	1.260
505	—	1.456	1.254
495		1.444	1.264
503		1.448	1.259
		1.259	
		.189 Difference.	

503 − 73 = 430°.

8th Run: Time, 3.13–3.18. Anemometer, 832242
824506

7736

Cubic feet of air per minute, 115.61.

Temperature.		Pressure.	
Stack	Room.	Left.	Right.
425	73	1.435	1.285
420	—	1.446	1.275
410			
		1.442	1.280
418		1.280	
		.162 Difference.	

418 − 73 = 345°.

9th Run: Time, 3.59–4.04. Anemometer, 824506
817480

7026

Cubic feet of air per minute, 105.457.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
290	75	1.435	1.305
295	—	1.436	1.308
290		1.435	1.311
292		1.435	1.308
		1.308	
		.127 Difference.	

292 − 75 = 217°

10th Run: Time, 5.15–5.20. Anemometer, 817480
811400

6080

Cubic feet of air per minute, 91.84.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
238	76	1.440	1.336
—		1.430	1.342
		1.435	1.339
		1.339	
		.096 Difference.	

238 − 76 = 162°.

11th Run: Time, 7.35–7.40. Anemometer, 811500
805920

5580

Cubic feet of air per minute, 84.794.

Temperature.		Pressure.	
Stack.	Room.	Left.	Right.
172	72	1.455	1.365
—	—	1.365	
		.090	

172 — 72 = 100°.

DISCUSSION.

Prof. J. B. Webb.—In the opening paragraph of this paper we are told that it is "time to ascertain by direct experiment whether the premises on which the mathematical treatment is based are those of practice," and that the writer planned an investigation to settle whether there is a temperature of maximum delivery within the range of temperature of chimneys as attached to the steam-boiler, and that the work was all done by students who contributed many ingenious devices for eliminating obstacles, etc.

We therefore naturally expect the paper of thirty-five pages to contain a complete description of a well-planned apparatus, including the ingenious devices, instead of which, but one paragraph is devoted to it, and we are told simply the size, shape, and material of the stack, and that it had under it a specially constructed removable grate, not otherwise described. No ingenious device is mentioned, unless it be this grate; the removability of which was certainly a highly ingenious device for getting rid of it and the fire, with all the trouble of regulating and measuring the latter.

This grate, constructed "to eliminate all uncertainty as to resistances," seems not to have done so, for we read (third paragraph) that uniform resistance was secured by allowing the "fire to burn completely out, leaving a bed of ash," or by entirely removing the fire and closing the bottom of the stack with a perforated cover.

Not only, then, was there no steam-boiler, but no fire—only a heavy bed of ashes—in the apparatus which was to "give a correct solution of the problem from the stand-point of the engineer" (see first paragraph); not of the mathematician, who is popularly supposed to be guilty of such vagaries, but, of all men, the engineer! The apparatus and its manner of use were, then, about as far removed from practical conditions as possible. As to the unchangeableness of the bed of ashes and the reliability of most of the other conditions influencing the data, there is room for a wide difference of opinion from that of the paper.

As to the data themselves, they are so incomplete as to justify little confidence in their accuracy. We are told (third paragraph) that the work was only begun formally when uncertainty had been eliminated by preliminary tests; and yet, in the first test given, there is so much that it has not been admitted to the graphical table.

No data are given as to the methods of standardizing and using the pyrometers and anemometer, nor as to the opening supplying the air. What the "temperature of the stack" means we cannot even guess, except that it is probably the indication of the pyrometer, which might have been placed in the stack in various ways and at various places, with scarcely any chance of getting the temperature of the hot air as distinguished from that of the walls of the stack. Moreover, as the wall of the stack was the main reservoir of heat, the air could not have maintained a constant temperature during its half-minute's passage through the stack. We notice, also, that the most of the space (twenty pages) over which these tests are spread is devoted to repeated readings of the two sides of the "very finely divided pressure-gauge" (second paragraph), and of the temperature of the room.

In the face of all this, as to the apparatus and data, we can hardly agree with the highly favorable view taken of these tests in the fourth paragraph, that, "Their study will probably settle any question which may arise relating to the subject thus investigated"

After devoting only five pages to the apparatus and tests, some random shots at theory are attempted.

The ninth paragraph (page 90) commences by stating that the usual formula is "*obviously* inaccurate in form," containing "an entirely false set of assumptions" (which, however, are mainly retained by the author). It then goes on to mention certain *conditions* under which this may seem to be the case, followed by others having a contrary effect, and concludes by saying: "In such cases the maximum will evidently be found, at least approximately, where Rankine places it."

This introduction is a type of what follows; things are "obvious" and "evident" until the attempt is made to explain them, when they change color or contradiction ensues.

Thus the eleventh paragraph contains a rare piece of logic. The first premise stated is, "Since the temperature of the chimney in any boiler plant is a direct function of the coal burned on the grate"; and the second is, "Since the quantity so burned depends, in turn, on the draught"; while a third premise introduces a syndicate of "maximums," and begs the question. Now, from the first and second, we might conclude that an increased supply of air necessitates a higher chimney temperature, but on page 97 we read that, "In all cases of good practice the air entering the fire is raised to the same maximum temperature, whatever its volume or weight"; and that, "Its variation is dependent, not upon the chimney temperature, but on the proportion of air admitted to dilute the products of combustion"; and, further, that, "*The movement of air through the fuel-bed and grate* *is effected at temperatures absolutely independent of the chimney temperatures,*" statements which seem to upset the above premises—with the rest of the paragraph, and to suggest, at least, that an increased supply of air might, by diluting the products of combustion, lower the chimney temperature.

The conclusion of this logical paragraph is, "The discovery of any one case in which the boiler chimney is delivering gas at a higher temperature (than 600°) is thus sufficient to prove the fallacy of the proposition," that a maximum temperature can exist; which conclusion may be answered by saying that the discovery of any one case in which such a maximum appears is sufficient to prove the fallacy of any attempt to show that it cannot exist, and that such cases are acknowledged in numerous places in the paper.*

We are not now taking ground either for or against a maximum temperature, but mildly criticising the paper, in fulfilment of the duty to maintain the standard of our *Transactions*—a duty which should be felt by every member.

We are not able, by looking at a palm, to predict a fortune, nor, by noticing hot gases pouring out of a chimney, to predict the non-possibility, or otherwise, of a maximum temperature, and we have no desire to discount the statement that (par. 11) "in hundreds of cases—thousands probably," exceedingly hot gases from certain chimneys have been noticed by persons of sufficient intelligence to know that they were red-hot or even flaming. But when a paper acknowledges cases in which a maximum may exist, and reports experiments showing one under other conditions, any one with this much intelligence may be pardoned for failing to see the evident obviousness of its arguments to the contrary.

A more lengthy review of the paper would not be profitable, and we leave it, with the mere suggestion of an apparatus, planned some years since for such experiments, but which may at some future time be presented to the Society in detail:

A brick chimney of rectangular section, say 30 to 50 feet high, with a strip of iron hung in it, 30 to 50 feet long, and as wide as the rectangle, by means of which the sectional area could be varied.

---

* The statement (page 96, third line from the top) that the ordinary analysis assumes the chimney friction to be the controlling quantity is not true, the great resistance of the grate being well understood and allowed for.

An upright tubular boiler, connected by an oil-seal with the chimney and hung upon a scale so that its weight can be automatically registered, with means also of measuring the water furnished and evaporated; by means of this boiler the change of temperature of the gases while passing through the flues could be varied at will.

A separate grate and ash pit, connected with the boiler by an oil-seal and supported on scales so that its weight could be automatically registered; this would enable the weight of fuel consumed per minute, and also the weights fired at intervals, to be known.

A blower, connected with the ash pit by an oil-seal, so that any pressure could be maintained therein; this would compensate for the lack of variability in the height of the chimney and enable the results to be reduced to any chimney height.

With such an apparatus experiments could be made, we believe, under practical conditions.

Prof. Jas. E. Denton.—A point which has not been mentioned in the discussion is the citation, by the author of the paper, of Isherwood's Report, entitled *Experiments with a Martin Water-tube Boiler*, for the purpose of proving that there was an increase of draught caused by an increase in the chimney temperature above the 600° which the theorem in Rankine's *Treatise on the Steam-engine* declares to be productive of about the maximum draught.

I think this reference is unfortunate, and might be even misleading. First, because the higher temperatures were not accurately determined, and of the two experiments at 700° one shows a reduction of draught as compared with that of 600°, indicating that duplicate experiments might show much less apparent gain in the draught at the highest temperature. Second, that inasmuch as the rows of 3-inch tubes, through which the chimney gases had to flow, were reduced from 22 to 13 in number when the temperature in the chimney was increased from 600° to the highest limit, the reduction of resistance to draught thus caused may have been responsible for the increase of draught which really occurred.

Prof. Horace B. Gale.—The author of these experiments renders a valuable service in recalling the discussion of chimney draught to the solid ground of facts. Those placed before us in this paper are both interesting, from a scientific stand-point, and of practical value.

The conclusions drawn from them, although not new, are now placed upon a firmer foundation than before. The same conclusions were announced as a result of independent investigations in a paper on *Theory and Design of Chimneys*,* presented by the writer at the November meeting of last year. The error in the generally accepted theory of Peclet and Rankine, to which reference is here made, was also pointed out in that paper, and a new theory was constructed on the same principles as are outlined in Prof. Thurston's analysis.

The interest awakened by the discussion at that time brought out a number of papers in defence of the old formulæ, one of which, by Prof. Wood,† was directed especially to the points covered by the present experiments. In the writer's discussion of that paper is made the statement, which is here derived independently by Prof. Thurston, that the value of G in Rankine's equation is variable, and proportional inversely to t_1^2.

The results given in the writer's paper of a year ago agree so thoroughly with those of the present experiments that each set serves as a confirmation of the other. The conditions of the experiments and the methods of investigation in the two cases were, however, entirely different. A comparison of the methods and results as set forth in the two papers will, I think, be interesting, and will throw some light upon recent discussions.

The method used in these experiments is simple and direct; but the conditions differ from the usual conditions of steam practice, in that the boiler, and the resistance offered by its contracted passages to the gases of combustion, are here absent.

The writer's experiments, on the other hand, were not directed especially to the question of maximum draught, but were made primarily to determine a series of constants which could be used in formulæ for the design of chimneys. All the measurements were made upon boiler furnaces, under the ordinary conditions of practice—the resulting formulæ being as yet the only ones in which all the coefficients necessary in a rational theory have been thus determined.

Let us first compare the *pressures* given by these formulæ with those of the experimental curves. Equation 2 of the recent paper‡ gives, as "the excess of the external over the internal

* *Transactions*, Vol. XI., p. 451.

† *Chimney Draught, Transactions*, Vol. XI., p. 974.

‡ Vol. XI., p. 455.

pressure at the base of the chimney in pounds on the square foot,"

$$P = H\left(\frac{39}{T_a} - \frac{40}{T_s}\right) - P_f.$$

H is the height of the chimney, or, in this case, 32 feet.

T_a is the absolute temperature of the air, which, in these experiments, was approximately 70 + 461, or 531° Fahr.

T_s is the mean absolute temperature of the gases in the stack.

P_f represents the pressure required to overcome the chimney friction.

The value of this last term, when worked out by Equation 4 of the paper,* is found to be so small as to be practically inappreciable for any rates of delivery observed in these tests. We may therefore neglect it, and put

$$P = H\left(\frac{39}{T_a} - \frac{40}{T_s}\right).$$

We may now apply this formula to compute the pressures for the points on the curves corresponding respectively to temperature differences of 200°, 400°, and 600°. The observed temperatures of the chimney which would give these differences would be 270°, 470°, and 670°. These temperatures, being taken near the base of the stack, would, however, require to be corrected for the cooling of the gases in passing from the bottom to the top, in order to obtain the true mean temperatures.

The writer's experiments give the value of this correction, for a round brick stack, as

$$.0003\,\frac{H\sqrt{A}(T_s - T_a)}{W}.$$

(See Equation 26, *Theory and Design of Chimneys*, Vol. XI.)

A is the area of the flue, which is here 2.18 square feet, and W is the weight of gas delivered per second. The value of W may be found by multiplying the mean delivery in cubic feet per minute, as shown by the curves for each of the given temperature differences, by .074 (or the weight of a cubic foot at 70°), and dividing by 60. The resulting values make the corrections for cooling of the gases in the stack for the three cases chosen, 22°, 33°, and 44°, respectively.

Subtracting these corrections from the corresponding observed

* Vol. XI., p. 455.

temperatures at the base of the stack, and adding 461 to each remainder, we obtain, as the probable *mean absolute temperatures* in the stack, 709°, 898°, and 1,087°.

Substituting these figures for T_s in the equation, and dividing the resulting values of P by 5.2 (to reduce to inches of water), we obtain calculated pressures as follows, which are here compared with the average values shown by the curves.

Temperature Difference.	Calculated Pressure.	Average of Observed Pressures.
200°	.105 inch.	.117 inch.
400°	.178 "	.175 "
600°	.225 "	.216 "

The calculated values agree with the observed values within the limits of variation of the different experiments.

Turning our attention now to the delivery curves, it appears that the maximum was not reached at the same temperature even when the conditions of the experiments were intended to be similar. In this regard, it may be observed that the determination of a maximum point in a curve of this nature is a matter of considerable delicacy, the slightest change in the form of the curve being sufficient to shift the maximum to a very noticeable extent. The accidental variations in the resistance of the fuel-bed, which Prof. Thurston mentions as a cause of these differences in the results, would probably be quite enough to account for them. Such irregularities might be eliminated by taking the average of a sufficient number of tests in which the conditions were kept as nearly alike as possible.

Leaving out Experiments 2 and 6, in which the fires were dumped, we have four curves representing the delivery of gas under approximately similar conditions. Three of these show fairly well-defined maximum points—No. 3 at 725°, No. 4 at 775°, and No. 5 at 1,150°, above the temperature of the air. Curve No. 7, if continued, would apparently also reach a maximum not far from the same point as No. 5.

The average for these four curves is 950°, corresponding to an observed temperature in the stack of 950 + 70, or 1,020°. The correction for cooling, as computed by the formula previously given, would reduce this figure to 948°, which result may be taken as representing our best knowledge of the probable mean temperature in the chimney required for maximum draught under conditions corresponding to the average of these tests.

In the foregoing it has been assumed that the temperature measurements were made at the base of the stack; if the thermometer was inserted above that point, the corrections for cooling should be somewhat less, and the mean temperature correspondingly higher. It is to be hoped that Prof. Thurston will add to his paper a few data of this kind which do not now appear in it—such, for instance, as the dimensions of the grate, the kind of fuel burned, the exact points where the measurements of pressure and temperature were made, whether the temperature of the room, as given in the reports, represented also the temperature of the column of air outside the chimney, etc. These data would be valuable in the comparison of these results with those of other experiments.

In the recent paper by the writer there was deduced a general expression for the temperature of maximum draught for any chimney and furnace of given dimensions. Now, if the theory there employed is the correct one, and if the experimental values of the various coefficients also were correctly determined, that formula, when applied to the conditions of these experiments, should make the temperature of maximum draught about 948° Fahr. The equation, which is No. 38 of the paper on *Theory and Design of Chimneys*, is written as follows:*

$$T'_{e} = \frac{40}{39} T_a + \frac{40}{39} T_a \sqrt{1 + 0.9 \frac{KA^3 T_a}{CMHa^2}}.$$

The symbols H, T_a, and A have the values already given.

C is the coefficient of friction of the gas on the sides of the chimney, whose value for a brick-lined stack, as determined by the writer's measurements, is given as .016.

M is the inside circumference of the stack, which in this case is 5.24 feet.

a is stated in the paper to be "the area of the openings between the grate bars, plus the area of any orifices for the admission of air above the fire." These data are not given in the report. Assuming the area a as approximately one-half that of the grate, and a ratio of grate to chimney area of 8 to 1, would

* Vol. XI., p. 471.

make a equal to 8.72 square feet. If these assumptions differ materially from the facts, Prof. Thurston will, I trust, correct them.

The only remaining factor to be evaluated is K, which is a coefficient of resistance for the furnace. In an ordinary boiler-furnace this quantity is the sum of the separate resistances of the grate and fuel-bed, the boiler-tubes, the smoke-flue, etc. In Equation 6 of the paper referred to, values based on the writer's measurements are assigned to these component parts of the resistance, the coefficient for the grate and fuel-bed * being given as .07. Now, in these experiments the resistance of the grate and fuel-bed was practically the whole resistance of the furnace. We may therefore put $K = .07$.

Substituting these values in the equation, and subtracting 461° from the resulting value of $T'_{\bullet}$, to reduce from the absolute to the common scale, we obtain, as the calculated temperature of maximum draught for the conditions of these experiments, 978°.

Thus these tests serve to confirm, to a certain extent, the theory of draught proposed at the meeting of last November, as well as the values determined by the writer for the experimental coefficients.

The general results of the investigation, so far as relates to the subject of temperature of maximum draught, were stated in the recent paper in the following words: "It is evident that the temperature of maximum draught is not invariable, but depends upon the dimensions of the chimney and furnace. . . . It is also evident that this temperature can never be as low as that given by Rankine's formula. . . . The ordinary values of the temperature of maximum draught would range from 1,000° to 2,000° on the common Fahr. scale." †

The author of the present paper concludes, from his investigation, that "the actual maxima fall more commonly at points on the scale between 800° and 2,000° Fahr." It is possible that, in reaching this conclusion, Prof. Thurston has somewhat underestimated the frictional resistance due to the boiler, which, if present, would increase the value of K, and thus raise the temperature of maximum delivery. The writer has in numerous cases found this resistance, as measured by the method of pressure differences, equal to, and in some instances even larger than, that of the grate and fuel-bed. A test of this kind was reported

* Vol. XI., p. 459. † Ibid., p. 471.

in detail at the last fall meeting. As a result of a considerable number of such measurements upon externally fired boilers of various kinds, it would seem that the resistance of the fire-bed alone is seldom more than two-thirds of the whole furnace resistance.

It follows that, if an ordinary boiler furnace had been used for these experiments instead of a special furnace without a boiler, the resistance offered to the draught would have been considerably greater, and would also have been steadier; for the friction in the boiler-tubes would be unaffected by variations in the fuel-bed.

The difficulty in the way of an experiment of this kind would be, of course, that the gases would be cooled so much by the boiler that the temperature of maximum delivery could not be reached. It might be approached, perhaps, if the ratio of tube surface to grate were made rather small. In that case the fire could be kept as hot as possible, and the chimney gases cooled gradually by pumping cold water through the boiler, allowing it to escape at the blow-off. An experiment of this kind would be comparatively easy for any one having a boiler and chimney adapted to the purpose, and might bring out some further facts of interest. One of the chief sources of error to be guarded against in such an experiment would be the leakage of air through the boiler-setting and walls of the chimney.

It would seem, however, that the work already done should settle the question, as far as the old formulæ for draught are concerned; being supported neither by tenable theories nor observed facts, they may safely be discarded for something better.

*Prof. R. H. Thurston.**—The two gentlemen who discuss this paper have given so admirable an illustration of opposite deductions, as of opposite attitudes and methods of treatment, that I am not sure that I may not with propriety leave the subject to them, and let the one balance the other—in so far as a destructive can balance a constructive criticism. But little need be added, I am sure. It would be fair to conclude, from the first of the two, that the speaker is competent to do a much better piece of work than this; and we may consider him pledged to do so by his closing statements. When that apparatus, which we are assured will illustrate the ingenuity and the perspicacity of its inventor, is finally set at work in the manner indicated by him, we shall, I

* Author's Closure, under the Rules.

am sure, be given vastly more complete and more valuable results. I sincerely hope that we may soon be permitted to admire and profit by them. At the moment, I can only regret that we could not have obtained some facts and some general information on the subject from him now.

I will, however, say, that it was not supposed, in preparing the paper, that it was necessary to load it with descriptions of details unessential to its immediate purpose; that the plan adopted was for the purpose of securing just the data obtained, evading the uncertainties which would enter with the introduction of unessential conditions; and that the standardizing of the apparatus was considered so simple and so familiar a matter to the average engineer interesting himself in such work, that it would have been a waste of space and time to go into it. I imagine that it is quite enough to say that it *was* standardized, and that the readings represent just what they purport to represent. The "temperature of the stack" means exactly what those terms imply. The pyrometers were placed in the middle of the body of the stack, and are to be taken as giving the temperature of its gases, and no reason is known for assuming any other.

The remark about uncertainty in this connection, if I may be pardoned for so frankly saying it, seems to me very absurd, as I think it must to any one familiar with such work. Nothing in the course of the experiments indicated that there was any reason to suppose that the temperature of the gases varied perceptibly in the "half-minute's passage through the stack." As to the result, finally, I am inclined still to cling, with due modesty, to my expressed opinion that they settle the question at issue —*i.e.*, whether a maximum always actually occurs at 600° Fahr. I further am confident that, were these results no more than approximate, as my critic seems inclined to assert, they would still unquestionably be amply sufficient to settle that matter.

The serious aspect of the "mare's nest" of logic, and the apparent contradictions finally thought to be discovered, will disappear when the paper is read understandingly. It will then be seen that it is perfectly true that the temperature of the chimney is "a direct function of the quantity of coal burned"— *i.e.*, that it increases, other things equal, when the rate of combustion increases, a matter of universal experience; that it is equally true that the quantity of coal burned depends on the draught, a still more familiar fact; and yet, that it is absolutely true that, as asserted in the body of the paper, equally complete combustion being assumed, *the movement of air through the grate and fuel-bed is effected at temperatures absolutely independent of the chimney temperatures; i.e.*—as is, I think, readily seen by a careful reader of the paper—at temperatures which are those of complete combustion, and constant for the same fuel and air-supply per unit of its weight, whatever the temperature of the furnace, the flues, or the chimney—all of which latter are dependent upon entirely different and independent conditions.

The remark, "The conclusion of all this logical paragraph is 'the discovery of any one case in which the boiler chimney is delivering gas at a higher temperature' (than 600°) 'is thus sufficient to prove the fallacy of the proposition,' that a maximum temperature can exist"; and the added proposition: "Which conclusion may be answered by saying that the discovery of any one case in which such a maximum appears is sufficient to prove the fallacy of any attempt to show that it cannot exist, and that such cases are acknowledged in numerous places in the paper," may, in my opinion, be taken as beautiful illustrations of that defective logic which is, to my mind at least, mistakenly assumed to exist in the "mare's nest."

A more careful reading of the paper will show that the assertion is nowhere made that it is impossible "that a maximum temperature can exist"; that the assertion which *is* made is, that such maximum does not exist in practice at the temperature unqualifiedly claimed for it by some and by the mathematics which they would sustain. My assertion is, in fact, that such a maximum would exist were it true, as assumed in that delusive reasoning, that the resistances at the grate and in the chimney were in constant proportion; that actually it rises as the conditions approximate those of usual practice; that the paper shows this fact by giving results obtained under conditions sufficiently approximating usual conditions, and conditions vastly more nearly in accordance with usual practice than the assumptions on which the critic bases his algebra and his "logic." The matter of his criticism thus falls to the ground; its foundation is pure imagination.

The suggestion which concludes these curious trains of destructive ratiocination, of an apparatus which shall give us exact data to contrast with these now given, includes some "ingenious devices," and gives some reason to hope that the obvious practical difficulties may all be overcome, as well as all which may appear when the attempt is made to do the work; and it is to be hoped that it may not prove to be merely "the airy fabric of a dream," but that we may, very soon, secure by its means a settlement of the question whether the disputed equation is correct in form and basis, and whether it is true that a maximum exists in all cases, as asserted, at 600° Fahr.

Prof. Gale's remarks impress me as amply sufficient, if anything of that kind is really required, to relieve any apprehensions which might arise in the mind of any candid reader looking for the truth in regard to a disputed point in physics and mathematics. I think it cannot be necessary for me to rewrite the text of the paper. I will content myself with expressing the hope that those who may have sufficient interest in the matter to read the paper, when published, will at least read it more carefully and to better purpose than has the first speaker, and with the further suggestions that the difficulties of exact investigation may prove more numerous and greater than can be realized until it is actually undertaken; and that, if those promised investigations fail to supply all that is expected of them, we shall still be very glad to get all the facts which they may reveal, nevertheless, even though they prove that a maximum temperature of effective draught *does* always exist at 600° Fahr., that the rates of flow of air through fuel and chimney are exactly proportional, and that G, in the disputed equation, is a constant.

CCCCXVIII.*

SOME PROPERTIES OF AMMONIA.

(*Second Paper.*)

BY DE VOLSON WOOD, HOBOKEN, N. J.

(Member of the Society.)

In my article in Vol. X. of the *Transactions*, on "Some Properties of Ammonia," I gave a formula, equation (23), for the specific heat of a liquid depending upon the properties of the saturated vapor of the substance. One algebraic sign in that equation should be changed from + to −, so I will here write it correctly:

$$c = \frac{v}{J}\frac{dp}{d\tau} - \frac{dh_e}{d\tau} - k_p\left(\frac{d\tau'}{d\tau} - 1\right); \quad . \quad . \quad . \quad . \quad . \quad (1)$$

in which

J is the mechanical equivalent of heat, and in English units is 778 foot-pounds.

p, the pressure in pounds per square foot.

v, the volume in cubic feet of a pound of the vapor.

τ, the absolute temperature at which the liquid is evaporated.

τ', the absolute temperature of the superheated vapor.

T, the temperature on the Fahrenheit scale.

h_e, the latent heat of evaporation per pound in ordinary thermal units; and

k_p, the ordinary specific heat of the vapor, which for ammonia is 0.50836.

In applying this formula, I assumed that $\frac{d\tau'}{d\tau}$ was unity, but I find that it has a finite value. I will recompute the value of c, and will bring forward all the necessary formulas.

From the formula

$$\log p = 8.4079 - \frac{2196}{\tau}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

we find

$$\frac{dp}{d\tau} = \frac{2.3026 \times 2196\,p}{\tau^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

* Presented at the Richmond Meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

From the equation

$$p = 91\frac{\tau_2}{v_2}\left(\frac{\tau}{\tau_2}\right)^{4.3641} - \frac{16920}{v^{1.97}}\left(\frac{\tau}{\tau_2}\right)^{6.6274}, \quad . \quad . \quad . \quad . \quad (4)$$

which is the equation of an adiabatic of the superheated vapor, find $d\tau$ and call it $d\tau'$, after which drop all the subscripts in the right-hand member, since the initial state must coincide with the general one, and thus find, by the aid of equation (3) above,

$$\frac{d\tau'}{d\tau} = \frac{2.3026 \times 2196}{397.13 - \frac{112135}{\tau v^{0.97}}} \cdot \frac{pv}{\tau^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

It was also found that

$$-\frac{dh_e}{d\tau} = 0.6130 + 0.000438T. \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

These reduce equation (1) to

$$c = 1.12136 + 0.000438T + \frac{pv}{\tau^2}\left[6.49922 - \frac{2570}{397.13 - \frac{112135}{\tau v^{0.97}}}\right]. (7)$$

I find that this formula gives a decreasing value of the specific heat for increase of temperature, a condition that has been proved experimentally only for water from 40° Fahr. to about 80° Fahr.; and the decrease was so small as to escape the observation of Regnault, who observed it for states differing considerably in temperature.

Equation (7) gives a decrease of the specific heat of about 0.0014 for each degree of increase of temperature, which is some forty times the positive rate of change of the specific heat of water. The fact that it gives negative results indefinitely and at so rapid a rate excites suspicion that the theory is defective. The cause appears to be in the denominator of the last term of the parenthesis of equation (7), for in the determination of the equation to the adiabatic the exponents and coefficients are considered constant, which they cannot be exactly, and the equation of the fluid, $pv = a\tau - \frac{b}{v^n}$, is only approximate. The equation should be used, if at all, only within the limits of Regnault's experiments—

that is, between 11 and 24 cubic feet. Using the following set of values, determined in my paper in Vol. X.—viz.: $v = 20.7985$, $\tau = 426.66$, $T = -34°$ Fahr., $p = 1823.7$—equation (7) gives

$$c = 1.093. \qquad (8)$$

We next try the effect of assuming that the adiabatic of the superheated vapor is that of a perfect gas. For this case we have

$$pv = R\tau = 89.343\tau.$$

$$\frac{\tau}{\tau_2} = \left(\frac{v_2}{v}\right)^{\gamma - 1}. \qquad (9)$$

$$p = R\frac{\tau_2}{v_2}\left(\frac{\tau}{\tau_2}\right)^{\frac{\gamma}{\gamma - 1}}. \qquad (10)$$

$$\frac{dp}{d\tau} = \frac{\gamma}{\gamma - 1}\cdot\frac{R}{v_2}\left(\frac{\tau}{\tau_2}\right)^{\frac{1}{\gamma - 1}}. \qquad (11)$$

Dropping the subscripts $_2$, and accenting $d\tau$, we have

$$\frac{dp}{d\tau'} = \frac{395.31}{v}. \qquad (12)$$

This, with the preceding equations, reduces equation (1) to

$$c = 1.12136 + 0.000438T - 0.00202\frac{pv}{\tau^2}. \qquad (13)$$

For the state

$$v = 20.7985,\ p = 1823.7,\ \tau = 426.66,\ T = -34°\ \text{Fahr.},$$

this becomes

$$c = 1.10647 - 0.00042 = 1.10605. \qquad (14)$$

For the state $T = 80°$ Fahr.,

$$v = 1.89,\ p = 22192,\ \tau = 540,$$

we have

$$c = 1.15640 - 0.00029 = 1.15611. \qquad (15)$$

It will be seen that the last term of equation (13) is so small as to affect only the fourth decimal figure, and hence may be omitted, in which case we have

$$c = 1.12136 + 0.0004387T. \quad . \quad . \quad . \quad . \quad . \quad (16)$$

We are no longer confined to mere theoretical values for the specific heat of liquid ammonia, for Dr. Hans Von Strombeck, consulting chemist for the De La Vergne Refrigerating Company, of New York, in the summer of 1890, found from the mean of eight experiments that the specific heat is

$$c = 1.22876, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

the temperature of the liquid being about 80° Fahr. The specific heat of liquids is, in practice, treated as constant. The value in equation (15) is nearly 6% less than the value found by experiment.

Dr. Von Strombeck also found the latent heat of vaporization of ammonia at 30°.45 Fahr., the mean of six experiments giving

$$h_e = 534.2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

The corresponding value in my Table of the Properties of Saturated Vapor of Ammonia, Vol. X., is

$$h_e = 546.4, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

which is only 2.2 units more than the value found by Dr. Von Strombeck, or $\frac{4}{10}$ of one per cent. more.

A determination made by Prof. Jacobus from Regnault's experiments, in a paper of this issue, assuming that the specific heat of liquid ammonia is unity, at 53° Fahr., gives

$$h_e = 521.6. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

My table in Vol. X. gives

$$h_e = 522.4 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

a value which agrees still nearer with the experimental one. The close agreement, however, of a particular value is not so important as a fair agreement of values at different temperatures, as in this case. These experiments show that my table is sufficiently exact for engineering purposes, if not for all others.

CCCCXIX.*

MECHANICAL AND PHYSICAL PROPERTIES OF SULPHUR DIOXIDE (SO_2).

BY DE VOLSON WOOD, HOBOKEN, N. J.
(Member of the Society.)

REGNAULT determined the relation between the pressure and temperature of the saturated vapor of sulphur dioxide (SO_2), and the relation between the pressure and volume of the superheated vapor (or vaporous gas), at 1.7° C.=35.06° Fahr. The experimental value of the latent heat of vaporization is not known, nor the specific heat of the liquid. Last year I deduced some formulas for determining the values of both these constants and applied them to determination of certain properties of ammonia.

I desired to have my formulas and methods applied to the determination of certain properties of sulphur dioxide (SO_2), and I was fortunate in enlisting the services of Messrs. A. C. Atristain, Arthur H. Hall, and W. F. Lawrence, members of the last graduating class at the Stevens Institute, to make the numerical computations, which they have faithfully done under my directions, their only consideration being the privilege of making the subject their graduation thesis. I find that the latent heats of vaporization from −20° Fahr. to 40° Fahr. are about 1⅓% less than those found by Ledoux; a much closer agreement than would be anticipated, considering that different formulas were used. He used Roche's formula for the relation between pressures and volumes, while we used Rankine's; he used Zeuner's formula for the relation between the pressure, volume, and temperature of the superheated vapor, while we used a part of Rankine's. The close agreement of the results within the range of temperatures ordinarily used in practice will inspire confidence in them, although they probably will never take the place of experimental values.

*Presented at the Richmond Meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

Special importance attaches to the properties of this substance because it is used in refrigerating machines. It remains in a state of vapor at comparatively low temperatures and does not congeal under ordinary pressures.

The data for the general expression for volumes of the *liquid* per pound at different temperatures were taken from a series of experiments by D'Andreèff.

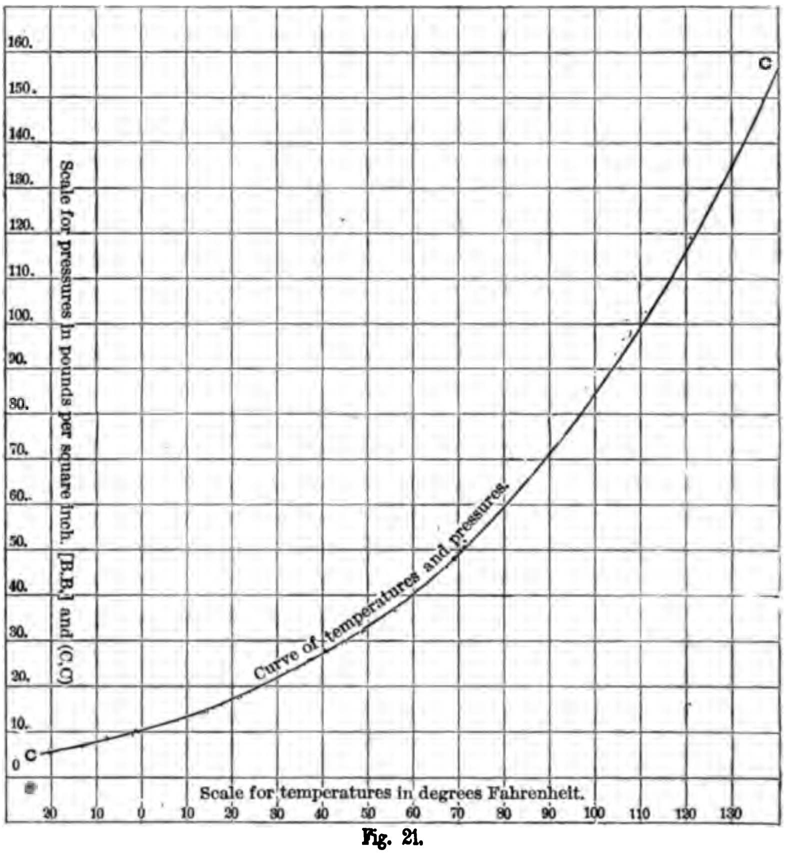


Fig. 21.

In *Relation des Expériences* (Vol. II., pages 583 to 585) are the results of three series of experiments by Regnault showing the pressures for various temperatures from $-22°$ to $144°$ Fahr. These have been reduced to English units and given in Table I., and plotted in Fig. 21, abscissas being temperatures, and ordinates pressures.

TABLE I.

RELATION BETWEEN PRESSURE AND TEMPERATURE OF SATURATED SO_2 AS OBSERVED BY REGNAULT.

Temp. deg. C.	Press. per. mill. of mercury.	Temp. deg. Fahr.	Press. per sq. inch. Pounds.
−30.24	318.69	−22.432	6.16
−30.11	318.99	−22.198	6.17
−30.11	318.05	−22.198	6.15
−29.45	324.80	−21.010	6.28
−27.63	325.56	−17.734	6.30
−27.50	328.28	−17.500	6.35
−27.33	331.08	−17.194	6.41
−27.21	332.82	−16.978	6.44
−27.03	333.90	−16.654	6.46
−26.86	337.28	−16.348	6.52
−26.22	348.85	−15.196	6.74
−24.88	394.40	−12.784	7.63
−22.02	444.89	− 7.636	8.61
−18.52	515.87	− 1.336	9.98
−18.74	501.14	− 1.732	9.69
−18.25	520.49	− 0.850	10.07
−13.48	648.61	+ 7.736	12.55
−13.37	647.12	+ 7.934	12.52
−10.55	739.96	13.010	14.31
−10.48	745.50	13.136	14.42
−10.34	745.75	13.388	14.42
−10.31	745.92	13.442	14.43
− 8.97	792.46	15.854	15.33
− 7.61	842.30	18.302	16.29
−13.27	654.15	8.114	12.65
− 5.76	912.17	21.632	17.64
− 5.74	912.33	21.668	17.65
− 4.16	979.42	24.512	18.94
± 0.	1170.47	32.	22.64
± 0.	1164.58	32.	22.53
+ 1.35	1228.81	34.43	23.57
+ 1.43	1231.31	34.574	23.82
+ 2.48	1286.57	36.464	24.89
3.56	1338.74	38.408	25.89
4.56	1393.87	40.208	26.96
6.55	1485.30	43.97	28.73
9.54	1698.23	49.172	32.85
9.64	1707.18	49.352	33.02
13.36	1953.17	56.048	37.78
19.70	2424.50	67.46	46.90
20.78	2532.01	69.404	48.98
21.31	2593.10	70.358	50.16
21.35	2594.82	70.43	51.19
29.95	3403.25	85.91	65.81
29.94	3403.25	85.892	65.81
30.12	3497.02	86.216	67.64
30.22	3495.17	86.396	67.60
37.15	4344.89	98.87	84.04
37.59	4326.89	99.662	83.89
39.18	4752.27	102.524	91.22
45.26	5427.30	113.468	104.91
48.66	6195.91	119.588	119.84
49.46	6112.88	121.028	118.21
62.52	8662.18	144.536	167.54

Assuming that the law of pressures and temperatures can be represented by Rankine's formula:

$$\log p = A - \frac{B}{\tau} - \frac{C}{\tau^2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

three points were taken on the above curve from which the values of A, B, and C were determined as follows:

When $p = 7$ lbs. per square inch, $t = -14°$ $F. = 446.66°$ absolute.
When $p = 50.5$ lbs. per square inch, $t = 71° F. = 531.66°$ absolute.
When $p = 144$ lbs. per square inch, $t = 134°.5 = 595.16°$ absolute.

From which A, B, and C were found to equal:

$$A = 5.2330. \quad B = 1439.0. \quad C = 232659.$$

The general formula for this vapor now becomes:

$$\text{com} \log p = 5.2330 - \frac{1439.0}{\tau} - \frac{232659}{\tau^2} \quad . \quad . \quad (2)$$

for pressures in pounds per square inch; or,

$$\text{com} \log p = 7.39136 - \frac{1439}{\tau} - \frac{232659}{\tau^2} \quad . \quad . \quad . \quad (3)$$

for pressures per square foot. This curve is plotted in Fig. 21.

By substituting values for τ in equation (3), Table II. was computed, in which the pressures per square foot are given for every five degrees of temperature from $-20°$ to $+140°$ Fahr., as also the pressures per square inch.

From thermodynamics we have, for the latent heat of vaporization:

$$H_e = \tau v \frac{dp}{d\tau} \text{ in foot pounds}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

or,

$$h_e = \tau v \frac{dp}{d\tau} \div 778 \text{ in thermal units.} \quad . \quad . \quad . \quad (5)$$

Also, by differentiating equation (3), we get:

$$\frac{dp}{p} = \left[0 + 1439 \frac{dt}{\tau^2} + 2 \times 232659 \frac{dt}{\tau^3} \right] \times 2.3026.$$

$$\therefore h_e = \tau v \frac{dp}{d\tau} = \left\{ 1439 \frac{pv}{\tau} + 2 \times 232659 \frac{pv}{\tau^2} \right\} \left[\frac{2.3026}{778} \right]. \quad (6)$$

It remains to find $\frac{pv}{\tau}$.

TABLE II.

RELATION BETWEEN PRESSURE AND TEMPERATURE OF SATURATED SO_2 FROM RANKINE'S EQUATION.

Temp. deg. Fahr.	Press. per sq. in.	Press. per sq. ft.	Temp. deg. Fahr.	Press. per sq. in.	Press. per sq. ft.
−20	5.878	846.53	+65	45.029	6484.18
−15	6.604	950.99	70	49.554	7135.78
−10	7.868	1133.10	75	54.418	7836.20
− 5	9.002	1296.40	80	59.612	8584.08
± 0	10.300	1483.22	85	65.226	9392.58
+ 5	11.741	1690.71	90	71.202	10253.13
+10	13.344	1921.65	95	77.580	11171.65
15	15.115	2176.59	100	84.380	12150.41
20	17.067	2457.70	105	91.659	13191.89
25	19.214	2766.81	110	99.255	14292.80
30	21.568	3105.81	115	107.443	15471.83
35	24.309	3476.63	120	116.076	16714.97
40	26.953	3881.26	125	125.207	18029.83
45	30.764	4430.72	130	135.332	19487.83
50	33.334	4800.13	135	145.032	20884.68
55	36.934	5318.56	140	155.756	22428.938
60	40.828	5879.19			

Regnault found the volume of a gramme of this gas to be 0.34970 litre at atmospheric pressure at the temperature of melting ice, and to reduce to cubic feet, multiply by $\frac{35.3161}{2.2046}$; hence,

$$V_0 = \frac{0.34970 \times 35.3161}{2.2046} = 5.60188 \text{ cubic feet per pound}; \quad . \quad (8)$$

and
$$R = \frac{p_0 v_0}{\tau_0} = \frac{2116.3 \times 5.60188}{492.66} = 24.06378. \quad . \quad . \quad . \quad (9)$$

Regnault's experiments show that pv for sulphur dioxide diminishes with decrease in volume; hence we assume the equation,

$$pv = a\tau - \frac{b}{v^n}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

a form which has been shown to be approximately correct for steam (*Transactions*, Vol. X., p. 675).

To find the constants in this equation, we must take values for p and v from the following table, containing pressures and volumes (relative) in French units. To reduce to English units, a ratio must be established between the *relative* volumes given by Regnault and the actual volumes desired. Since the gas is

Assuming that the law of pressures and temperatures can be represented by Rankine's formula:

$$\log p = A - \frac{B}{\tau} - \frac{C}{\tau^2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

three points were taken on the above curve from which the values of A, B, and C were determined as follows:

When $p = 7$ lbs. per square inch, $t = -14^\circ$ $F. = 446.66^\circ$ absolute.
When $p = 50.5$ lbs. per square inch, $t = 71^\circ F. = 531.66^\circ$ absolute.
When $p = 144$ lbs. per square inch, $t = 134^\circ.5 = 595.16^\circ$ absolute.

From which A, B, and C were found to equal:

$$A = 5.2330. \quad B = 1439.0. \quad C = 232659.$$

The general formula for this vapor now becomes:

$$\text{com log } p = 5.2330 - \frac{1439.0}{\tau} - \frac{232659}{\tau^2} \quad . \quad . \quad (2)$$

for pressures in pounds per square inch; or,

$$\text{com log } p = 7.39136 - \frac{1439}{\tau} - \frac{232659}{\tau^2} \quad . \quad . \quad . \quad (3)$$

for pressures per square foot. This curve is plotted in Fig. 21.

By substituting values for τ in equation (3), Table II. was computed, in which the pressures per square foot are given for every five degrees of temperature from -20° to $+140^\circ$ Fahr., as also the pressures per square inch.

From thermodynamics we have, for the latent heat of vaporization;

$$H_e = \tau v \frac{dp}{d\tau} \text{ in foot pounds; } \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

or,

$$h_e = \tau v \frac{dp}{d\tau} \div 778 \text{ in thermal units.} \quad . \quad . \quad . \quad (5)$$

Also, by differentiating equation (3), we get:

$$\frac{dp}{p} = \left\{ 0 + 1439 \frac{dt}{\tau^2} + 2 \times 232659 \frac{dt}{\tau^3} \right\} \times 2.3026.$$

$$\therefore h_e = \tau v \frac{dp}{d\tau} = \left\{ 1439 \frac{pv}{\tau} + 2 \times 232659 \frac{pv}{\tau^2} \right\} \left\{ \frac{2.3026}{778} \right\}. \quad (6)$$

It remains to find $\frac{pv}{\tau}$.

in pounds per square foot; or, in millimetres of mercury,

$$2128.585 \times \frac{760}{2116.3} = 764.4159.$$

To interpolate in the table for the corresponding volume, a value for R must be found from the nearest pressure and volume; viz.,

$$p = 746.63, \text{ and } v = 800; \text{ also, } 1.7^\circ \text{ C.} = 275.4 \text{ absolute};$$

then, $$R = \frac{pv}{\tau} = \frac{746.63 \times 800}{275.4} = 213.255. \quad . \quad . \quad . \quad . \quad (12)$$

This may be called a *relative* value of R, since the value of v is relative; then,

$$v = \frac{R\tau}{p} = \frac{213.255 \times 275.4}{764.42} = 781.396.$$

Hence the required ratio is $\frac{5.60188}{781.396}$; and this must be multiplied into the volumes in the foregoing table to reduce them to cubic feet, and the pressures in pounds per square foot will be the millimetres into $\frac{2116.3}{760}$.

In this manner Table IV. was calculated. We are now prepared to find the constants in equation (10); viz., a, b, and n, by taking three pairs of values of p and v from the table,

where $p = 2079.07$; $v = 5.7353$; $\therefore pv = 11924.08$;

where $p = 2783.71$; $v = 4.250$; $\therefore pv = 11830.76$;

where $p = 3347.74$; $v = 3.5077$; $\therefore pv = 11742\ 86$.

Then, $$\tau a - \frac{b}{(5.7353)^n} = 11924.08,$$

$$\tau a - \frac{b}{(4.25)^n} = 11830.76,$$

$$\tau a - \frac{b}{(3.5077)^n} = 11742.86,$$

which by reduction becomes

$$\frac{(24.376)^n - (14.907)^n}{(20.117)^n - (14.907)^n} = 1.9419. \quad . \quad . \quad . \qquad (13)$$

By trial, n was found to be 1.59. Substituting 1.59 for n, we get

$$a = 23.87 \text{ and } b = 2457.45,$$

and equation (11) becomes

$$\frac{pv}{\tau} = 23.87 - \frac{2457.45}{\tau v^{1.59}}. \quad . \quad . \quad . \quad . \quad . \quad . \qquad (14)$$

Therefore, equation (6) gives, for the latent heat of vaporization,

$$h_e = \left(23.87 - \frac{2457.45}{\tau v^{1.59}}\right)\left(1439 + \frac{2 \times 232659}{\tau}\right) \times \frac{2.3026}{778}. \qquad (15)$$

For the state where

$$v = 5.60188,$$

equation (14) will be

$$p = \frac{23.87}{5.60188}\tau - \frac{2457.45}{(5.60188)^{2.59}}; \quad . \quad . \quad . \quad . \quad . \qquad (16)$$

and by substituting this in equation (3) we get

$$\log\left(\frac{23.87}{5.60188}\tau - \frac{2457.45}{(5.60188)^{2.59}}\right) = 7.39136 - \frac{1439}{\tau} - \frac{232659}{\tau^2};$$

$\therefore$ (by trial) $\tau = 472.03°$: and this value in equation (16) gives $p = 1988.84$ pounds per square foot.

Also, p and τ in equation (15) give

$$h_e = 168.172 \text{ B.T.U.}$$

For the state $v = 2$ on the curve of saturation, we find,

$$\tau = 519.96°,$$
$$p = 5796.574 \text{ pounds},$$
$$h_e = 151.227 \text{ B.T.U.}$$

For the state $v = 8$ on the curve of saturation, we find,

$$\tau = 457.22°,$$
$$p = 1352.972 \text{ pounds},$$
$$h_e = 172.126 \text{ B.T.U.}$$

Assuming the form adopted by Regnault,

$$h_e = d + eT + fT^2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (17)$$

the above results enable us to find the constants d, e, and f, and the equation becomes

$$h_e = 171.26 - .25605 T - 0013795 T^2, \quad . \quad . \quad . \quad (18)$$

as a more practical formula for the latent heat of vaporization The column of latent heats in the general table was calculated from this formula. Fig. 22 represents this formula.

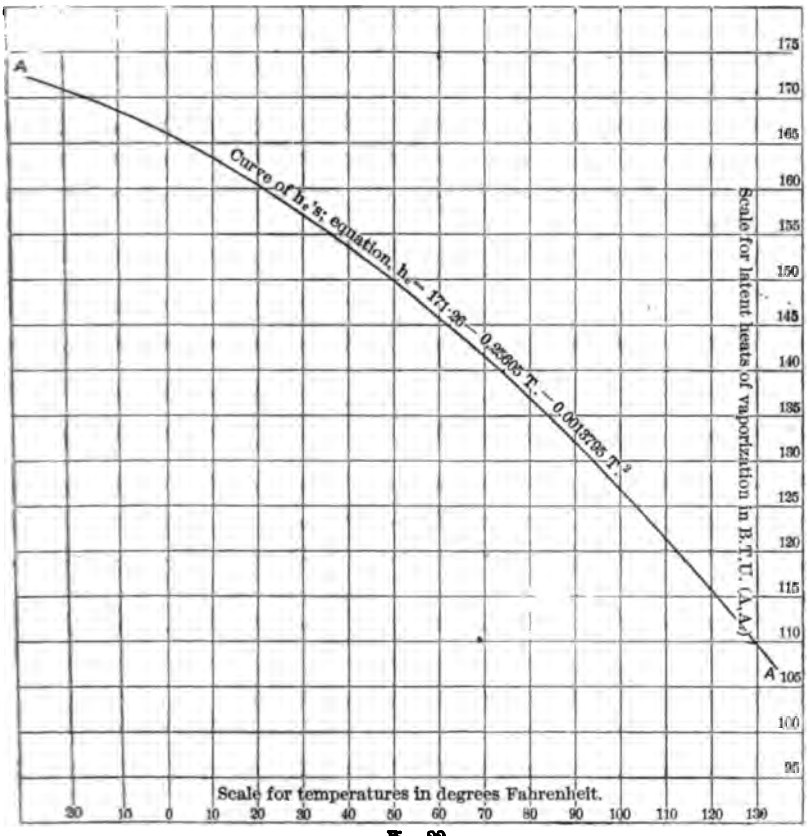


Fig. 22.

To find the specific volume of sulphur dioxide in the gaseous state we have, from thermodynamics,

$$v_2 - v_1 = \frac{778\ h_e}{\tau \dfrac{dp}{d\tau}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

From equation (3), by differentiating,

$$\tau \frac{dp}{dt} = \frac{p}{\tau}\left(1439 + \frac{2 \times 232659}{\tau}\right) \times 2.3026 ; \quad . \quad . \quad (20)$$

therefore, $$v = \frac{h_e}{\frac{p}{\tau}\left(1439 + \frac{2 \times 232659}{\tau}\right)} \times \frac{778}{2.3026} + v_1. \quad . \quad . \quad (21)$$

The value of v_1 is given in equation (40), farther on.

The column of specific volumes in the general table was calculated from formula (21).

Isothermals of sulphur dioxide. If the vapor be saturated, the isothermal will be parallel to the axis of v. If the vapor be superheated, the equation will be (10), substituting values for the constants:

$$pv = 23.87\tau_1 - \frac{2457.45}{v^{1.59}}. \quad . \quad . \quad . \quad . \quad . \quad (22)$$

Adiabatics of sulphur dioxide. If the vapor be saturated, the equation will be

$$u = xv = \left(c \log \frac{\tau_1}{\tau} + \frac{x_1 h_e}{\tau_1}\right)\frac{\tau v}{h_e}, \quad . \quad . \quad . \quad . \quad (23)$$

where u is the volume of the xth part of a pound of vapor and c the specific heat of liquid sulphur dioxide.

If the vapor be superheated, substitute, in the general equation,

$$dH = K_v d\tau + \tau \left(\frac{dp}{d\tau}\right) dv, \quad . \quad . \quad . \quad . \quad . \quad (24)$$

the value of $\left(\frac{dp}{d\tau}\right)_v$, from equation (10), and find,

$$\therefore dH = K_v d\tau + \tau \frac{a}{v} dv. \quad . \quad . \quad . \quad . \quad . \quad (25)$$

But $dH = o$, for adiabatic expansion, and (25) becomes

$$K_v d\tau = -\tau \frac{a}{v} dv ; \text{ or, } K_v \frac{d\tau}{\tau} = -a \frac{dv}{v} ; \quad . \quad . \quad . \quad (26)$$

then, $$K_v \log \frac{\tau}{\tau} = a \log \frac{v_1}{v} ; \therefore \frac{\tau}{\tau_1} = \left(\frac{v_1}{v}\right)^{\lambda}, \quad . \quad . \quad . \quad . \quad (27)$$

where τ_1 and v_1 are initial limits.

Regnault gives, as the specific heat of sulphur dioxide gas at constant pressure, 0.15438 (*Relation des Expériences*, Vol. II., page 146), and $K_v = K_p - R$, from thermodynamics, when the respective specific heats are considered constant.

Equation (9) gives $R = 24.06378$, which in thermal units becomes

$$24.06378 \div 778 = .03093;$$

hence, at this state,

$$k_v = .15438 - .03093 = .12345 \text{ in thermal units.} \quad (28)$$

To find λ we have, in equation (27),

$$\lambda = \frac{a}{K_v} = \frac{23.87}{778 \times .12345} = .24853.$$

To obtain an equation between p and v, eliminate τ from (10 and (27), giving, for the adiabatic,

$$pv = a\tau_1 \left(\frac{v_1}{v}\right)^{\lambda} - \frac{b}{v^n}; \quad \ldots \ldots \quad (29)$$

or, by substituting values for the constants,

$$\frac{\tau}{\tau_1} = \left(\frac{v_1}{v}\right)^{.24853}, \quad \ldots \ldots \quad (30)$$

and

$$p = 23.87 \frac{\tau_1}{v_1} \left(\frac{v_1}{v}\right)^{1.24853} - \frac{2457.45}{v^{2.59}} \quad \ldots \quad (31)$$

$$= 23.87 \frac{\tau}{v_1} \left(\frac{\tau}{\tau_1}\right)^{5.0237} - \frac{2457.45}{v_1^{2.59}} \left(\frac{\tau}{\tau_1}\right)^{10.4214}, \quad \ldots \quad (32)$$

the last of which is in terms of p and τ as variables.

Specific heat of the saturated vapor of sulphur dioxide.

We have, for the specific heat:

$$s = c - \frac{he}{\tau} + \frac{dh_e}{d\tau} \text{(see Wood's } \textit{Thermo.}\text{, page 147).} \quad (33)$$

From equation (3), by differentiating,

$$\tau \frac{dp}{dt} = \frac{p}{\tau}\left(1439 + \frac{2 \times 232659}{\tau}\right) \times 2.3026 ; \quad . \quad . \quad (20)$$

therefore, $$v = \frac{h_e}{\frac{p}{\tau}\left(1439 + \frac{2 \times 232659}{\tau}\right)} \times \frac{778}{2.3026} + v_1. \quad . \quad . \quad (21)$$

The value of v_1 is given in equation (40), farther on.

The column of specific volumes in the general table was calculated from formula (21).

Isothermals of sulphur dioxide. If the vapor be saturated, the isothermal will be parallel to the axis of v. If the vapor be superheated, the equation will be (10), substituting values for the constants:

$$pv = 23.87\tau_1 - \frac{2457.45}{v^{1.59}}. \quad . \quad . \quad . \quad . \quad . \quad (22)$$

Adiabatics of sulphur dioxide. If the vapor be saturated, the equation will be

$$u = xv = \left(c \log \frac{\tau_1}{\tau} + \frac{x_1\ h_e}{\tau_1}\right)\frac{\tau v}{h_e}, \quad . \quad . \quad . \quad . \quad (23)$$

where u is the volume of the xth part of a pound of vapor and c the specific heat of liquid sulphur dioxide.

If the vapor be superheated, substitute, in the general equation,

$$dH = K_v d\tau + \tau \left(\frac{dp}{d\tau}\right) dv, \quad . \quad . \quad . \quad . \quad . \quad (24)$$

the value of $\left(\frac{dp}{d\tau}\right)_v$, from equation (10), and find,

$$\therefore dH = K_v d\tau + \tau \frac{a}{v} dv. \quad . \quad . \quad . \quad . \quad . \quad (25)$$

But $dH = o$, for adiabatic expansion, and (25) becomes

$$K_v d\tau = -\tau \frac{a}{v} dv ; \text{ or, } K_v \frac{d\tau}{\tau} = -a \frac{dv}{v} ; \quad . \quad . \quad . \quad (26)$$

then, $$K_v \log \frac{\tau}{\tau} = a \log \frac{v_1}{v} ; \therefore \frac{\tau}{\tau_1} = \left(\frac{v_1}{v}\right)^{\lambda}, \quad . \quad . \quad . \quad . \quad (27)$$

where τ_1 and v_1 are initial limits.

Therefore, by substituting the values given from (34), (35), and (36), in equation (33),

$$s = \frac{3.5284}{\tau} - 0.390 - 0.00137\tau, \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

which is negative for all ordinary temperatures, and hence will be classed with steam-like vapors.

Density of liquid sulphur dioxide as compared with water. Table V. shows the results of a series of experiments by D'Andreèf on the specific gravity of liquid sulphur dioxide (Smithsonian *Miscellaneous Collections*, Vol. XXXII., 1888), which experiments are plotted in Fig. 24.

TABLE V.

TAKEN FROM THE SMITHSONIAN "MISCELLANEOUS COLLECTIONS," VOL. XXXII.

Specific gravity.	Temp. deg. C.	Temp. deg. Fahr.	Authority.
1.42	......	......	Faraday, P.T., 189. 1823.
1.45	......	......	Bussy, P.A., 1287.
1.4911	−20.5	− 4.9	D'Andreèf, Ann. [3], 58, 817
1.4609	− 9.9	+14.58	
1.4384	− 2.08	28.26	
1.4318	− 0.25	31.55	
1.4252	+ 2.8	37.04	
1.4205	+ 4.51	41.20	
1.4102	8.27	46.85	
1.4017	16.43	61.57	
1.3769	20.63	69.13	
1.3673	23.91	71.04	
1.3537	26.9	80.42	
1.3513	29.57	85.43	
1.3415	32.96	91.33	
1.3350	35.29	95.52	
1.3258	38.65	101.57	

These may be expressed by the formula:

$$\delta = a - bT$$

$$=1.48402 - .0015659T. \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

The values for δ in Table VI. were calculated from this formula,

and, assuming the volume of a pound of water to be 0.016 of a cubic foot, we have :

TABLE VI.

Temp. deg. Fahr.	δ = density.	v_2 = volume.	Temp. deg. Fahr.	δ = density.	v_2 = volume.
−20	1.52	.01056	65	1.3822	.01158
−15	1.51	.01061	70	1.3744	.01164
−10	1.4997	.01066	75	1.3666	.01171
− 5	1.4918	.01070	80	1.3587	.01177
± 0	1.4840	.01078	85	1.3509	.01184
+ 5	1.4762	.01084	90	1.3431	.01191
10	1.4645	.01092	95	1.3352	.01198
15	1.4605	.01096	100	1.3274	.01205
20	1.4527	.01101	105	1.3196	.01212
25	1.4449	.01107	110	1.3118	.01219
30	1.4370	.01113	115	1.3039	.01224
35	1.4292	.011196	120	1.2961	.01234
40	1.4214	.01125	125	1.2883	.01242
45	1.4135	.01131	130	1.2804	.01249
50	1.4057	.01138	135	1.2726	.01257
55	1.3979	.01145	140	1.2648	.01265
60	1.3901	.01151			

$$v_1 = \frac{0.016}{1.48402 - .0015659\,T}. \qquad . \quad . \quad . \quad . \quad . \quad (40)$$

which gives sufficiently accurate values for temperatures from −5° to 100° Fahr. The column of specific volumes in the general table was calculated from this formula.

Specific heat of liquefied sulphur dioxide. As an expression for the specific heat we have, from my paper on *Some Properties of Ammonia*, presented at the present meeting,

$$c = \frac{v}{J}\frac{dp}{d\tau} - \frac{dh_e}{d\tau} - k_p \left[\frac{d\tau^1}{d\tau} - 1\right] \cdot \quad . \quad . \quad . \quad . \quad . \quad (41)$$

From equation (3) find,

$$\frac{dp}{d\tau} = 2.3026 \left[1439 + \frac{465318}{\tau}\right] \frac{p}{\tau^2}; \quad . \quad . \quad . \quad . \quad (42)$$

from (37),

$$-\frac{dh_e}{d\tau} = -1.0149 + 0.002759\tau; \quad . \quad . \quad . \quad . \quad (43)$$

and from (32), changing $d\tau$ to $d\tau'$, and dropping all subscripts in the right-hand member, find,

$$\frac{dp}{d\tau'} = \frac{119.9157}{v} - \frac{25610.0694}{\tau v^{2.59}}. \quad . \quad . \quad . \quad . \quad (44)$$

These, substituted in equation (41), give,

$$c = \left(1439 + \frac{465318}{\tau}\right)\left(\frac{2.3026}{778} - \frac{0.35651}{119.9157 - \dfrac{21610.0694}{\tau v^{1.59}}}\right)\frac{pv}{\tau^2}$$

$$- 0.8602 + 0.002759\tau. \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

We have found for,

$v = 0.906$, $\tau = 560.66$, $T = 100°$ Fahr., $p = 12150.41$.
$v = 2$, $\tau = 519.96$, $T = 59.3°$ Fahr., $p = 5796.57$.
$v = 8$, $\tau = 457.92$, $T = -2.74°$ Fahr., $p = 1352.97$.
$v = 12.395$, $\tau = 440.66$, $T = -20°$ Fahr., $p = 846.526$.

These, in (45), give for

$T = 100°$ Fahr., $c = 0.5511$.
$T = 59.3°$ Fahr., $c = 0.5058$.
$T = -2.74°$ Fahr., $c = 0.3957$.
$T = -20°$ Fahr., $c = 0.3513$.

The specific heat, according to these results, increases between $-20°$ and -2.74, at the rate of 0.00257 per degree Fahr., between -2.74 and $59.3°$; at the rate 0.00177, and between $59.3°$ and $100°$, at the rate 0.00112, so that the rate of decrease of the specific heat decreases with increase of temperature, as it ought for imperfect fluids. Assuming that the rate of change is linear, or

$$c = a + bT, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (46)$$

between $-20°$ Fahr. and $100°$ Fahr., and we find

$$c = 0.3846 + 0.00166T.$$

But this will give values less than equation (45) for all temperatures between $-20°$ and $100°$. Making the function pass through $T = -2.74°$ Fahr. with the same slope, we have, at $T = o$,

$$c = 0.3957 + 0.00166 \times 2.74 = 0.4002,$$

and for any temperature T,

$$c = 0.4002 + 0.00166T. \qquad (47)$$

The rate of increase is greater than has been found experimentally for any liquid.

GENERAL TABLE.

SATURATED SULPHUR DIOXIDE.

Temperature, Fahr. T	Temperature, absolute, τ	Pressure, lbs. per square foot.	Pressure, lbs. per sq. in.	Heat of vaporization. Thermal units $= h_e$	External heat. Thermal units $= \frac{hv}{J}$	Internal heat, thermal units. $p = h_e - \frac{hv}{J}$	Vol. of vapor per lb. cu. ft. v	Vol. of liquid per lb. cu. ft. v_1	Weight of a cu. ft. of vapor lbs. $\frac{1}{v}$
−20°	440.66°	845.526	5.878	175.829	13.487	162.342	12.40572	.01056	.08068
−15°	445.66°	950.986	6.604	174.790	13.580	161.210	11.12000	.01061	.09001
−10°	450.66°	1133.102	7.868	173.683	13.754	159.829	9.45485	.01066	.10590
− 5°	455.66°	1296.399	9.002	172.506	13.877	158.629	8.33867	.01070	.12008
± 0°	460.66°	1483.226	10.300	171.260	13.991	157.269	7.34900	.01078	.13627
+ 5°	465.66°	1690.709	11.741	169.945	14.095	155.850	6.49701	.01084	.15418
+10°	470.66°	1921.648	13.344	168.562	14.199	154.363	5.75960	.01092	.17395
15°	475.66°	2176.586	15.115	167.109	14.281	152.828	5.11554	.01096	.19909
20°	480.66°	2457.702	17.067	165.587	14.360	151.227	4.55676	.01101	.21999
25°	485.66°	2776.812	19.214	163.997	14.430	149.567	4.06859	.01107	.24645
30°	490.66°	3105.809	21.568	162.337	14.490	147.847	3.64071	.01113	.27551
35°	495.66°	3476.634	24.309	160.608	14.540	146.068	3.26692	.01110	.30734
40°	500.66°	3881.261	26.953	158.811	14.580	144.231	2.93377	.01125	.34217
45°	505.66°	4430.718	30.764	156.944	14.609	142.335	2.57652	.01131	.38983
50°	510.66°	4800.128	33.334	155.009	14.627	140.382	2.38216	.01138	.42180
55°	515.66°	5318.562	36.934	153.004	14.634	138.370	2.15218	.01145	.46713
60°	520.66°	5879.188	40.828	150.931	14.630	136.301	1.94756	.01151	.51652
65°	525.66°	6484.184	45.029	148.788	14.614	134.174	1.76507	.01158	.57029
70°	530.66°	7135.782	49.554	146.577	14.587	131.990	1.60197	.01164	.62880
75°	535.66°	7836.195	54.418	144.297	14.546	129.751	1.45590	.01171	.69248
80°	540.66°	8584.079	59.612	142.447	14.544	127.903	1.32998	.01177	.75861
85°	545.66°	9392.581	65.226	139.529	14.414	125.115	1.10205	.01184	.83755
90°	550.66°	10253.125	71.202	137.042	14.339	122.703	1.10022	.01191	.91886
95°	555.66°	11171.652	77.580	134.485	14.257	120.228	1.00381	.01198	1.00722
100°	560.66°	12150.406	84.380	131.860	14.154	117.706	.91822	.01205	1.10342
105°	565.66°	13191.894	91.659	129.166	14.031	115.135	.83960	.01212	1.20849
110°	570.66°	14292.799	99.255	126.403	13.896	112.507	.76858	.01219	1.32207
115°	575.66°	15471.833	107.443	123.570	13.746	109.824	.70344	.01224	1.44678
120°	580.66°	16714.965	116.076	120.669	13.583	107.086	.64454	.01234	1.58177
125°	585.66°	18029.826	125.207	117.699	13.403	104.296	.59077	.01242	1.72904
130°	590.66°	19487.829	135.332	114.670	13.206	101.464	.53973	.01249	1.89668
135°	595.66°	20884.678	145.032	111.552	12.998	98.554	.49677	.01257	2.06528
140°	600.66°	22428.933	155.756	108.375	12.742	95.663	.45462	.01265	2.26258

It will be seen from Table IV. and from Fig. 23, that the isothermal of 35° Fahr., found experimentally by Regnault, is only from .055 to 1.51 pounds above the curve of saturation at corresponding volumes, and is limited between 3.50 and 5.73 cubic feet. These ranges are too small to give confidence in deduced values much above the highest or below the lowest; and particularly so in the case of the specific heat of the liquid which involves so much refined analysis, and other deduced values. We therefore will find a value for it between volumes of 3.5 and 5.7 cubic feet, say at 4.5 cubic feet, and consider the value as constant; this gives

$$c = 0.41. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (48)$$

If the superheated vapor be considered a perfect gas in establishing equation (45), the resulting value of c will be over 0.6

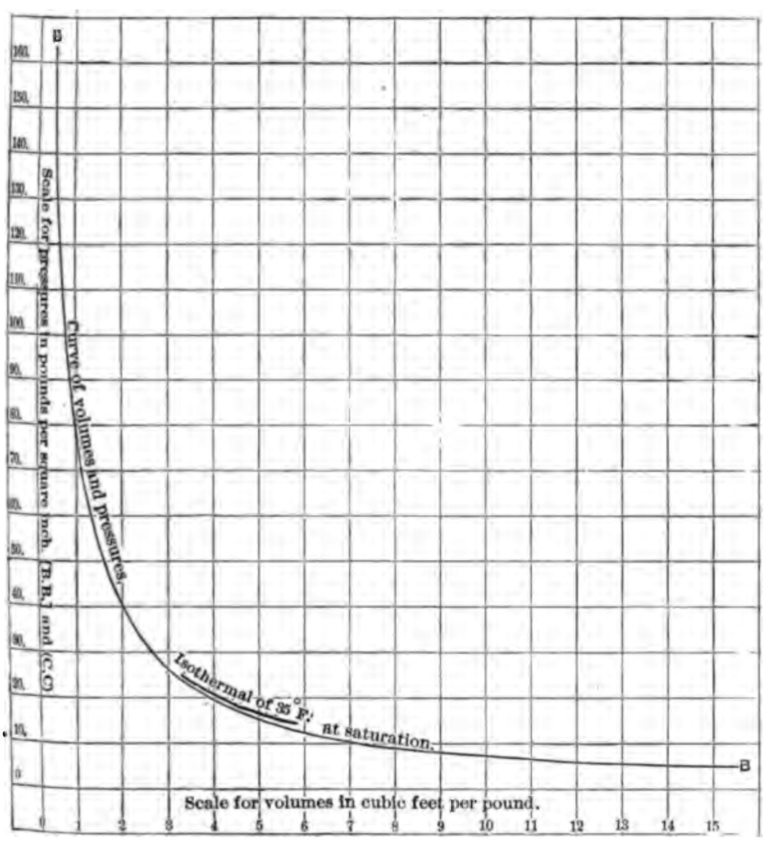


Fig. 23.

for T = 100°, and about the same as found for T = − 20°, so that the rate of change would be somewhat greater than that given in equation (47).

By comparing this table with Table III., in the paper by Prof. Jacobus, it will be found that:

Temperature deg. Fahr.	Experimental.	Wood.	Difference.
−10	170.20	173.68	3.48
0	165.06	171.26	6.20
20	152.07	165.58	13.51

The difference between the experimental values and those of the writer increases too rapidly. Taking the rate of difference between the first two values, and determining the rate of decrease from the first and third of the above experiments, we may write the empirical formula,

$$h_e = 171 - 0.605\,T, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

which will be sufficiently exact from − 15° Fahr. to 40° Fahr.

CCCCXX.*

THEORETICAL INVESTIGATION OF THE EFFICIENCY OF VAPOR ENGINES.

BY DE VOLSON WOOD, HOBOKEN, N. J.

(Member of the Society.)

IN the year 1884 Messrs. Gantt and Maury made an investigation similar to that contained in this paper, but having discovered certain defects therein, especially in the case of ether, I desired to have the entire investigation reviewed, and for this purpose was fortunate in securing the services of Messrs. J. T. Wescott and L. R. Mendoza, members of the graduating class of Stevens Institute of 1890, who made all the computations, and my labor consists chiefly of condensing, arranging, and editing their work. Some of the methods and references are different from those of the previous investigators. The fluids considered are: steam (H_2O), alcohol (C_2H_6O), carbon disulphide (CS_2), chloroform ($CHCl_3$), ether ($C_4H_{10}O$), and ammonia (NH_3). All these, except ammonia, were investigated by Gantt and Maury.

The following notation will be used throughout this work:

T_1 = initial temperature in degrees Fahrenheit.

T_2 = final temperature in degrees Fahrenheit.

τ_1 = absolute initial temperature.

τ_2 = final initial temperature.

p_1 = initial pressure in pounds per square foot.

p_2 = final pressure in pounds per square foot.

h_{e_1}, h_{e_2} = the corresponding latent heats of evaporation of 1 pound of the fluid in B. T. U.

H_{e_1}, H_{e_2} = latent heats of 1 pound in foot pounds.

L_1, L_2 = latent heats of 1 cubic foot in foot pounds.

w_1, w_2 = weights of 1 cubic foot of vapor.

v_1, v_2 = volumes that the fluid would occupy if it were all vapor.

u_1, u_2 = initial and final actual volumes.

K = mean specific heat of the liquid between the temperatures τ_1 and τ_2.

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

From equation (3), by differentiating,

$$\tau \frac{dp}{dt} = \frac{p}{\tau}\left(1439 + \frac{2 \times 232659}{\tau}\right) \times 2.3026 \, ; \quad . \quad . \quad (20)$$

therefore,
$$v = \frac{h_e}{\frac{p}{\tau}\left(1439 + \frac{2 \times 232659}{\tau}\right)} \times \frac{778}{2.3026} + v_1. \quad . \quad . \quad (21)$$

The value of v_1 is given in equation (40), farther on.

The column of specific volumes in the general table was calculated from formula (21).

Isothermals of sulphur dioxide. If the vapor be saturated, the isothermal will be parallel to the axis of v. If the vapor be superheated, the equation will be (10), substituting values for the constants:

$$pv = 23.87\tau_1 - \frac{2457.45}{v^{1.59}}. \quad . \quad . \quad . \quad . \quad . \quad (22)$$

Adiabatics of sulphur dioxide. If the vapor be saturated, the equation will be

$$u = xv = \left(c \log \frac{\tau_1}{\tau} + \frac{x_1 \, h_e}{\tau_1}\right)\frac{\tau v}{h_e}, \quad . \quad . \quad . \quad . \quad (23)$$

where u is the volume of the xth part of a pound of vapor and c the specific heat of liquid sulphur dioxide.

If the vapor be superheated, substitute, in the general equation,

$$dH = K_v d\tau + \tau \left(\frac{dp}{d\tau}\right) dv, \quad . \quad . \quad . \quad . \quad . \quad (24)$$

the value of $\left(\frac{dp}{d\tau}\right)_v$, from equation (10), and find,

$$\therefore dH = K_v d\tau + \tau \frac{a}{v} dv. \quad . \quad . \quad . \quad . \quad . \quad (25)$$

But $dH = o$, for adiabatic expansion, and (25) becomes

$$K_v d\tau = -\tau \frac{a}{v} dv \, ; \text{ or, } K_v \frac{d\tau}{\tau} = -a \frac{dv}{v} \, ; \quad . \quad . \quad . \quad (26)$$

then,
$$K_v \log \frac{\tau}{\tau} = a \log \frac{v_1}{v} \, ; \therefore \frac{\tau}{\tau_1} = \left(\frac{v_1}{v}\right)^{\lambda}, \quad . \quad . \quad . \quad . \quad (27)$$

where τ_1 and v_1 are initial limits.

Regnault gives, as the specific heat of sulphur dioxide gas at constant pressure, 0.15438 (*Relation des Expériences*, Vol. II., page 146), and $K_v = K_p - R$, from thermodynamics, when the respective specific heats are considered constant.

Equation (9) gives $R = 24.06378$, which in thermal units becomes

$$24.06378 \div 778 = .03093;$$

hence, at this state,

$$k_v = .15438 - .03093 = .12345 \text{ in thermal units.} \quad . \quad (28)$$

To find λ we have, in equation (27),

$$\lambda = \frac{a}{K_v} = \frac{23.87}{778 \times .12345} = .24853.$$

To obtain an equation between p and v, eliminate τ from (10 and (27), giving, for the adiabatic,

$$pv = a\tau_1 \left(\frac{v_1}{v}\right)^{\lambda} - \frac{b}{v^n}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

or, by substituting values for the constants,

$$\frac{\tau}{\tau_1} = \left(\frac{v_1}{v}\right)^{.24853}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

and

$$p = 23.87 \frac{\tau_1}{v_1} \left(\frac{v_1}{v}\right)^{1.24853} - \frac{2457.45}{v^{2.59}} \quad . \quad . \quad . \quad . \quad (31)$$

$$= 23.87 \frac{\tau}{v_1} \left(\frac{\tau}{\tau_1}\right)^{5.0237} - \frac{2457.45}{v_1^{2.59}} \left(\frac{\tau}{\tau_1}\right)^{10.4214}, \quad . \quad . \quad (32)$$

the last of which is in terms of p and τ as variables.

Specific heat of the saturated vapor of sulphur dioxide.

We have, for the specific heat:

$$s = c - \frac{he}{\tau} + \frac{dh_e}{d\tau} \text{ (see Wood's } \textit{Thermo.}\text{, page 147).} \quad . \quad (33)$$

From equation (47), for the value of the specific heat of the liquid,

$$c = 0.41. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (34)$$

From equation (18),

$$h_e = -3.5284 + 1.0349\tau - .00138\tau\ ;$$

or,

$$-\frac{h_e}{\tau} = \frac{3.5284}{\tau} - 1.0349 + .00138\tau. \quad . \quad . \quad . \qquad (35)$$

By differentiating, we get

$$+\frac{dh_e}{d\tau} = -0.25605 - 0.002759T \quad . \quad . \quad . \qquad (36)$$

$$= 1.0149 - 0.00276\tau. \quad . \quad . \quad . \quad . \quad . \qquad (37)$$

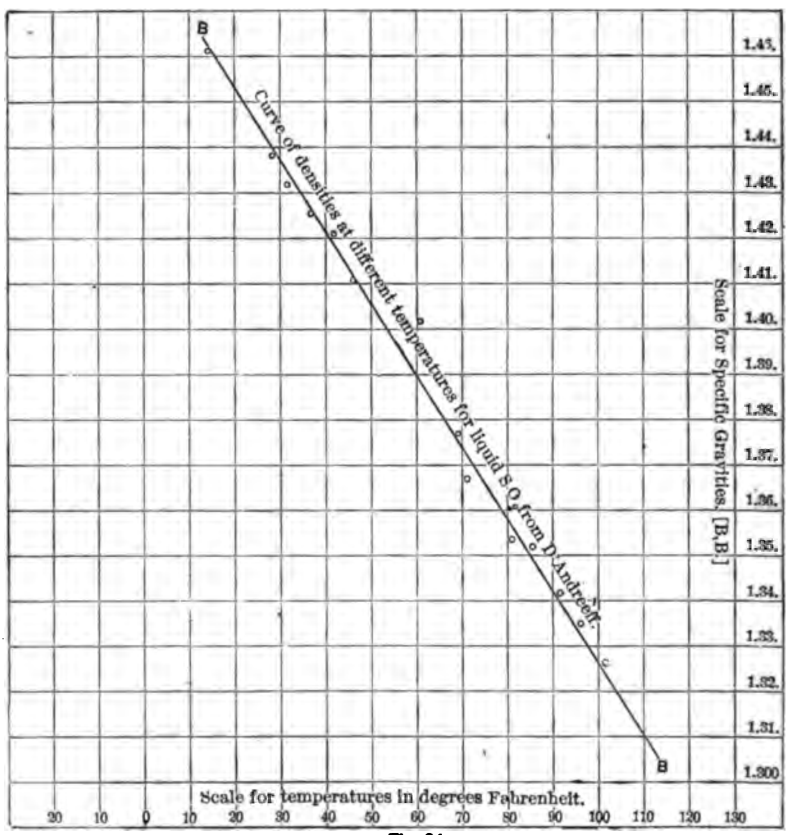


Fig. 24.

Therefore, by substituting the values given from (34), (35), and (36), in equation (33),

$$s = \frac{3.5284}{\tau} - 0.390 - 0.00137\tau, \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

which is negative for all ordinary temperatures, and hence will be classed with steam-like vapors.

Density of liquid sulphur dioxide as compared with water. Table V. shows the results of a series of experiments by D'Andreëf on the specific gravity of liquid sulphur dioxide (Smithsonian *Miscellaneous Collections*, Vol. XXXII., 1888), which experiments are plotted in Fig. 24.

TABLE V.

TAKEN FROM THE SMITHSONIAN "MISCELLANEOUS COLLECTIONS," VOL. XXXII.

Specific gravity.	Temp. deg. C.	Temp. deg. Fahr.	Authority.
1.42	......	......	Faraday, P.T., 189. 1823.
1.45	......	......	Bussy, P.A., 1237.
1.4911	−20.5	− 4.9	D'Andreëf, Ann. [3], 56, 317
1.4609	− 9.9	+14.58	
1.4384	− 2.08	28.26	
1.4318	− 0.25	31.55	
1.4252	+ 2.8	37.04	
1.4205	+ 4.51	41.20	
1.4102	8.27	46.85	
1.4017	16.43	61.57	
1.3769	20.63	69.13	
1.3673	23.91	71.04	
1.3537	26.9	80.42	
1.3513	29.57	85.43	
1.3415	32.96	91.33	
1.3350	35.29	95.52	
1.3258	38.65	101.57	

These may be expressed by the formula:

$$\delta = a - bT$$

$$= 1.48402 - .0015659\,T. \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

The values for δ in Table VI. were calculated from this formula,

and, assuming the volume of a pound of water to be 0.016 of a cubic foot, we have :

TABLE VI.

Temp. deg. Fahr.	δ = density.	v_1 = volume.	Temp. deg. Fahr.	δ = density.	v_1 = volume.
−20	1.52	.01056	65	1.3822	.01158
−15	1.51	.01061	70	1.3744	.01164
−10	1.4997	.01066	75	1.3666	.01171
− 5	1.4918	.01070	80	1.3587	.01177
± 0	1.4840	.01078	85	1.3509	.01184
+ 5	1.4762	.01084	90	1.3431	.01191
10	1.4645	.01092	95	1.3352	.01198
15	1.4605	.01096	100	1.3274	.01205
20	1.4527	.01101	105	1.3196	.01212
25	1.4449	.01107	110	1.3118	.01219
30	1.4370	.01113	115	1.3039	.01224
35	1.4292	.011196	120	1.2961	.01234
40	1.4214	.01125	125	1.2883	.01242
45	1.4135	.01131	130	1.2804	.01249
50	1.4057	.01138	135	1.2726	.01257
55	1.3979	.01145	140	1.2648	.01265
60	1.3901	.01151			

$$v_1 = \frac{0.016}{1.48402 - .0015659\,T}. \quad . \quad . \quad . \quad . \quad . \quad (40)$$

which gives sufficiently accurate values for temperatures from −5° to 100° Fahr. The column of specific volumes in the general table was calculated from this formula.

Specific heat of liquefied sulphur dioxide. As an expression for the specific heat we have, from my paper on *Some Properties of Ammonia*, presented at the present meeting,

$$c = \frac{v}{J}\frac{dp}{d\tau} - \frac{dh_e}{d\tau} - k_p\left[\frac{d\tau^1}{d\tau} - 1\right] \cdot \cdot \cdot \cdot \cdot \cdot \quad (41)$$

From equation (3) find,

$$\frac{dp}{d\tau} = 2.3026\left[1439 + \frac{465318}{\tau}\right]\frac{p}{\tau^2}; \quad . \quad . \quad . \quad . \quad (42)$$

from (37),

$$-\frac{dh_e}{d\tau} = -1.0149 + 0.002759\tau; \quad . \quad . \quad . \quad . \quad (43)$$

and from (32), changing $d\tau$ to $d\tau'$, and dropping all subscripts in the right-hand member, find,

$$\frac{dp}{d\tau} = \frac{119.9157}{v} - \frac{25610.0694}{\tau v^{2.59}}. \quad . \quad . \quad . \quad . \quad (44)$$

These, substituted in equation (41), give,

$$c = \left(1439 + \frac{465318}{\tau}\right)\left(\frac{2.3026}{778} - \frac{0.35651}{119.9157 - \dfrac{21610.0694}{\tau v^{1.59}}}\right)\frac{pv}{\tau^2}$$

$$- 0.8602 + 0.002759\tau. \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

We have found for,

$v = 0.906$, $\tau = 560.66$, $T = 100°$ Fahr., $p = 12150.41$.
$v = 2$, $\tau = 519.96$, $T = 59.3°$ Fahr., $p = 5796.57$.
$v = 8$, $\tau = 457.92$, $T = -2.74°$ Fahr., $p = 1352.97$.
$v = 12.3995$, $\tau = 440.66$, $T = -20°$ Fahr., $p = 846.526$.

These, in (45), give for

$T = 100°$ Fahr., $c = 0.5511$.
$T = 59.3°$ Fahr., $c = 0.5058$.
$T = -2.74°$ Fahr., $c = 0.3957$.
$T = -20°$ Fahr., $c = 0.3513$.

The specific heat, according to these results, increases between $-20°$ and -2.74, at the rate of 0.00257 per degree Fahr., between -2.74 and $59.3°$, at the rate 0.00177, and between $59.3°$ and $100°$, at the rate 0.00112, so that the rate of decrease of the specific heat decreases with increase of temperature, as it ought for imperfect fluids. Assuming that the rate of change is linear, or

$$c = a + bT, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (46)$$

between $-20°$ Fahr. and $100°$ Fahr., and we find

$$c = 0.3846 + 0.00166T.$$

But this will give values less than equation (45) for all temperatures between $-20°$ and $100°$. Making the function pass through $T = -2.74°$ Fahr. with the same slope, we have, at $T = 0$,

$$c = 0.3957 + 0.00166 \times 2.74 = 0.4002,$$

and for any temperature T,

$$c = 0.4002 + 0.00166T. \qquad (47)$$

and, assuming the volume of a pound of water to be 0.016 of a cubic foot, we have :

TABLE VI.

Temp. deg. Fahr.	δ = density.	τ_1 = volume.	Temp. deg. Fahr.	δ = density.	τ_1 = volume.
−20	1.52	.01056	65	1.3822	.01158
−15	1.51	.01061	70	1.3744	.01164
−10	1.4997	.01066	75	1.3666	.01171
− 5	1.4918	.01070	80	1.3587	.01177
± 0	1.4840	.01078	85	1.3509	.01184
+ 5	1.4762	.01084	90	1.3431	.01191
10	1.4645	.01092	95	1.3352	.01198
15	1.4605	.01096	100	1.3274	.01205
20	1.4527	.01101	105	1.3196	.01212
25	1.4449	.01107	110	1.3118	.01219
30	1.4370	.01113	115	1.3039	.01224
35	1.4292	.011196	120	1.2961	.01234
40	1.4214	.01125	125	1.2883	.01242
45	1.4135	.01131	130	1.2804	.01249
50	1.4057	.01138	135	1.2726	.01257
55	1.3979	.01145	140	1.2648	.01265
60	1.3901	.01151			

$$v_1 = \frac{0.016}{1.48402 - .0015659\,T}. \quad . \quad . \quad . \quad . \quad . \quad (40)$$

which gives sufficiently accurate values for temperatures from $-5°$ to $100°$ Fahr. The column of specific volumes in the general table was calculated from this formula.

Specific heat of liquefied sulphur dioxide. As an expression for the specific heat we have, from my paper on *Some Properties of Ammonia*, presented at the present meeting,

$$c = \frac{v}{J}\frac{dp}{d\tau} - \frac{dh_e}{d\tau} - k_p\left[\frac{d\tau^1}{d\tau} - 1\right] \cdot \quad . \quad . \quad . \quad . \quad . \quad (41)$$

From equation (3) find,

$$\frac{dp}{d\tau} = 2.3026\left[1439 + \frac{465318}{\tau}\right]\frac{p}{\tau^2}; \quad . \quad . \quad . \quad . \quad (42)$$

from (37),

$$-\frac{dh_e}{d\tau} = -1.0149 + 0.002759\tau; \quad . \quad . \quad . \quad . \quad (43)$$

and from (32), changing $d\tau$ to $d\tau'$, and dropping all subscripts in the right-hand member, find,

$$\frac{dp}{d\tau} = \frac{119.9157}{v} - \frac{25610.0694}{\tau v^{2.59}}. \quad . \quad . \quad . \quad . \quad (44)$$

These, substituted in equation (41), give,

$$c = \left(1439 + \frac{465318}{\tau}\right)\left(\frac{2.3026}{778} - \frac{0.35651}{119.9157 - \dfrac{21610.0694}{\tau v^{1.59}}}\right)\frac{pv}{\tau^2}$$

$$- 0.8602 + 0.002759\tau. \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

We have found for,

$v = 0.906$, $\tau = 560.66$, $T = 100°$ Fahr., $p = 12150.41$.
$v = 2$, $\tau = 519.96$, $T = 59.3°$ Fahr., $p = 5796.57$.
$v = 8$, $\tau = 457.92$, $T = -2.74°$ Fahr., $p = 1352.97$.
$v = 12.395$, $\tau = 440.66$, $T = -20°$ Fahr., $p = 846.526$.

These, in (45), give for

$T = 100°$ Fahr., $c = 0.5511$.
$T = 59.3°$ Fahr., $c = 0.5058$.
$T = -2.74°$ Fahr., $c = 0.3957$.
$T = -20°$ Fahr., $c = 0.3513$.

The specific heat, according to these results, increases between $-20°$ and -2.74, at the rate of 0.00257 per degree Fahr., between -2.74 and $59.3°$; at the rate 0.00177, and between $59.3°$ and $100°$, at the rate 0.00112, so that the rate of decrease of the specific heat decreases with increase of temperature, as it ought for imperfect fluids. Assuming that the rate of change is linear, or

$$c = a + bT, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (46)$$

between $-20°$ Fahr. and $100°$ Fahr., and we find

$$c = 0.3846 + 0.00166\,T.$$

But this will give values less than equation (45) for all temperatures between -20 and $100°$. Making the function pass through $T = -2.74°$ Fahr. with the same slope, we have, at $T = o$,

$$c = 0.3957 + 0.00166 \times 2.74 = 0.4002,$$

and for any temperature T,

$$c = 0.4002 + 0.00166\,T. \qquad (47)$$

The rate of increase is greater than has been found experimentally for any liquid.

GENERAL TABLE.

SATURATED SULPHUR DIOXIDE.

Temperature, Fahr. T	Temperature, absolute, τ	Pressure, lbs. per square foot.	Pressure, lbs. per sq. in.	Heat of vaporization. Thermal units $= h_e$	External heat. Thermal units $= \frac{hv}{J}$	Internal heat, thermal units. $p = h_e - \frac{hv}{J}$	Vol. of vapor per lb. cu. ft. v	Vol. of liquid per lb. cu. ft. v_1	Weight of a cu. ft. of vapor lbs. $\frac{1}{v}$
−20°	440.66°	845.526	5.878	175.829	13.487	162.342	12.40572	.01056	.08068
−15°	445.66°	950.986	6.604	174.790	13.580	161.210	11.12000	.01061	.09001
−10°	450.66°	1133.102	7.868	173.683	13.754	159.829	9.45485	.01066	.10590
− 5°	455.66°	1296.399	9.002	172.506	13.877	158.629	8.33867	.01070	.12008
± 0°	460.66°	1483.226	10.300	171.260	13.991	157.269	7.34900	.01078	.13627
+ 5°	465.66°	1690.709	11.741	169.945	14.095	155.850	6.49701	.01084	.15418
+10°	470.66°	1921.648	13.344	168.562	14.199	154.363	5.75960	.01092	.17395
15°	475.66°	2176.586	15.115	167.109	14.281	152.828	5.11554	.01096	.19909
20°	480.66°	2457.702	17.067	165.587	14.360	151.227	4.55676	.01101	.21999
25°	485.66°	2776.812	19.214	163.997	14.430	149.567	4.06859	.01107	.24645
30°	490.66°	3105.809	21.568	162.337	14.490	147.847	3.64071	.01113	.27551
35°	495.66°	3476.634	24.309	160.608	14.540	146.068	3.26692	.01110	.30734
40°	500.66°	3881.261	26.953	158.811	14.580	144.231	2.93377	.01125	.34217
45°	505.66°	4430.718	30.764	156.944	14.609	142.335	2.57652	.01131	.38983
50°	510.66°	4800.128	33.334	155.009	14.627	140.382	2.38216	.01138	.42180
55°	515.66°	5318.562	36.934	153.004	14.634	138.370	2.15218	.01145	.46713
60°	520.66°	5879.188	40.828	150.931	14.630	136.301	1.94756	.01151	.51652
65°	525.66°	6484.184	45.029	148.788	14.614	134.174	1.76507	.01158	.57029
70°	530.66°	7135.782	49.554	146.577	14.587	131.990	1.60197	.01164	.62880
75°	535.66°	7836.195	54.418	144.297	14.546	129.751	1.45590	.01171	.69243
80°	540.66°	8584.079	59.612	142.447	14.544	127.903	1.32998	.01177	.75861
85°	545.66°	9392.581	65.226	139.529	14.414	125.115	1.10205	.01184	.88755
90°	550.66°	10253.125	71.202	137.042	14.339	122.703	1.10022	.01191	.91886
95°	555.66°	11171.652	77.580	134.485	14.257	120.228	1.00381	.01198	1.00722
100°	560.66°	12150.406	84.380	131.860	14.154	117.706	.91822	.01205	1.10342
105°	565.66°	13191.894	91.659	129.166	14.081	115.135	.83960	.01212	1.20849
110°	570.66°	14292.799	99.255	126.403	13.896	112.507	.76858	.01219	1.32207
115°	575.66°	15471.833	107.443	123.570	13.746	109.824	.70344	.01224	1.44678
120°	580.66°	16714.965	116.076	120.669	13.583	107.086	.64454	.01234	1.58177
125°	585.66°	18029.826	125.207	117.699	13.403	104.296	.59077	.01242	1.72904
130°	590.66°	19487.829	135.332	114.670	13.206	101.464	.53973	.01249	1.89668
135°	595.66°	20884.678	145.032	111.552	12.998	98.554	.49677	.01257	2.06528
140°	600.66°	22428.933	155.756	108.375	12.742	95.668	.45462	.01265	2.26258

It will be seen from Table IV. and from Fig. 23, that the isothermal of 35° Fahr., found experimentally by Regnault, is only from .055 to 1.51 pounds above the curve of saturation at corresponding volumes, and is limited between 3.50 and 5.73 cubic feet. These ranges are too small to give confidence in deduced values much above the highest or below the lowest; and particularly so in the case of the specific heat of the liquid which involves so much refined analysis, and other deduced values. We therefore will find a value for it between volumes of 3.5 and 5.7 cubic feet, say at 4.5 cubic feet, and consider the value as constant; this gives

$$c = 0.41. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (48)$$

If the superheated vapor be considered a perfect gas in establishing equation (45), the resulting value of c will be over 0.6

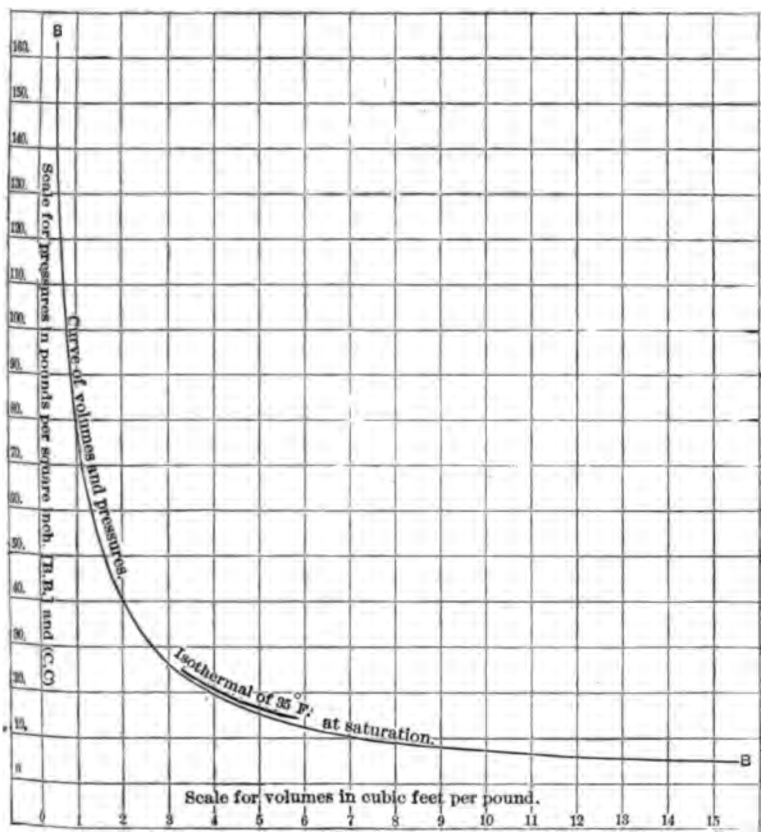


Fig. 23.

for T = 100°, and about the same as found for T = − 20°, so that the rate of change would be somewhat greater than that given in equation (47).

By comparing this table with Table III., in the paper by Prof. Jacobus, it will be found that:

Temperature deg. Fahr.	Experimental.	Wood.	Difference.
−10	170.20	173.68	3.48
0	165.06	171.26	6.20
20	152.07	165.58	13.51

The difference between the experimental values and those of the writer increases too rapidly. Taking the rate of difference between the first two values, and determining the rate of decrease from the first and third of the above experiments, we may write the empirical formula,

$$h_e = 171 - 0.605\,T, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

which will be sufficiently exact from − 15° Fahr. to 40° Fahr.

CCCCXX.*

THEORETICAL INVESTIGATION OF THE EFFICIENCY OF VAPOR ENGINES.

BY DE VOLSON WOOD, HOBOKEN, N. J.
(Member of the Society.)

IN the year 1884 Messrs. Gantt and Maury made an investigation similar to that contained in this paper, but having discovered certain defects therein, especially in the case of ether, I desired to have the entire investigation reviewed, and for this purpose was fortunate in securing the services of Messrs. J. T. Wescott and L. R. Mendoza, members of the graduating class of Stevens Institute of 1890, who made all the computations, and my labor consists chiefly of condensing, arranging, and editing their work. Some of the methods and references are different from those of the previous investigators. The fluids considered are: steam (H_2O), alcohol (C_2H_6O), carbon disulphide (CS_2), chloroform ($CHCl_3$), ether ($C_4H_{10}O$), and ammonia (NH_3). All these, except ammonia, were investigated by Gantt and Maury.

The following notation will be used throughout this work:

T_1 = initial temperature in degrees Fahrenheit.
T_2 = final temperature in degrees Fahrenheit.
τ_1 = absolute initial temperature.
τ_2 = final initial temperature.
p_1 = initial pressure in pounds per square foot.
p_2 = final pressure in pounds per square foot.
h_{e_1}, h_{e_2} = the corresponding latent heats of evaporation of 1 pound of the fluid in B. T. U.
H_{e_1}, H_{e_2} = latent heats of 1 pound in foot pounds.
L_1, L_2 = latent heats of 1 cubic foot in foot pounds.
w_1, w_2 = weights of 1 cubic foot of vapor.
v_1, v_2 = volumes that the fluid would occupy if it were all vapor.
u_1, u_2 = initial and final actual volumes.
K = mean specific heat of the liquid between the temperatures τ_1 and τ_2.

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

W_E = total work of expansion.
W_C = total work of compression.
H = the heat used in foot pounds.
E = efficiency.
J = dynamic unit of heat = 778 foot pounds.

CASE I.

In this case the fluids are worked in a Carnot's cycle, and it is designed to show, by numerical solutions, that the efficiency is the same for each of the fluids when worked between the same limits of temperature.

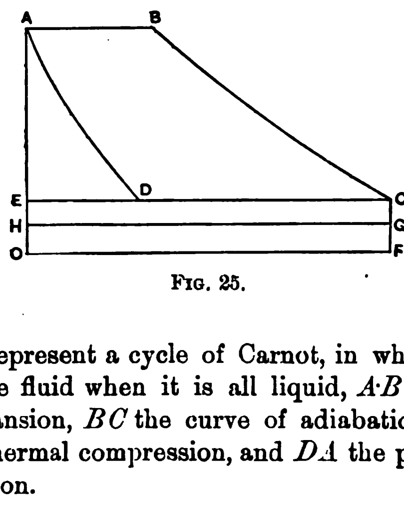


FIG. 25.

Let Fig. 25 represent a cycle of Carnot, in which A represents the state of the fluid when it is all liquid, AB its path during isothermal expansion, BC the curve of adiabatic expansion, CD the line of isothermal compression, and DA the path during adiabatic compression.

In order to find the work done by any of the fluids during expansion, it is necessary to derive an equation for the adiabatic BC. If the vapor be saturated, the equation is given in Wood's *Thermodynamics*, page 184, equation (a), as follows:

$$xv = \left(K \log_e \frac{\tau_1}{\tau} + \frac{x_1 h_{e_1}}{\tau_1}\right)\frac{\tau v}{h_e}, \quad . \quad . \quad . \quad . \quad . \quad (1)$$

in which x is the fractional part of 1 pound of the fluid that has been converted into vapor, $x_1 v_1$ = the volume of 1 pound of saturated vapor at the beginning of expansion, and xv = the volume of vapor at any point during expansion, the volume of the liquid being neglected.

If the fluid be all vapor at the beginning of expansion, $x_1 = 1$; and we have

$$xv = u = \frac{v\tau}{h_e}\left(K \log_e \frac{\tau_1}{\tau} + \frac{h_{e_1}}{\tau_1}\right) \quad . \quad . \quad . \quad . \quad . \quad (2)$$

If the fluid be all liquid at the beginning of expansion, $x_1 = 0$; and

$$u = \frac{v\tau}{h_e} K \log_e \frac{\tau_1}{\tau} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3')$$

If the adiabatic is that of a superheated vapor, then it will follow approximately the law of a perfect gas, and its equation will be of the form

$$u = u_1 \left(\frac{\tau_1}{\tau}\right)^{\frac{1}{\gamma - 1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3'')$$

where γ is the ratio of the two specific heats.

To find the work of expansion, W_E.

We have (Fig. 25).

$$W_E = ABCFO = ABCE + OC,$$

$$\text{but } ABCE = \int_{p2} udp \text{ and } OC = p_2 u_2.$$

$$\therefore W_E = \int_{p2}^{p1} udp + p_2 u_2.$$

By the aid of equation (1) we have

$$\int_{p2}^{p1} udp = JK \left\{ \tau_1 - \tau_2 \left(1 + \log_e \frac{\tau_1}{\tau_2}\right) \right\}$$
$$+ x_1 H_e \frac{\tau_1 - \tau_2}{\tau^1}.$$

$$\therefore W_E = p_2 u_2 + JK \left\{ \tau_1 - \tau_2 \left(1 + \log_e \frac{\tau_1}{\tau_2}\right) \right\}$$
$$+ x_1 H_{e_1} \frac{\tau_1 - \tau_2}{\tau_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Equation (3) is the expression for the work done by one pound of fluid.

If we consider the action of u_1 cubic feet, then equation (3) is to be multiplied by w_1u_1; and this value substituted in (4) will give

$$W_E = JKw_1u_1 \left\{ \tau_1 - \tau_2 \left(1 + log_e \frac{\tau_1}{\tau_2}\right) \right\}$$

$$+ x_1 w_1 H_{e_1} u_1 \frac{\tau_1 - \tau_2}{\tau_1} + p_2u_2 \quad . \quad . \quad . \quad . \quad . \quad (5)$$

If the fluid is all vapor at the point of cut-off, $x_1 = 1$; and substituting in (5), observing that $w_1 H_{e_1} = L_1$, we get

$$W_E = JKw_1u_1 \left\{ \tau_1 - \tau_2 \left(1 + log_e \frac{\tau_1}{\tau_2}\right) \right\} + p_2u_2$$

$$+ L_1u_1 \frac{\tau_1 - \tau_2}{\tau_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

This expression for the work of expansion is true for all vapors retaining the state of saturation during the adiabatic expansion. Later will be given the equations for superheated vapors.

To find the work of compression.

If the fluid expands adiabatically from state A till its temperature falls to τ_2, the work done by it will be the same as the work done upon it when compressed, without transmission of heat from D to A.

Therefore, making $x_1 = 0$ in (5), we obtain,

$$W_C = JKw_1u_1 \left\{ \tau_1 - \tau_2 \left(1 + log_e \frac{\tau_1}{\tau_2}\right) \right\} + p_2u_2. \quad . \quad . \quad (7)$$

Therefore, the effective work will be,

$$W_E - W_C = L_1u_1 \frac{\tau_1 - \tau_2}{\tau_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

The heat used is the latent heat of evaporation of u_1 cubic feet ef the fluid; or,

$$H = L_1u_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

$$\therefore E = \frac{W_E - W_C}{H} = \frac{\tau_1 - \tau_2}{\tau_1}; \quad . \quad . \quad . \quad . \quad (10)$$

being the efficiency of the perfect elementary engine, and equation (8) is the expression for the second law of thermodynamics.

Now we will proceed to find the numerical values of all the quantities that enter in the equations which we have established, and these values are tabulated at the end of this case.

First, we will consider those saturated vapors whose specific heat is negative, or "steam-like" vapors.

We have selected for the initial and final temperatures the respective values,

$$T_1 = 341.6° \text{ F. and } T_2 = 194° \text{ F.};$$

or,

$$\tau_1 = 802.26° ; \ \tau_2 = 654.66°.$$

To find the pressures corresponding to these temperatures, we have used Rankine's formula for saturated vapors, which is,

$$log_{10} p = A - \frac{B}{\tau} - \frac{C}{\tau^2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

in which A, B, and C are constants having the following values:

	A	$log\ B$	$log\ C$
Steam	8.28203	3.441474	5.583973
Alcohol	8.68170	3.4721707	5.4354446
Carbon Disulphide.	7.4263	3.3274293	5.1344146
Chloroform........	4.3807	[This B is 3.288394 negative]	6.1899631
Ammonia.........	8.4079	3.341632	....

Making the corresponding substitutions in formula (11) we find the following values for the pressures:

	p_1 lbs. per sq. ft.	p_2 lbs. per sq. ft.
Steam	17408	1469
Alcohol.................	36150	3279
Carbon Disulphide.......	36745	7269
Chloroform	24871	5432
Ammonia	468700	113100

Differentiating equation (11) we obtain,

$$\tau \frac{dp}{d\tau} = p \left(\frac{B}{\tau} + \frac{2C}{\tau^2} \right) \times 2.3026 = L_1. \quad . \quad . \quad . \quad (12)$$

Substituting in this equation we get for the latent heats,

	L_1	L_2
Steam	186905	20558
Alcohol................	381359	43791
Carbon Disulphide......	259844	64970
Chloroform	136930	58358
Ammonia..............	2954108	873561

To find the latent heats of evaporation per pound in B. T. U. we have resorted to the following formula,

$$h_e = a - bT - cT^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

in which a, b, and c are constants having the following values (reduced to British units from the corresponding metric values determined by Regnault):

	a	b	c
Steam	1121.7	0.6946	0.00002222
Alcohol	524.07	0.92211	−0.000679
Carbon Disulphide	164.57	0.0716	0.0002746
Chloroform	123.6	0.093	0.000282
Ammonia	555.5	0.613	0.000219

Substituting in equation (13) we have,

	h_{e_1} (Thermal units.)	h_{e_2} (Thermal units.)
Steam	873.59	978.44
Alcohol	288.19	370.73
Carbon Disulphide	108.07	140.34
Chloroform	58.92	94.94
Ammonia	320.54	428.34

Multiplying these quantities by 778 (the dynamic unit of heat), we obtain,

	H_{e_1} (Foot pounds.)	H_{e_2} (Foot pounds.)
Steam	679653	761226
Alcohol	224216	288428
Carbon Disulphide	84077	109188
Chloroform	45843	73867
Ammonia	249383	333246

The weight of 1 cubic foot of saturated vapor is the ratio of the two latent heats; *i.e.*,

$$w = \frac{L}{H_e}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

From this we obtain,

	w_1 (Weight in 1 cu. ft.)	w_2 (Weight in 1 cu. ft.)
Steam	0.275	0.027
Alcohol	1.701	0.151
Carbon Disulphide	3.091	0.595
Chloroform	2.978	0.723
Ammonia	11.842	2.621

The amount of heat to be used in this case will be made the same for each of the fluids, and equal to the amount of heat

The effect of condensation would modify the efficiencies, also compression to fill the clearance space in actual engines would modify the efficiency a small amount; still, considering the size of

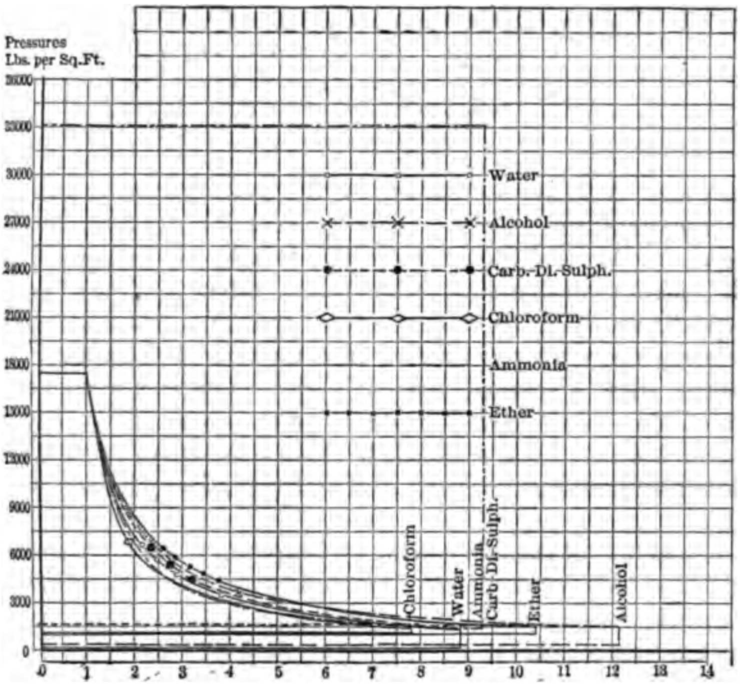


FIG. 29.

the engine, the cost of construction, the cost of the fluid, and the facility and safety in handling, it is safe to conclude that *water is by far the best medium of the substances considered for transforming heat into power.*

To find the latent heats of evaporation per pound in B. T. U. we have resorted to the following formula,

$$h_e = a - bT - cT^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

in which a, b, and c are constants having the following values (reduced to British units from the corresponding metric values determined by Regnault):

	a	b	c
Steam	1121.7	0.6946	0.00002222
Alcohol	524.07	0.92211	−0.000679
Carbon Disulphide	164.57	0.0716	0.0002746
Chloroform	123.6	0.093	0.000282
Ammonia	555.5	0.613	0.000219

Substituting in equation (13) we have,

	h_{e_1} (Thermal units.)	h_{e_2} (Thermal units.)
Steam	873.59	978.44
Alcohol	288.19	370.73
Carbon Disulphide	108.07	140.34
Chloroform	58.92	94.94
Ammonia	320.54	428.34

Multiplying these quantities by 778 (the dynamic unit of heat), we obtain,

	H_{e_1} (Foot pounds.)	H_{e_2} (Foot pounds.)
Steam	679653	761226
Alcohol	224216	288428
Carbon Disulphide	84077	109188
Chloroform	45843	73867
Ammonia	249383	333246

The weight of 1 cubic foot of saturated vapor is the ratio of the two latent heats; *i.e.*,

$$w = \frac{L}{H_e}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

From this we obtain,

	w_1 (Weight in 1 cu. ft.)	w_2 (Weight in 1 cu. ft.)
Steam	0.275	0.027
Alcohol	1.701	0.151
Carbon Disulphide	3.091	0.595
Chloroform	2.978	0.722
Ammonia	11.842	2.621

The amount of heat to be used in this case will be made the same for each of the fluids, and equal to the amount of heat necessary to evaporate 1 cubic foot of steam at the temperature 341°.6 F.

$$\therefore H = 186905. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Let the fluid be all vapor at B (Fig. 25); then, since the heat used is H, the volume occupied by the vapor at that point will be

$$u_1 = \frac{H}{L_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

This equation gives the following values:

	Steam.	Alcohol.	Carb. Dis.	Chloroform.	Ammonia.
$v_1 = u_1$	1	0.49	0.719	1.365	0.063

If the vapor did not condense during expansion, then its volume at the end of the expansion would be

$$v_2 = \frac{w_1 v_1}{w_2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

which gives

	Steam.	Alcohol.	Carb. Dis.	Chloroform.	Ammonia.
$v_2 =$	10.185	5.519	3.734	5.647	0.285

But in vapors whose specific heat is negative there is a partial condensation during adiabatic expansion, and therefore the volume occupied by the fluid at the end of the expansion will be less than v_2.

To find this actual volume we make use of formula (2), from which we derive the following values:

	Steam.	Alcohol.	Carb. Dis.	Chloroform.	Ammonia.
$u_2 =$	8.80	5.39	3.26	4.80	0.26

Comparing these results with the preceding shows that the condensation of vapor due only to expansion will be, in per cent. of the original mass,

Steam.	Alcohol.	Carb. Dis.	Chloroform.	Ammonia.
$18\frac{3}{10}$	$2\frac{3}{10}$	13	15	9

This does not include the initial condensation, nor, indeed, any effect, whether of condensation or of reëvaporation due to the transmission of heat through the walls of the cylinder.

It was not observed by the computers until after the work was completed, that between the limits of temperature used in this case the specific heat of the saturated vapor of ammonia is positive, and hence for these limits it should be classed with "ether-like" vapors. I have shown that the specific heat of the saturated vapor of ammonia is positive for temperatures above 95° F. (*Thermodynamics*, p. 335). The discussion of the *ether engine*, further on, will show how ammonia vapor should have been treated.

The total heat of a liquid at T° is the number of B.T.U. necessary to raise the temperature of 1 pound of the liquid from that of melting ice to T°. This total heat has been determined by Regnault at atmospheric pressure, and is given in B.U. in equation (4), p. 407, Wood's *Thermodynamics*, as follows:

$$q = a + bt + ct^2 + dt^3 \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

The specific heat at any temperature will be the differential coefficient of the preceding equation.

$$\therefore K = \frac{dq}{dt} = b + 2ct + 3dt^2 \quad . \quad . \quad . \quad . \quad . \quad (19)$$

If t is the temperature in degrees F., the constants will have the following values:

	b	c	d
Steam	0.99957333	0.000002222	0.0000000926
Alcohol	0.50954300	0.00056407	0.000000617284
Carbon Disulphide	0.2323140	0.00004555	0.0000000000
Chloroform	0.2305470	0.00002817	0.00000000
Ammonia*	....	....	....

Substituting the corresponding values in equation (19) we get for the mean value of K between τ_1 and τ_2,

	Steam.	Alcohol.	Carb. Dis.	Chloroform.	Ammonia.
$K =$	1	0.954	0.257	0.245	1†

Having found the values of the quantities entering the equation of works we are now ready to substitute them in (6) and (7), and have

	W_E	W_C
Steam	50323	15932
Alcohol	61043	26652
Carbon Disulphide	64538	30147
Chloroform	71843	37452
Ammonia	72222	37831

* The constants for ammonia are not known.

† It is accurate enough to consider the specific heat of liquid ammonia as constant and equal to 1. Since this was written the specific heat of liquid ammonia has been found to be 1.229. (See writer's second paper on *Some Properties of Ammonia*, in this volume.)

The difference of these two works is the effective work, and is given by equation (8),

$$\therefore \text{ Effective work} = 34391, \quad . \quad . \quad . \quad . \quad (20)$$

which is the same for all the fluids, since the same quantity of heat is worked between the same limits of temperature.

From (20) and (15) we get

$$E = 18.4 \text{ per cent.}, \quad . \quad . \quad . \quad . \quad . \quad (21)$$

which is evidently the same for all the fluids.

We will now consider *ether*, the specific heat of whose saturated vapor is *positive.*

As we have supposed that the fluid is all vapor at the point of cut-off, this vapor, when ether, will superheat during adiabatic expansion, and the equation to the curve of expansion will be of the form,

$$\frac{\tau}{\tau_2} = \left(\frac{v_2}{v}\right)^{\lambda}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

where $\lambda = \dfrac{a}{K_v}$, a being obtained from an equation for saturated vapors of the form

$$pv = a\tau - \frac{b}{v^n}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

the constants a and b being determined by experiment. (*Transactions*, Vol. X., p. 673.)

As these constants for ether are not known, we have used formula (3″) which is accurate enough for our purpose.

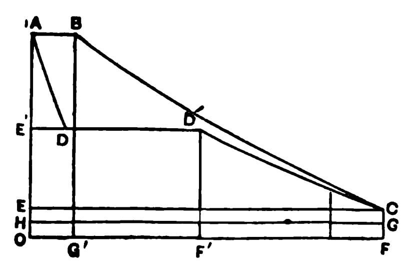


FIG. 26.

The annexed figure (Fig. 26) gives the path of ether, when worked in Carnot's cycle. BC shows the adiabatic of superheated vapor, CD' the isothermal of superheated vapor, $D'D$ the isothermal of saturated vapor, and DA the path of adiabatic compression.

The total work done during expansion is $ABG'O + BCFG'$; and that done during compression is $CFF'D' + D'F'OE' + E'DA$.

Let $F'D' = p_4$, $E'D' = u_4$, and $E'D = u_5$;

then (using the same notation as before for the other quantities) we have

$$W_E = p_1 u_1 + \int_{u_1}^{u_2} p du$$

$$= p_1 u_1 \left[1 + \frac{1}{\gamma - 1}\left(\frac{\tau_1 - \tau_2}{\tau_1}\right)\right] \quad . \quad . \quad . \quad . \quad (24)$$

$$W_C = \int_{u_4}^{u_2} p du + p_4 u_4 + \int_{p_4}^{p_1} u dp$$

$$= p_1 u_1 \frac{\tau_2}{\tau_1} \log \frac{u_2}{u_4} + p_4 u_4 + JKw_1 u_1 \left[\tau_1 - \tau_2\left(1 + \log_e \frac{\tau_1}{\tau_2}\right)\right] \quad . \quad (25)$$

For adiabatics of perfect gases we have

$$p = \frac{p_1 u_1^{\gamma}}{u^{\gamma}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

$$\left(\frac{u_1}{u}\right)^{\gamma - 1} = \frac{\tau}{\tau_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

To find the values of the quantities entering these equations we have proceeded in the following manner:

Eq. (11) gives $p_1 = 55350$, $p_4 = 10850$.
" (12) " $L_1 = 391782$, $L_2 = 97642$.
" (13) " $h_{e_1} = 99.49$, $h_{e_2} = 143.98$.
$H_{e_1} = 77341$, $H_{e_2} = 112032$.
" (14) " $w_1 = 5.065$, $w_2 = 0.9$.
" (16) " $u_1 = 0.476$.
" (19) " $K = 0.606$.
" (26) " $p_2 = 1440$.
" (27) " $u_2 = 14.938$.

$$\gamma = \frac{k_p}{k_v} = \frac{.4797}{.453} = 1.0588 \quad . \quad . \quad . \quad . \quad . \quad (28)$$

Substituting the above values in equations (24) and (25), we get

$W_E = 115000$, $W_C = 80609$.

Therefore,

$$\text{Effective work} = W_E - W_C = 34391,$$

which is the same as that before found, equation (20).

The heat used being the same for the six fluids under consideration, the efficiency will be the same as before, *i.e.*,

$$E = 18.4 \text{ per cent.}$$

This case presents many points of interest. We have not been able, by means of equations (24), (25), and (9), to deduce the last member of equation (10). The final numerical result being 18.4, as in the other cases, shows that the constants, K and γ, involving the specific heats of the liquid and of the gas, must have been very accurately determined.

Assuming the equation

$$\frac{p_1u_1\left[1 + \frac{1}{\gamma - 1}\cdot\frac{\tau_1 - \tau_2}{\tau_1} - \frac{\tau_2}{\tau_1}\log\frac{u_2}{u_1}\right] - p_4u_4 - JKw_1u_1\left[\tau_1 - \tau_2\left(1 + \log_e\frac{\tau_1}{\tau_2}\right)\right]}{L_1u_1}$$

$$= \frac{\tau_1 - \tau_2}{\tau_1},$$

the reduction gives a peculiar and complex relation between the three specific heats—the specific heat of the liquid, K, and the ratio of the specific heat of the vapor at constant pressure to that at constant volume.

To illustrate this case more thoroughly, we will plot an indicator card for each fluid.

In order to do this we must determine at least two more points on the expansion curve, and the point where adiabatic compression begins. In the case of ether we must also determine the point on the path of isothermal compression where the vapor passes from the superheated state to that of saturation.

We have taken two intermediate temperatures between T_1 and T_2, which we call T' and T''. Take

$$T' = 300°, \quad T'' = 250°; \quad \therefore \tau' = 760.66°, \quad \tau'' = 710.66°.$$

The pressures, p' and p'', corresponding to these temperatures, are obtained by means of equations (11) and (26), and are given in Table I.

The abscissas of these points, u' and u'', are determined by means of equations (2) and (27).

The point D (Figs. 25 and 26), or u_5, where adiabatic compression begins, is found from equation (3).

To obtain the point D' (Fig. 26) for ether, combining (11) and the equation of the isothermal, $pu = p_2u_2$, we have

$$u_4 = 1.982.$$

Having these points, we can draw approximately the indicator diagrams for each fluid, thus showing the work done by each of the vapors in question.

Table I. contains all the given and computed quantities of the case, and Fig. 27 gives the ideal indicator diagrams.

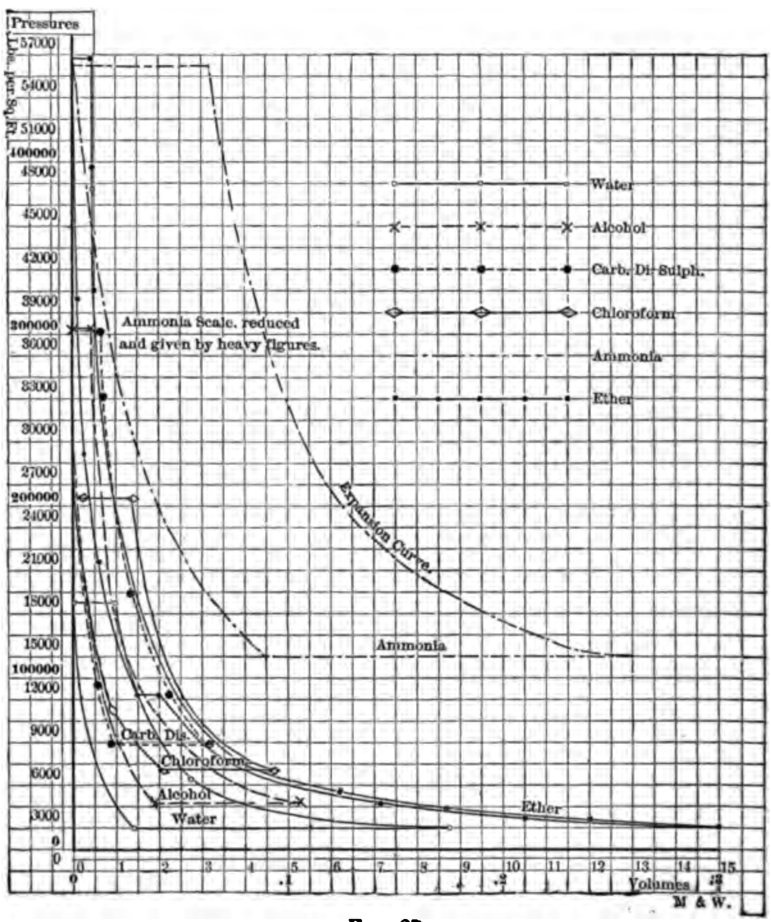


FIG. 27.

TABLE I.

VAPS.	T_1	T_2	T'	T''	τ_1	τ_2	τ'	τ''	p_1	p_2	p'	p''	L_1	L_2	L'	L''	h_{c_1}	h_{c_2}	h_c'	h_c''	H_{c_1}	H_{c_2}	H_c'	H_c''
H_2O....	341.6	194	300	250	802.26	654.56	760.66	710.66	17408	1469	9683	4299	186905	20553	110269	53526	873.59	978.44	903.97	931.11	679653	761226	703291	730625
C_2H_6O..	341.6	194	300	250	802 26	654.56	760.66	710.66	36450	3279	20494	9292	381359	43791	228397	112392	288.19	370.73	308.55	335 97	224216	288428	240050	261384
CS_2 ...	341.6	194	300	250	802.26	654.56	760.66	710.66	36745	7269	27880	14639	259844	64970	209602	118978	108.07	140.34	118.37	129.51	84077	109188	92096	100757
$CHCl_3$..	341.6	194	300	250	802.26	654.56	760.66	710.66	24871	5432	18100	11189	136980	53358	108696	86051	58.92	94.94	70.32	82.72	45843	73867	54709	64360
NH_3 ...	341.6	194	300	250	802.26	654.56	760.66	710.66	468700	113100	331900	207539	2954108	873581	2206139	1476268	320.54	428.34	351.89	388.56	249383	333246	273770	302301
$C_4H_{10}O$	341.6	194	300	250	802.26	654.56	760.66	710.66	55350	1440 p_4 10850	21406	6358	391782	97642			99.49	143.98			77341	112032		

VAPS.	w_1	w_2	w'	w''	v_1	v_2	v'	v''	u_1	u_2	u'	u''	K	H	W_E	W_C	v_5	$EFFW$	$E\%$
H_2O.....	.275	.027	.157	.073	1	10.185	1.75	3.767	1	8.80	1.671	3.138	1	186905	50323	15982	1.38	34391	18.4
C_2H_6O...	1.701	.151	.951	.430	.49	5.519	.880	1.938	.490	5.89	.880	1.526	.954	186905	61043	26652	1.89	34391	18.4
CS_2......	3.091	.595	2.275	1.181	.719	3.734	.977	1.881	.719	3.26	.938	1.421	.257	186905	64538	30147	.91	34391	18.4
$CHCl_3$..	2.978	.722	1.986	1.337	1.365	5.647	2.053	3.049	1.365	4.80	1.918	2.000	.845	186905	71848	37455	1.94	34391	18.4
NH_3.....	11.842	2.621	8.058	4.883	.068	.285	.092	.153	.068	.26	.087	.115	1	186905	72222	37831	.088	34391	18.4
$C_4H_{10}O$..	5.065	.9			.476	14.998	1.173	3.716	.476	14.988	1.173	3.716	.606	186905	115000	80609	1.55 v_4 1.982	34391	18.4

CASE II.

Case I. is never realized, even approximately, in practice, and in Case II. we take a step toward actual conditions, by assuming that the cylinder communicates with a condenser, that the exhaust is not opened until the end of the stroke, and that it remains open throughout the back stroke, in which case there will be no compression. We assume the same initial temperature as in Case I., and let the temperature of the condenser be

$$104^\circ F. = T_3; \qquad \therefore \tau_3 = 564.66.$$

The pressures of the saturated vapors corresponding to this temperature are given by equation (21), and the results are given in Table II.

From Fig. 25 we get

$$W = ABCE + ECGH$$

$$= \int_{p_2}^{p_1} udp + u_2(p_2 - p_3)$$

$$= JKw_1u_1 \left\{ \tau_1 - \tau_2 \left(1 + \log_e \frac{\tau_1}{\tau_2}\right) \right\}$$

$$+ L_1u_1 \frac{\tau_1 - \tau_2}{\tau_1} + u_2(p_2 - p_3). \quad . \quad . \quad . \quad . \quad . \qquad (29)$$

This equation is true for all fluids whose specific heat is negative.

For vapors whose specific heat is positive, like ether, we have (Fig. 26),

$$W = ABG'O + BCFG' - HGFO. \quad . \quad . \quad . \quad . \qquad (30)$$

The sum of the first and second terms of (30) is equal to the second member of equation (24); and $HGFO = u_2p_3$;

$$\therefore W = p_1u_1 \left[1 + \frac{1}{\gamma - 1}\left(\frac{\tau_1 - \tau_2}{\tau_1}\right)\right] - u_2p_3 \quad . \quad . \quad . \qquad (31)$$

The heat used for any of the fluids in question will be obtained from the formula

$$H = JKw_1u_1(T_1 - T_3) + L_1u_1. \quad . \quad . \quad . \quad . \qquad (32)$$

To make the comparison more complete, we will find the relative sizes of the cylinders to produce the same power.

Let V be the volume of the cylinder in cubic feet, $W' =$ the work done by steam, $u'_2 =$ the volume of the cylinder required to produce the work W', W and u_2 the corresponding quantities for any other fluid; then we have

$$V = \frac{W'}{W} u_2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

and if we call the volume of the cylinder in the case of water unity, and M the relative size of the other cylinders, we obtain

$$M = \frac{W'}{W} \cdot \frac{u_2}{u'_2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

Fig. 28 contains the ideal indicator diagrams. It will be observed that the ether is expanded to a pressure lower than that of the condenser.

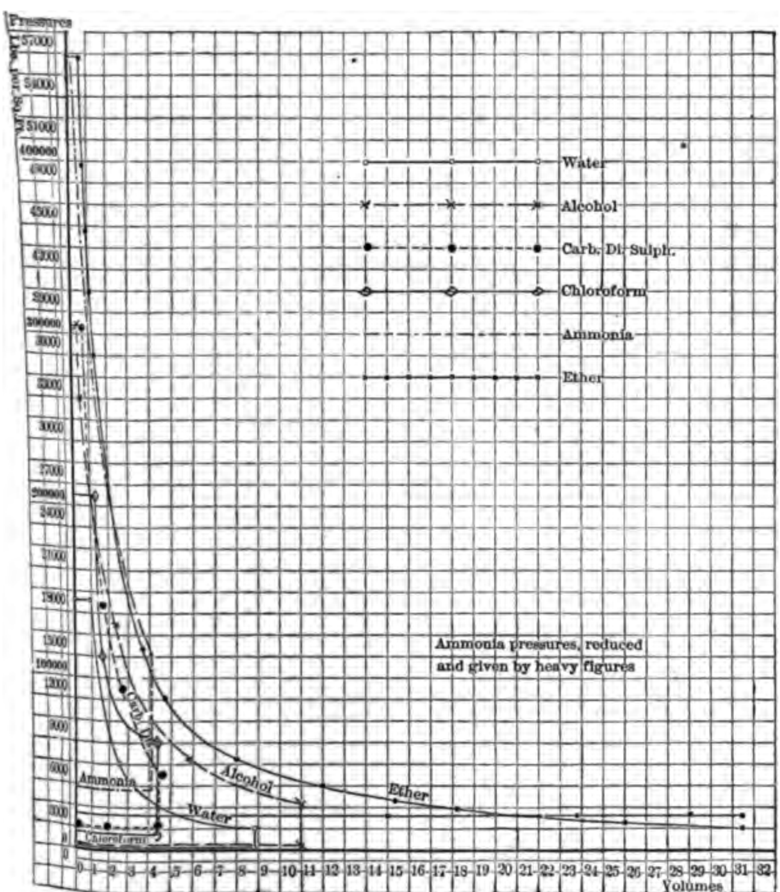


FIG. 28.

TABLE II.

VAP.	T_1	T_2	T	T'	τ_1	τ_2	τ'	τ''	p_1	p_2	p'
H_2O	341.6	194	300	250	802.26	654.66	760.66	710.66	17408	1469	9683
C_2H_6O. ...	341.6	194	300	250	802.26	654.66	760.66	710 66	36450	3279	20494
CS_2	341.6	194	300	250	802.26	654.66	760.66	710.66	36745	7269	27880
$CHCl_3$	341.6	194	300	250	802.26	654.66	760.63	710.66	24871	5432	18100
NH_3.. ..	341.6	194	300	250	802.26	654.66	760.66	710.66	468700	113100	331900
$C_4H_{10}O$ ...	341.6	194	300	250	802.26	654.66	760.66	710.66	55350	1440	21406

VAP.	p''	L_1	L_2	L'	L''	h_{e_1}	h_{e_2}	h'_e	h''_e	H_{e_1}	H_{e_2}	H'_e	H''_e
H_2O	4299	186905	20553	110269	58526	878.59	978.44	903.97	931.11	679653	761226	703291	730625
C_2H_6O. ...	9292	381359	48791	228397	112392	288.19	370.73	308.55	335.97	224216	288128	240050	261384
CS_2	14039	259844	64970	209602	118978	108.07	140.34	118.37	129.51	84077	109188	92096	100757
$CHCl_3$	11189	136930	58358	108696	86051	58.92	94.94	70.32	872.25	45843	73867	54709	64360
NH_3.. ..	207539	2954108	873561	2206139	1476268	320.54	428.34	351.89	388.56	249383	333246	273770	302301
$C_4H_{10}O$ ...	6358	391782	97642	......	....	99.49	143.98	.....	......	77341	112032	......	.. ...

VAP.	w_1	w_2	w'	w''	r_1	r_2	v'	v''	u_1	u_2	u'
H_2O..............	.275	.027	.157	.073	1	10.185	1.75	3.767	1	8.80	1.671
C_2H_6O.	1.701	.151	.951	.430	1	11.265	1.788	3.955	1	11.012	1.842
CS_2..	3.091	.595	2.275	181		5.195	1.358	2.617	1	4.592	1.294
$CHCl_3$	2.978	.722	1.986	.337		4.137	1.504	2.234	1	4.511	1.263
NH_3................	11.842	2.621	8.058	.883		4.518	1.469	2.425	1	4.144	1.433
$C_4H_{10}O$	5.065	.9	......	....		31.38	2.465	7.807	1	31.38	2.465

VAP.	u''	M	T_3	τ_3	p_3	K	W	H	E
H_2O..............	3.138	1	104	564.66	152.3	1	49083.3	237739	20.64%
C_2H_6O.	3.698	.509	104	564.66	374.7	.954	120479.5	549071	21.94%
CS_2..	2.003	.308	104	564.66	1717	.257	81916.9	406655	20.14%
$CHCl_3$	1.413	.467	104	564.66	920	.245	53807.3	272201	19.40%
NH_3................	1.828	.024	104	564.66	33020	1	966385.3	5142235	18.79%
$C_4H_{10}O$	7.807	1.179	104	564.66	2580	.606	148576.0	959166	15.5%

CASE III.

Thus far the same limits of temperature have been assigned. In this case we assign the same limits of *pressure* for all the fluids between which they are *expanded*, and the *temperature* of the condenser is assumed at 104° F. as before. This temperature being less than the final temperatures in the case of sulphur dioxide and ammonia, gives peculiar results, more especially in the case of the latter, in which the pressure of the boiler is less than hat of the condenser, as shown and explained further on. We assume pressures between which steam is often worked in ordinary practice. The pressures assumed are:

$p = 17408$ lbs. per square foot $= 121.6$ lbs. per square inch (nearly).

$p_1 = 1469$ lbs. per square foot $= 10.2$ lbs. per square inch.

The remaining values are to be found from the formulas already established in the preceding cases, and are given in Table III.

A plate of diagrams illustrating this case is also given at the end of this case (Fig. 29).

The results found for ammonia in this case are unpractical and differ greatly from the other values, this difference being interpreted in the following manner:

The pressure of the condenser being greater than that of the boiler, will necessitate the use of another engine driven by the condenser to work the one under consideration, and the efficiency 165% which we have obtained is that of the outside engine or condenser, being one and sixty-five hundredths more efficient than our considered engine. The first case of ammonia engine considered is virtually a compressor. We have added a second case of the ammonia engine, in which the inferior limit of pressure is $p_1 = 10374$ and the corresponding temperature is 39.34° F., and also have taken the temperature of the condenser at the inferior limit of the temperature of expansion, 39.34° F. Although this temperature is too low to be realized in practice, except by artificial means, yet it furnishes a better means of comparison than the other hypothetical cases of this fluid.

Attention is called to the relative sizes of the cylinders in working the several fluids as shown in the column marked *M*. It appears that, excluding ammonia, of all the vapors considered the vapor of water requires the smallest cylinder, and the efficiency of this fluid is also greatest, as shown in column marked *E*.

TABLE III.

VAP.	T_1	T_2	T'	T''	T_3	τ_1	τ_2	τ'	τ''	τ_3	p_1	p_2	p'	p''	p_3	L_1	L_2	L'	L''	h_{e_1}	h_{e_2}	h'_e	h''_e	H_{e_1}
H_2O ……	341.60	194.00	300	250	104	802.23	654.66	760.66	710.66	564.66	17408	1469	9683	4299	152	186905	20552	110269	53526	873.59	978.4	903.97	939.11	679653
C_2H_6O …	286.34	158.34	250	200	104	747.00	619.00	710.66	660.66	564.66	17408	1469	9292	3701	375	199649	20992	112247	48816	280.03	385.1	335.97	366.12	217809
CS_2 ……	275.34	94.34	230	150	104	736.00	565.00	690.66	610.66	564.66	17408	1469	11580	3809	1717	135730	15586	97156	36870	124.40	159.9	143.82	147.75	96783
$CHCl_3$ ……	295.34	125.34	250	200	104	756.00	586.00	710.66	660.66	564.66	17408	1469	11189	5926	920	114109	18786	86051	56593	74.90	107.5	82.72	93.72	55272
NH_3 ……	66.34	−41.66	40	10	104	527.00	419.00	500.66	470.66	564.66	17408	1469	10375	5602	33020	166700	17726	104709	60180	513.89	580.6	530.63	549.34	399806
	66.34	39.34	……	……	39.34	527.00	500.00	……	……	500.00	17408	10406	……	……	10406	166700	10648	……	……	513.89	531.0	……	……	399806
$C_4H_{10}O$ …	231.34	141.34	200	175	104	692.00	602.00	660.66	635.66	564.66	17408	1469	7730	3881	2530	146542	……	……	……	134.68	……	……	……	104781

VAP.	H_{e_2}	H'_e	H''_e	w_1	w_2	w'	w''	v_1	v_2	v'	v''	u_1	u_2	u'	u''	K	M	W	H	E
H_2O ……	761296	703291	730652	0.275	0.027	0.157	0.073	1	10.185	1.75	3.77	1	8.80	1.67	3.14	1.000	1.000	49084	237789	20.64%
C_2H_6O ……	332601	261384	284841	0.916	0.063	0.430	0.171	1	14.539	2.14	5.36	1	12.24	1.77	4.17	0.859	1.193	57222	311249	18.38%
CS_2 ……	124410	111892	114949	1.402	0.125	0.868	0.333	1	11.216	1.62	4.21	1	9.32	1.43	3.75	0.249	1.484	35109	182255	19.27%
$CHCl_3$ ……	83643	64360	72914	2.006	0.224	1.837	0.776	1	8.955	1.50	2.57	1	7.31	1.46	1.98	0.242	1.135	37617	186359	20.18%
NH_3 ……	451722	412830	427387	0.417	0.039	0.253	0.141	1	10.692	1.65	2.96	1	9.29	1.59	2.75	1.000	−0.204	−254827	−154488	165%
	413149	……	……	0.417	0.258	……	……	1	1.648	……	……	1	1.592	……	……	1.000	1.008	8579	306812	2.87%
$C_4H_{10}O$ ……	190528	……	……	1.398	……	……	……	1	10.42	2.19	4.24	1	10.42	2.19	4.24	0.579	4.877	29108	226734	12.84%

The effect of condensation would modify the efficiencies, also compression to fill the clearance space in actual engines would modify the efficiency a small amount; still, considering the size of

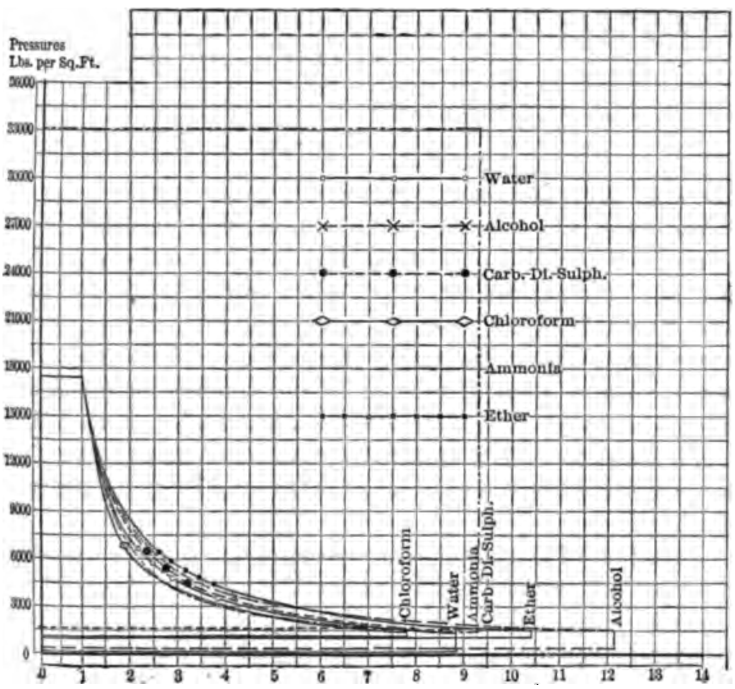


FIG. 29.

the engine, the cost of construction, the cost of the fluid, and the facility and safety in handling, it is safe to conclude that *water is by far the best medium of the substances considered for transforming heat into power.*

CCCCXXI.*

HEAT TRANSMISSION THROUGH CAST-IRON PLATES PICKLED IN NITRIC ACID.

BY R. C. CARPENTER, ITHACA, N. Y.

(Member of the Society.)

DURING the past season the writer made some experiments to ascertain the relative heat transmission through cast-iron plates in the condition left by the foundry, as compared with the same plates treated with dilute nitric acid for periods respectively of 9, 18, and 40 days, the acid baths not being strengthened during the period of immersion of the plates. The results show a marked change in the conducting power of the plates, due to prolonged treatment with dilute nitric acid, and would indicate that a decidedly beneficial effect might be expected by treating the castings of steam-engine cylinders in such a bath.

The action of the nitric acid, by dissolving the free iron and not attacking the carbon, forms a protecting surface to the iron, which is largely composed of carbon; in the plates actually treated, one surface was converted into a black enamel, and the other covered with a large amount of rust, which difference, as yet, has not been accounted for. The investigation was made at the request of Dr. R. H. Thurston, and it is proposed to continue it under various conditions and with improved facilities.

The effect of covering cast-iron surfaces with varnish had been previously investigated, under the direction of Dr. Thurston, by Mr. P. M. Chamberlain. He subjected the plate to the action of strong acid for a few hours, and then applied a non-conducting varnish. By simple treatment with acids, under such conditions, he found no especial difference in the conductivity of the plates, but the same plates subsequently oiled with a drying oil and then varnished, showed a remarkable decrease in conducting power. In the experiment by Mr. Chamberlain, one side of the plate was polished, the other rough lathe finished. The following table exhibits his results, one surface only treated:

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

No. of Test.	Total, Heat Units Transmitted.	Heat Units per sq. foot per hour, for each degree, $\tau - \tau'$.	
1	5550.9	170	As finished—greasy.
2	5565.4	169	" " "
3	5444.8	165	" " washed clean with benzine and dried.
4	5139.3	159	" " " " " " " "
5	5055.8	152	" " " " " " " "
6	5523.9	169	Oiled with lubricating oil.
7	5444.8	168	" " " "
8	5620.4	173	Washed clean with benzine.
9	5515.6	171	" " " "
10	5101.9	162	After exposure to nitric acid sixteen hours, then oiled, linseed oil.
11	5489.7	170	" " " " " " " " " " "
12	5581.8	171	" " " " " " " " " " "
13	5347.5	166	" " " hydrochloric acid 12 " " " " "
14	5741.1	174	" " " " " " " " " " "
15	3920.4	118	" " " sulphuric acid 1, water 2, for 48 hours, then oiled,
16	3104.9	117	varnished, and allowed to dry for 24 hours.

NOTE.—The last column was computed from Chamberlain's data. It shows about 50% greater conductivity than obtained by the writer with cast-iron with the foundry skin not removed. From the different methods of investigation the results are probably not directly comparable.

In above tests, $\tau - \tau'$ equals about 235°.

In making the following experiments, a series of cast-iron plates was obtained, each cast at the same heat and from the same pattern, each measuring 8.4 inches by 5.4 inches by .45 inch thick.

A bottle of concentrated nitric acid was obtained and two solutions in rain water were made—the first containing 1% of nitric acid by measurement, the second 5% of nitric acid. In each of these solutions three of the cast-iron plates were placed. The plates were immersed in a horizontal position, and separated from each other by small blocks at the corners, so as to permit a free action of the acid. The solution was shaken and left for some time in an inclined position to permit any air beneath the plates to escape.

The plates were left in these solutions, respectively, 9 days, 18 days, and 40 days, at which time a test of the relative conducting power was made. A test was also made of the relative conducting power of a plate of cast-iron of the same dimensions and not treated in any way, and also of a plate of pine wood of same dimensions as the cast-iron plates.

The following method of testing the relative conducting power of the plates was adopted, after several trials, and proved quite satisfactory. A box, shown in section in Fig. 31, was made of boards an inch thick, with internal dimensions 8.5 by 5.5 inches and 6.5 inches deep. Near the centre of the box and extending completely around on the inside, a strip of wood $\frac{5}{8}$ by $\frac{1}{2}$ inch was rabbeted in place, as shown in the sketch. The top of this strip, marked *D* in Fig. 31, was made accurately parallel to the top of the box, and neatly fitted in place. A gasket of rubber packing was fitted to the projecting top of this strip. Two frames, made of hard wood, with a horizontal piece about one inch square, were securely fastened to the box, as shown in Figs. 31 and 32. From the horizontal piece, *G*, of each frame, two braces, *HH*, were set with the lower ends resting on the plates to be tested. These braces were cut of such a length that by forcing them into nearly a vertical position, a strong pressure was made to act on the corners of the plate to be tested, and no difficulty was found in producing a steam-tight joint. The plate to be tested was intro-

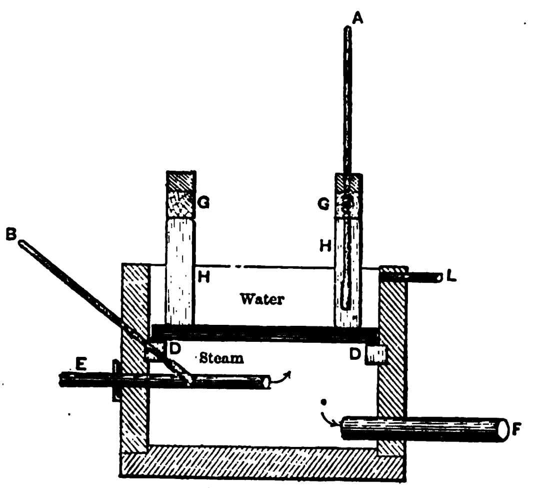


FIG. 31.

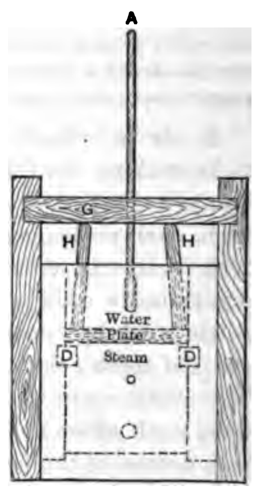


FIG. 32.

duced in the box as shown in the sketch, thus forming a horizontal partition, *DD*. Into the portion of the box below the plate a one-quarter-inch pipe, *E*, was introduced, terminating near the centre of the box; this pipe was connected by a rubber hose to a steam supply; on the opposite side and near the bottom of the box was inserted a one-half-inch nipple, *F*, for the discharge of the condensed steam. A thermometer, *BD*, was inserted in the steam chamber as shown in the sketch, and maintained in the same position during all the tests. Water was put in the box above the plate and the heat transmitted through the plate was measured by the increase in temperature of this water. To measure the temperature of the water, a thermometer, shown at *A* in sketch, was held in position by a cross-piece of one of the frames, the thermometer being kept with its bulb immersed and in the same position throughout all the tests by maintaining a mark on the stem opposite the lower edge of the cross-piece.

In making a test, the plate was first put in position, the box levelled, and steam turned on in the pipe *E*, with pipe *F* closed, until it was ascertained that there were no steam leaks around the edge of the plate. Steam was then turned off in pipe *E*, pipe *F* opened, the water was added in the chamber above the plate until it passed off in the overflow pipe *L*. Steam was then admitted through the pipe *E*, and as soon as the water above the plate reached a temperature of about 70°, readings of both thermometers were taken, and continued each minute until the water attained a temperature of about 150°. The temperature in the steam chamber could not be maintained quite uniform, as shown by the readings of the thermometer *B*. There was also a slight variation in the degree of temperature at the beginning and ends of the different tests, but not enough to make any material difference in the results attained. The process used being exactly duplicated for each plate tested, gives the comparative transmission of heat for each plate, but does not give with exactness the number of thermal units transmitted, because of our ignorance of the currents existing in the water chamber. Assuming no horizontal currents to exist, which is probably true, the average number of thermal units transmitted per inch of surface is equal to the weight of the water, 3.125 lbs., divided by the exposed surface of the plate, 38.7 square inches, multiplied by the rise in temperature; or .0807, multiplied by the rise in temperature of the water.

This can be reduced to thermal units transmitted per square foot per hour as follows, neglecting the coefficient of the internal thermal resistance.

Letting

K represent the number of thermal units transmitted per hour for each degree that temperature of steam chamber exceeds that of the water chamber,

τ represent the average temperature of the steam chamber,

τ' represent the average temperature of the water receiving heat,

a represent the effective area of the plate in square feet = .27 sq. ft.,

w = weight in pounds of the water heated = 3.125 lbs.,

G = average gain of temperature of w. per minute,
g = average gain of temperature of w. per hour = 60 w.

Then $$(\tau - \tau')\, aK = gw = 60\, Gw$$

$$K = \frac{60\, Gw}{(\tau - \tau')\, a} = 694.4\, \frac{G}{\tau - \tau'} = \frac{694.4}{\frac{\tau - \tau'}{G}}.$$

Radiation into the atmosphere from the surface of the water exposed can be neglected, as time of heating was very short.

In the proposed investigations of other conditions, relating to the transmission of heat, the temperatures in the steam chamber and water chamber will be kept constant, and the heat units measured by the weight of water heated in a given time.

In Mr. Chamberlain's investigation no change in the conducting power was noticed until the plates were varnished, when the conducting power was reduced 33%. In the investigation made by the writer the loss of conducting power of the plates immersed in the 1% solution of the acid varied directly with the time of immersion up to eighteen days, after which it remained nearly constant. This is best shown in the diagram appended (Fig. 33). Those plates immersed in the 5% solution showed a greater loss of conductivity for a short time of immersion, but at the end of forty days the result from either solution was substantially the same.

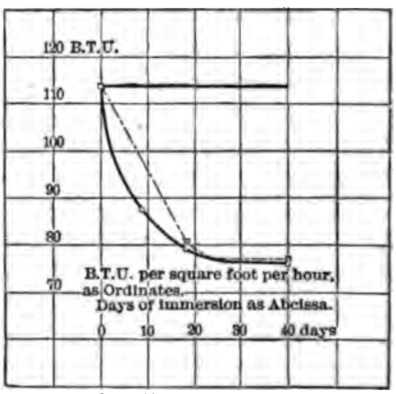


FIG. 33.

These changes are clearly shown in the diagram following the article, and in the following summary of results, as well as in the appended tables.

SUMMARY OF RESULTS.

Character of Plates, each plate 8.4 inches by 5.4 inches, exposed surface .27 square foot.	Increase in temperature of 3.125 lbs. of water each minute.	Proportionate thermal units transmitted for each degree of difference of temperature per square foot per hour.	Relative transmission of heat.
Cast iron—untreated skin on, but clean, free from rust	13.90	113.2	100.00
Cast iron—nitric acid, 1% sol., 9 days	11.5	97.7	86.3
" " 1% sol., 18 days	9.7	80.08	70.7
" " 1% sol., 40 days	9.6	77.8	68.7
" " 5% sol., 9 days	9.98	87.0	76.8
" " 5% sol., 18 days	....	80.0*	70.0*
" " 5% sol., 40 days	10.6	77.4	68.5
Plate pine wood same dimensions as cast iron	0.83	1.9	1.6

* Approximately.

AVERAGE RESULTS OF TRANSMISSION TRIALS OF HEAT THROUGH CAST-IRON PLATES.

Plates .45 thick.

No. of Test.	Gain in Temperature per minute.	τ' Average Temperature, Water.	$\tau - \tau'$ Average difference temperature, Steam chamber and Water.	Relative number of Thermal units for each degree $\tau - \tau'$ per square foot per hour.	
1	12.6	115.5	91.8	107.5	Clean cast-iron plate.
2	15.0	120.2	88.2	114.8	" " "
3	13.6	123.6	84.4	113.4	" " "
4	14.7	128.4	86.4	118.5	" " "
5	14.1	120.5	88.8	110.0	" " "
6	13.7	126.0	83.2	115.2	" " "
Average	......	....	....	113.2	
1	11.6	127.5	81.5	97.6	Plate pickled 9 days in 1% solution nitric acid.
2	10.4	128.2	77.9	91.8	" " " " " " "
3	12.6	127.0	81.1	103.9	" " " " " " "
Average	......	....	....	97.7	
1	9.8	118.7	88.7	79.24	Plate pickled 18 days in 1% solution nitric acid.
2	10.2	123.3	83.6	83.00	
3	9.7	118.9	86.0	78.00	
Average	9.7	....	....	80.08	
1	9.4	119.2	88.4	73.5	Plate pickled 40 days in 1% solution nitric acid.
2	9.4	128.2	80.8	80.3	" " " " " " "
3	10.	121.9	87.1	79.7	" " " " " " "
Average	9.6	....	....	77.8	" " " " " " "
1	9.7	128.7	80.3	88.17	Plate pickled 9 days in 5% solution nitric acid.
2	9.4	130.0	74.6	87.5	
3	10.7	121.	86.5	85.2	
Average	9.98	....	....	87.0	
1	......	....	....	....	Plate pickled 18 days in 5% solution nitric acid. These results were unfortunately lost, but did not differ much from 80.0 thermal units.
2	......	....	....	....	
3	......	....	....	....	
Average	......	....	....	....	
1	10.9	117.9	92.6	80.3	Plate pickled in 5% solution nitric acid 40 days.
2	11.2	119.2	90.8	75.7	
3	9.8	120.5	87.7	76.3	
Average	10.6	....	....	77.4	

DISCUSSION.

Prof. Jas. E. Denton.—There is a remark in the paper which indicates that it is believed that these tests, showing differences in conduction, apply to the phenomenon of cylinder condensation in steam-engines. It strikes me as peculiar that they can be conducted as they are, in that case. What is reported is a test for the conduction of heat through a plate like a surface condenser. Now we know that in steam-engines, if it is true that the walls of the cylinder at the cut-off are responsible for the cylinder condensation, it follows that the rate of transmission of heat to and from a square foot of surface, which the heat is supposed to enter but not to traverse, is many times the greatest possible amount we can send through metal in surface condensers. I do not see, therefore, how any measure of the effect of the acid regarding cylinder condensation can be made by measuring the steady flow of heat clear through the metal. Again, would not the removal of the skin of the coating by boring destroy any influence due to pickling?

Referring to previous attempts to reduce cylinder condensation, I am reminded of that engineer who noticed that bismuth was a metal which had low specific heat, low conducting power, etc., and had, in fact, all the physical properties which would recommend it for putting on a cylinder head and piston head, and thereby reducing the cylinder condensation. He fitted up an engine to try it; putting the metal on the piston and cylinder heads. Bismuth has a melting point of about 400°, but it turned out that it had a plastic point of about 220°; consequently it came out with the exhaust after a few strokes. Then he dove-tailed it in, but still it melted out. I believe the Westinghouse firm went to some pains once to put porcelain on the cylinder head and piston, but no definite results were gotten from it. The difficulty is to keep the steam from getting under the plate. If it gets under the plate you have twice as many surfaces as before. This varnish will have to get into the pores pretty closely to prevent the steam getting there also.

Mr. Wm. Kent.—Prof. Carpenter's results may "indicate," as he says, that a beneficial effect might be expected by treating the castings of steam-engine cylinders in a dilute nitric-acid bath, but the experiments do not go far enough to make the indication a very strong one, and it is to be desired that the tests should be carried further under conditions which more nearly approximate those existing in steam-engine cylinders. I beg to offer the following remarks upon his method of test:

1. The untreated cast-iron plate with the skin on does not represent the condition of the inside of a cylinder, which first has the skin taken off, then is bored as smooth as possible, and finally is polished by the action of the piston under the conditions of high temperature and lubrication with oil and hot water.

2. If a cylinder, after being bored smooth, were treated with a pickling solution for forty days, and the carbon coating formed, it is not certain that the changed condition of the surface, or that portion or layer of the surface which has any important effect in varying the heat transmission, is any more than "skin deep," and that such layer will not be worn off, or at least changed again by the action of the piston to the condition of the surface of a cylinder not so treated.

3. The particular losses of heat in a steam-engine, to lessen which is apparently the object of the treatment, is not the heat lost by transmission through the walls of the cylinder (which may be lessened by felting the outside), but the loss by cylinder condensation, due to surface action inside; that is, to the heat given to the surface, or, probably, to a thin layer of the metal, during admission and abstracted therefrom during exhaust, the temperature varying with every reciprocation of the piston. These conditions are not represented by a cast-iron plate exposed on one side to steam of a constant temperature, and on the other to water of gradually increasing temperature.

I doubt if the conditions existing in the steam-engine can be imitated properly by an extemporized apparatus. Two actual steam-engine cylinders should be used, one treated and one untreated, and the test should be made under the conditions of actual practice, after both cylinders had been run for a considerable time so as to become polished as in actual service.

Prof. Carpenter's tests, however, are of great value in informing us that the rate of transmission of heat through a metal plate may be greatly changed by a simple treatment of its surface, and they may lead to important practical results in other directions than in treating steam-engine cylinders. Suppose, for instance, that boiler-plates and steam-pipes can be treated so as to diminish their heat conductivity, we might be able to save some of the felting needed for the outside of vertical tubular and locomotive boiler shells and for steam-pipes. Suppose, by some other treatment, we can increase the conductivity of plates and tubes, we can make a square foot of heating surface of a boiler, a feed-water heater, a surface condenser, or a steam-radiator more efficient than it is at present, and so reduce the area required to produce a given effect, and, hence, the cost of these appliances. The subject offers a most promising field for experimental research, and it is to be hoped that Prof. Carpenter and others will continue experiments upon it until more definite results are reached than we have at present.

Mr. W. M. McFarland.—It is, of course, understood that the nitric-acid treatment of cast-iron plates is part of Prof. Thurston's patent for preventing cylinder condensation, the remainder being the use of a gum or resin of some sort for filling the interstices left by the dissolution of the iron from among the carbon particles.

Owing to my position as one of the assistants to the Engineer-in-Chief of the Navy, I recently conducted a correspondence for him with Dr. Thurston in consequence of his request that the Bureau of Steam Engineering should make a test of his patent on an engine of sufficient size to make it more than a laboratory experiment.

Commodore Melville, who is always ready to encourage any device which promises economy or an increase of efficiency, looked into the matter very carefully with a desire to make the test on as large a scale as practicable. Unfortunately, nearly all our engines are building by contract, and it could hardly be expected that the contractors, who are working under the risk of heavy penalties both for time and for results, should be willing to undertake a thing which is still in the experimental stage. For similar reasons, it was impracticable to make the experiment with the engines which we are building ourselves at the New York Navy Yard, because we are, to a certain extent, competing with the contract work as to time and cost, and are handicapped by getting only eight hours' work while they get ten for the same wages.

However, we are building a small triple-expansion engine for a ferry-boat at the Portsmouth Navy Yard, with a low-pressure cylinder 16 inches diameter by 10 inches stroke, and Commodore Melville wrote to Dr. Thurston that he was willing to try the experiment on that engine under certain conditions. These were that Dr. Thurston should indicate how long the plates should be pickled, how deep the non-conducting layer should be, what gum or resin to use—in fact, give all necessary instructions for carrying out his patent; and then indicate what series of experiments he would want so far as steam-pressures and ranges of expansion were concerned. With these data, an effort would be made to carry out the experiment. I am free to say that I was very much interested in the whole scheme and anxious to see it carried out.

In his reply, Dr. Thurston unfortunately outlined such an elaborate series of experiments that it would practically have made it necessary to lay up the boat and devote the engine to these experiments. Our idea had been that the experiments would be something like this: After having the engine treated according to this process, we intended, as opportunity offered, to make tests lasting 10 or 12 hours each with various steam-pressures and grades of expansion, determining carefully the steam used per horse-power and the coal per horse-power, as well as the dryness of the steam. By comparison of these records with those of other engines of about the same size and working under nearly the same conditions, except that they had not been treated by the Thurston process, we would be able to find whether there was any marked gain. If it should turn out that there was a saving of, say, 25%, we would be justified in going ahead with elaborate experiments to determine the best way to apply the process. If, on the contrary, there was a gain of only 2 or 3% or a loss, then there would be nothing further to bother about.

Dr. Thurston's desire for an elaborate series of tests (though he has since expressed himself as satisfied with the ones we proposed) makes it uncertain whether the experiment will be carried out. I hope it will be.

I should be glad to hear an expression of opinion from other members of the Society as to the sufficiency of the series of tests I have outlined—not for determining the best way to apply the process, but whether its merits are such as to make it at all desirable.

*Mr. R. C. Carpenter.**—Taking the points brought out during the discussion in their order, as presented, it is easy to relieve the speakers of any misapprehensions and to supply all needed points additional to those already given.

* Author's Closure, under the Rules.

The first speaker in order does not understand how these facts bear on the subject of "cylinder condensation." That is a very simple matter, and I will explain. It is true, as he states, that this condensation occurs on the interior surface of the cylinder; that it may take place in immense quantity; and that its rate may greatly exceed that of condensation in the surface condenser, in which apparatus the heat must be carried through the walls of the condensing surfaces, instead, as in the engine, being simply stored in a superficial film of metal for the moment, and restored to the steam later. It is further true that the finishing of the cylinder, by boring or otherwise, would remove any such surface as might be produced by the process here under discussion. It is also true that, if "this varnish" does not "get into the pores pretty closely" it will probably have comparatively small effect.

As to the first point: Whatever process of treatment reduces the conductivity and the heat-storing power of the condensing surfaces of the engine-cylinder, must reduce correspondingly its wastes by cylinder condensation. The process described does precisely this thing. It reduces the heat-conducting power, as seen in these particular cases, some 40%. This is proved by the experiments described; for it is well known, and is so stated by all authorities, that the resistance to flow is measured, in such cases, mainly by the resistance of the surface, and not by the interior resistance of the metal. Rankine, for example, shows this. Any reduced flow, in this case, must therefore be due to reduced conductivity of the surface treated; since it is on that treated surface, in the engine, that absorption of heat would occur. It is fairly concluded that such application in the engine would proportionally reduce its wastes by cylinder condensation. Further, the heat-storing power is in part measured by the specific heat per unit of volume of metal absorbing heat from the steam. This is reduced by the solution of the metal, leaving the residue, which is principally oxide and graphite, as an apparently non-metallic mass, and thus the heat-storing capacity of the surface acting to produce waste is still further diminished. Both Chamberlain's experiments and those here described are amply sufficient to show that this presumption of proportional saving and increase in efficiency of steam may be anticipated.

As to the second point: Regarding its resistance to wear, all the objections raised apply to a method of treatment first proposed in this discussion. It is not practicable, nor do I understand that it is proposed, that this process should be applied in the treatment of rubbing surfaces. When the engineer is seeking to get his cylinders as hard as the tools will work them, he is not likely thus to proceed to convert their working surfaces into a comparatively soft, spongy mass. It is, however, well known that this waste occurs principally upon the parts which are not subject to abrasion; and, again, those parts which are under wear are so polished by that action, in well-cared-for engines, as to reduce largely the power of transmission of heat across their surfaces, thus making it less important that they should be treated in this way. It is only proposed thus to treat the heads, the sides of the piston, and the interior of the port-passages; which parts, in economically operated engines, certainly are most active in producing initial condensation. If we take Chamberlain's experiments as proving the possibility of reducing heat-storage through the action of the "varnish" permeating the pores of the metal by the amount of 30% or 40%, and the experiments of this paper as showing that an equal reduction may be effected by superficial solution alone, we may certainly take it as fairly to be anticipated that the gain in efficiency of working steam, in regular operation, in an engine which has thus been treated, is worthy of careful investigation. Such investigations are to be made by various interested persons, and we may expect, undoubtedly, to get their results in due time.

The remarks of the second speaker are based upon the misapprehension just adverted to: the supposition that it is necessary to treat the rubbing surfaces, the finished parts of the engine-cylinder. That error being removed, they call for no special reply. The nature and method of cylinder condensation is now so perfectly familiar to every engineer that it was not supposed that it would be assumed that the writer of the paper could stray so far from familiar fact as seems to have been taken for granted in this case, or a paragraph would have been devoted to that now trite subject, with a more complete description of the proposed application. The lines of research suggested had already been noted. In good time, it is hoped to be able to give some interesting facts that will fully satisfy every demand. Meantime, those here given are perhaps not wholly without interest.

It should be added that the Chamberlain experiments were made upon the head of an "experimental engine" in the Sibley

CCCCXXIV.*

HYDRAULIC TRAVELLING CRANES.

BY ERWIN GRAVES, CAMDEN, N. J.
(Member of the Society.)

IN the manufacturing and engineering operations of the day, the requirements for lifting and lowering, handling and transportation, of the innumerable heavy articles to be moved, have called into use a variety of appliances which may be classed as Cranes; of these, the type known as Travelling Cranes, through their general adaptability, has grown in favor to such an extent that in some fields of the best present practice they are used almost entirely.

The ordinary travelling crane consists of an overhead beam or bridge, the ends being supported by carriages which traverse upon parallel tracks, usually secured at an elevated position to the side walls of the building, or placed upon a special trestle prepared for them. This bridge is provided with a track, upon which traverses a trolley, from which the load is suspended. The requirements necessary for a crane of this kind are: 1st, Means for hoisting and lowering from a point attached to the trolley; 2d, means for moving the trolley, with its suspended load, across the bridge in both directions; 3d, means for moving the bridge and load along its tracks in both directions. It will thus be seen that a complete travelling crane requires three independent motions, and means for the reversal of each of these, with a further provision for holding the load suspended at any point. The methods of accomplishing these results might be classed as follows: 1st, Hand-power, when all the operations are performed by manual labor, applied through endless-chain wheels, by an operator upon the ground-level; or, the operator may be upon the bridge and apply the power through hand-gearing. 2d, By locating an engine and boiler upon the bridge or trolley and using steam-power for performing the various operations. 3d, By conducting power to the bridge by means of a rapidly running rope belt, or a square shaft, located along one of the walls or trestles; the

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

power being taken off such belt or shaft by the proper appliances attached to one end of the bridge. With power so conducted, it is utilized through trains of gearing to perform the work, the operator for controlling the motions being stationed on the bridge. 4th, The mechanism of the crane may be driven by electric motors, located thereon; the proper current being communicated from conductors along one of the walls.

As will be readily understood, hand power cranes are slow and tedious in their operations, particulary if the machines have capacity for heavy work. It is also obvious that those carrying an engine and boiler are not well adapted for use inside of buildings, and are therefore confined almost exclusively to work out-of-doors.

Having thus briefly referred to the various forms of travelling cranes, the writer will describe a crane of this type, devised by him, in which the actuating power for each of the various motions is obtained independently and from hydraulic pressure. He believes this application and arrangement to be new, and to possess merit. As this paper is intended to be more of an explanation of methods and results than a description of mechanical details, only such details will be introduced as are necessary to a proper understanding of the subject in hand.

Fig. 42 shows a bridge with ends mounted upon carriages, fashioned to move along tracks on wall. On one end of bridge are secured two horizontal grooved wheels, *C* and *G*, and at opposite end, one horizontal wheel. The trolley *T*, which is free to move along the bridge, has, as shown, four vertically hung grooved wheels. At some point, as in one corner of building, is placed a cylinder and plunger with sheaves, and rove with a wire rope as shown. This wire rope leads over the wheel *B*, thence horizontally along the wall and around the wheel *C*, thence around one of the wheels in the trolley, and down, and under the wheel in the fall-block, and up, and over the wheel in the trolley, and to the wheel *E*. After passing around this wheel, it is returned in same manner around the trolley and fall-block wheels to the wheel *G*, from which point it leads to, and is secured at, *H*. It is clear that, with an object suspended from the fall block, the trolley is free to be moved along the bridge; also, that the bridge is free to be moved along its track, and that such suspended load can be brought over any point within the rectangular space covered by these motions. No explanation is

FIG. 42.

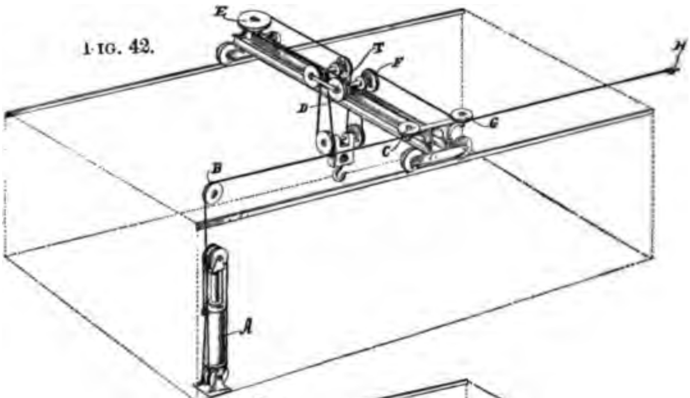


FIG. 43.

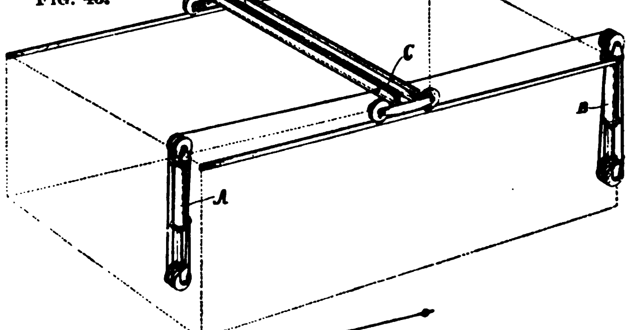
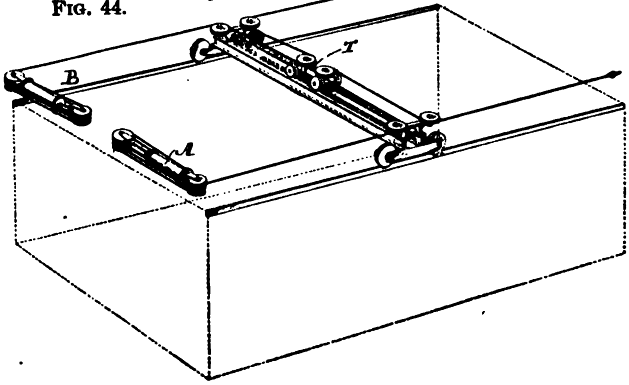


FIG. 44.



required to see that the pushing out of the ram results in lifting the fall block, and that with the rope passed twice about the cylinder and ram sheaves, one foot of motion in the ram produces an equal amount in the fall block.

power being taken off such belt or shaft by the proper appliances attached to one end of the bridge. With power so conducted, it is utilized through trains of gearing to perform the work, the operator for controlling the motions being stationed on the bridge. 4th, The mechanism of the crane may be driven by electric motors, located thereon; the proper current being communicated from conductors along one of the walls.

As will be readily understood, hand-power cranes are slow and tedious in their operations, particulary if the machines have capacity for heavy work. It is also obvious that those carrying an engine and boiler are not well adapted for use inside of buildings, and are therefore confined almost exclusively to work out-of-doors.

Having thus briefly referred to the various forms of travelling cranes, the writer will describe a crane of this type, devised by him, in which the actuating power for each of the various motions is obtained independently and from hydraulic pressure. He believes this application and arrangement to be new, and to possess merit. As this paper is intended to be more of an explanation of methods and results than a description of mechanical details, only such details will be introduced as are necessary to a proper understanding of the subject in hand.

Fig. 42 shows a bridge with ends mounted upon carriages, fashioned to move along tracks on wall. On one end of bridge are secured two horizontal grooved wheels, *C* and *G*, and at opposite end, one horizontal wheel. The trolley *T*, which is free to move along the bridge, has, as shown, four vertically hung grooved wheels. At some point, as in one corner of building, is placed a cylinder and plunger with sheaves, and rove with a wire rope as shown. This wire rope leads over the wheel *B*, thence horizontally along the wall and around the wheel *C*, thence around one of the wheels in the trolley, and down, and under the wheel in the fall-block, and up, and over the wheel in the trolley, and to the wheel *E*. After passing around this wheel, it is returned in same manner around the trolley and fall-block wheels to the wheel *G*, from which point it leads to, and is secured at, *H*. It is clear that, with an object suspended from the fall block, the trolley is free to be moved along the bridge; also, that the bridge is free to be moved along its track, and that such suspended load can be brought over any point within the rectangular space covered by these motions. No explanation is direction is produced. A form of valve for producing these results in the proper way, by a simple lever motion, will be described farther on. As in hoisting and lowering, it is clear that the valve controls all speeds, from the fastest to the slowest.

The means of moving the trolley upon the bridge is illustrated by Fig. 44. Here are shown, as located horizontally in one end of the building and at the level of the crane, two cylinders and rams with sheaves, and having ropes rove about them. The cut shows how these ropes pass along the wall, out over the bridge to the trolley, back to and along the wall to the opposite end of the building, where the ends are secured. When pressure is admitted to both cylinders, the resulting pull through the ropes is in opposite directions at the trolley, holding it in one position upon the bridge, but the bridge remains free to be moved along its tracks. By the same process of cutting off, exhausting, and reversing the water supply, as described for the bridge-moving rams, the requisite motions in both directions are obtained, and with the same range of speeds.

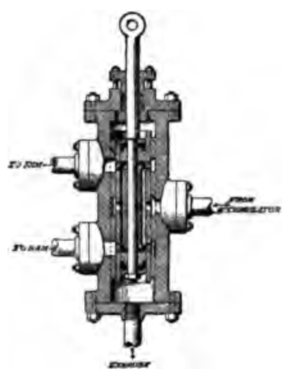


FIG. 45.

Fig. 45 shows a well-known form of hydraulic valve, modified so as to operate the bridge-moving and trolley rams in the way described, and, by omitting one of the ports, the hoisting and lowering also. As shown in the cut, the piston is in the middle of its stroke, and it will be observed that both ports admitting pressure to the rams are slightly open; this results in the rams pulling against each other, as described. A slight motion of the piston in one direction closes one of these ports, and opens wider the way for the pressure supply through the other; further motion to this valve piston begins to open the way for the cut-off cylinder to exhaust. The result of this is motion to the bridge or trolley, and the speed of such motion is controlled by regulating the rate of the exhaust. Should there occur a too sudden shutting off, while the bridge was in quick motion, the effect of the momentum would be to put an increased pressure upon the holding-back cylinder. An examination of the detail of the valve will show that if from any cause the pressure in a shut-off cylinder should rise above the working pressure in the valve chamber, it would find relief through acting against the external surface of the cup leather packing; such relief will be back into the pressure pipe, and the result on the motion to the bridge will be like the application of a brake. If with the crane at a standstill the valve is thrown suddenly open, there will be no sudden stress put upon any part; the bridge will move with an accelerating speed, as the actuating force can overcome its inertia, just as a locomotive puts in motion a train of closely coupled cars.

It is an easy matter to combine in one machine the separate methods of producing the requisite motions as before described, such combination working together in a very simple manner. The details of doing this can be modified to suit governing circumstances.

Fig. 46 shows a design which may be used up to 25 tons capacity or more. In this the bridge is simply a box girder, in which vertical strength and lateral stiffness are obtained in the most direct manner, and the end carriages, rope-wheel brackets, trolley, and fall block are of the simplest construction.

For light capacity cranes, the bridge can be two I-beams, connected only at the ends, and with the trolley working on the top flanges; the fall block being suspended between the beams by two parts of rope only. Otherwise the general arrangements would remain the same, except that the hoisting rope, in leading back across the bridge to its primary end, would not engage with the fall block. In such a case, with the same combination of ropes at the hoisting ram, a given stroke would produce double the hoist at the fall-block hook.

There is another modification which can be introduced, in which the trolley-moving rams are omitted, and the cross motion obtained, when with light loads, by direct pushing on the part of the workman; or, when with heavier loads, by the application of brakes to the wire rope wheels on the trolley. Referring to Fig. 42, it will readily be seen that if the rope wheel *D* be locked, hoisting or lowering will produce a motion to the trolley of four feet for each one foot of vertical motion to the hook block; also that moving the bridge with this wheel locked will cause the hook to traverse diagonally across space covered. With brakes on wheels on the opposite sides of the trolley, and under

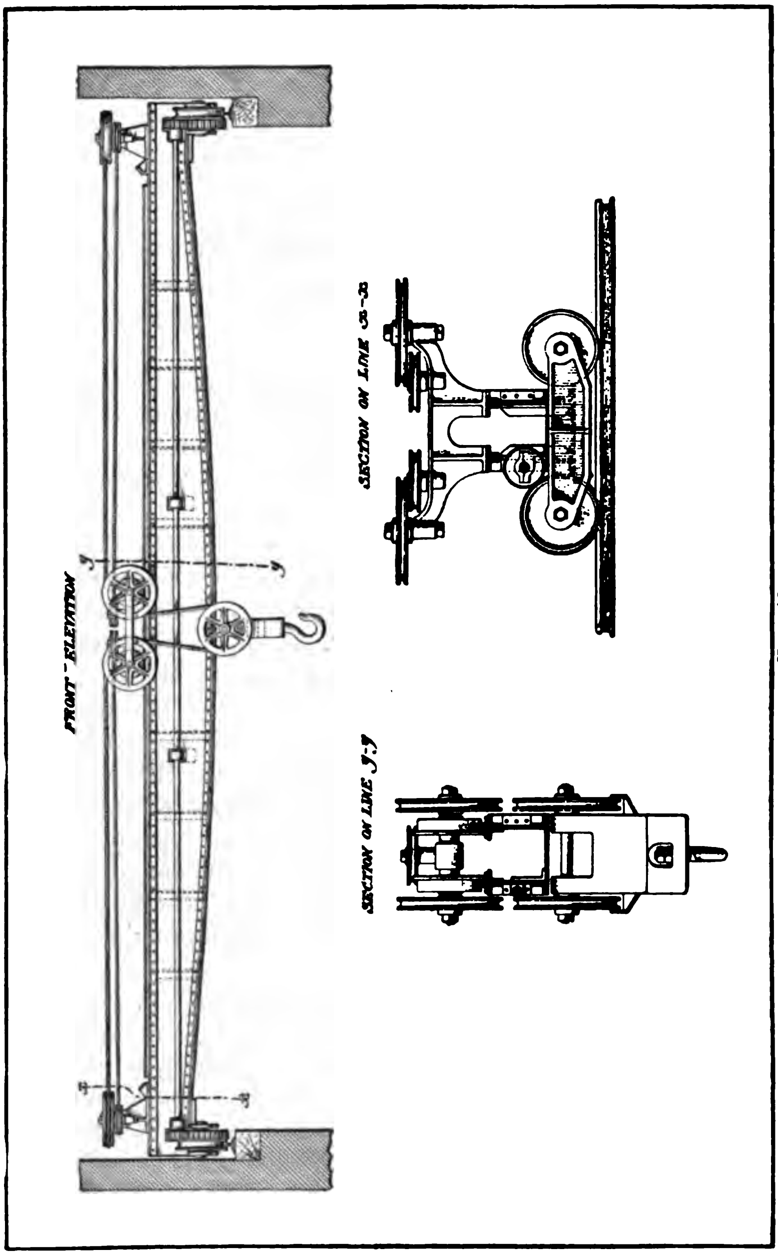


FIG. 46.

control of the workman, very satisfactory work can be accomplished.

In these hydraulic travelling cranes the most desirable locations for the operating rams are: hoisting cylinder with ram working upwards, and placed in one corner of room; adjoining this, one of the bridge moving cylinders, with ram working vertically downwards, while in the corresponding position at the opposite end of building should be the other bridge moving ram. The trolley-moving cylinders and rams may be placed horizontally on the end or side walls. These locations permit of the ropes being led in the most direct manner to the bridge, without intervening sheaves. Should circumstances demand, any of these cylinders can be placed elsewhere, or they may all be located at one point. It is desirable, however, that the bridge and trolley rams should either act downwards or horizontally, so that during nights and at other times, when the pressure may be off, they will remain in position in the cylinder—not dropping down and deranging the ropes.

When, as is usually the case, there has to be performed a variety of lifting, both light and heavy, it is desirable, both for economy of power and increased speed, that rams of smaller diameter be employed for the lighter work, holding in reserve the larger one for the heavy work. Fig. 47 shows an effective and cheap method for accomplishing such results, and, at the same time, giving the operator means to make the change at a moment's notice, without leaving his station. The method is simply to use telescopic plungers, allowing two or more powers to be provided, in which the larger ones, with the greater lifting power, are clamped down when not required. The cut will show how this is accomplished, without explanation.

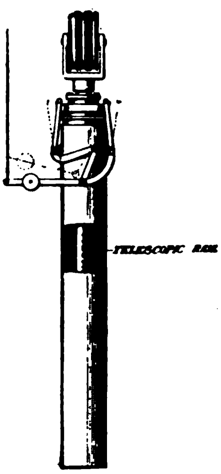


Fig. 47.

The movements of the crane may be controlled by an operator stationed in a "pulpit," conveniently located to provide a clear view over the space which crane covers; this method permits also two or more cranes to be operated by one man. In such cases the valves would be placed where the operator is stationed, and pipes laid to the different cylinders. Valves, such as before described, can be combined together, so that one supply and one discharge pipe answers for all. The three operating levers con-

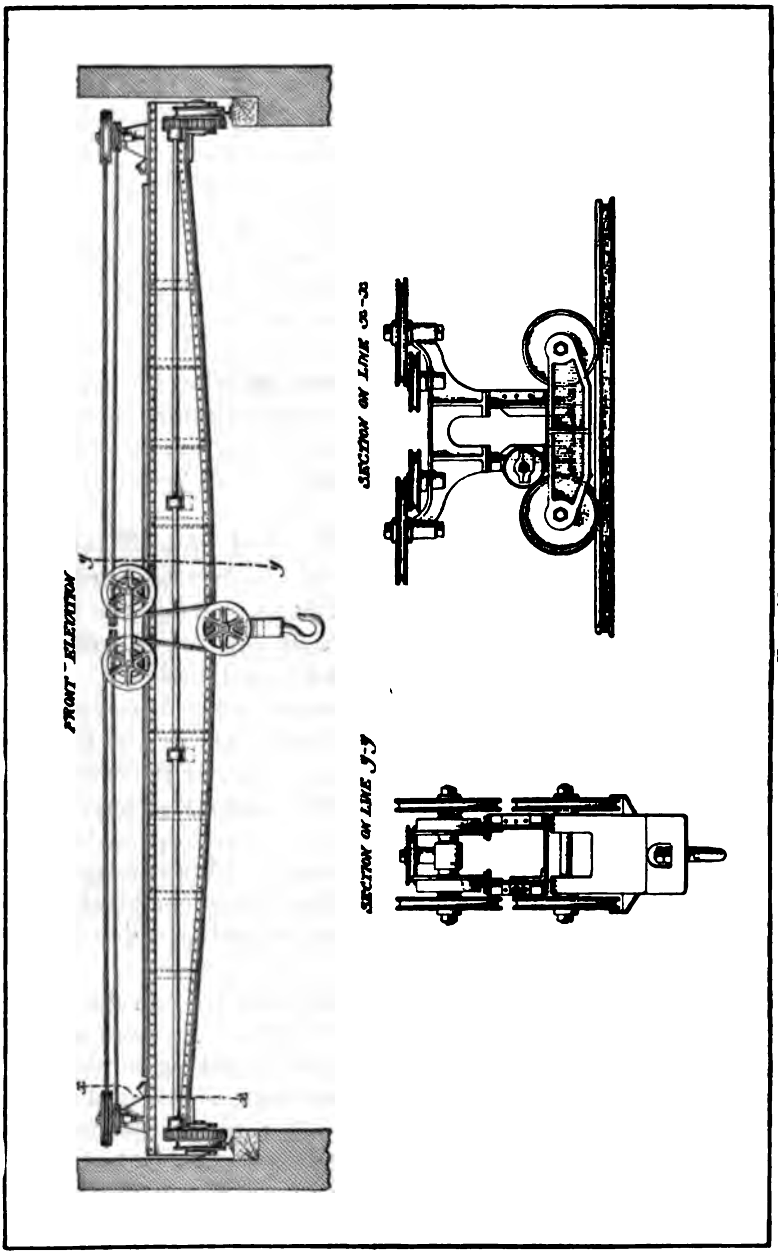


FIG. 46.

control of the workman, very satisfactory work can be accomplished.

In these hydraulic travelling cranes the most desirable loca-

trol all the motions and speeds of the crane, and act with Where the area covered by the crane is too extended for veniently operating from one point, there may be other po or there can be an elevated gallery, leading along one side of building, wide enough to afford passage for one person, and valves operated by means of wire ropes stretched along it. should it be desired, there can be an operator's platform tached to the bridge, and the movements controlled from as in the case of passenger elevators.

For the purpose of showing that the application of hydra power to travelling cranes is not limited to those of light pacity, Fig. 48 is introduced. This shows a design by the wr for two cranes, to be placed on the same track, each of 45 capacity, 43 feet hoist, 50 feet span, and 300 feet of travel. these the same simplicity and directness is obtained as in t of smaller capacity, and they are now in process of construct practically as shown.

Somewhat more than three years ago the writer designed a foundry then being built, two cranes of 15,000 lbs. capa each. He believes these to have been the first application of principles herein described, that of employing hydraulic p from a fixed location, and with separate motors, for act ing the different motions of a travelling crane. These cranes were constructed, and have been in constant use two and a half years. Their performance, under the requ ments of daily work, has been satisfactory in the highest gree; during this time there have been no repairs requ whatever, except one or two renewals of the flax packing in stuffing-boxes of the rams. The care and operation of these for the whole time, been in charge of ordinary foundry he one man operating both cranes. There has never been a injured, or a mishap of any kind resulting from their use. person seeing the easy, noiseless, and rapid manner in w these cranes perform their work, and the simplicity of t parts, cannot but be favorably impressed.

The number of wheels about which the hoisting-rope pa may suggest the thought that there is here a large loss of p from friction; actual results do not show this to be the The loss of power from bending wire rope about wheel such diameter is extremely small. The principal loss oc from the axle friction. In these, the bearing strains from

pull of the rope are all which have to be provided for, and, as there are no sudden shocks, these axles can be of small diameters, or they may be on roller bearings; in any event, the large diameter of wheels reduces this friction to a small amount.

With a crane in operation, the total friction can be readily determined by simply attaching a gauge to the cylinder and noting the difference of pressure, when a load of known weight is raised and lowered. One-half of this difference of pressure multiplied by the area of the plunger in inches, and further multiplied by the ratio of plunger speed to hoisting speed, gives at once the result in pounds. This result includes the friction of the ram in the stuffing-box, as well as all the other moving parts. Tests made in this manner, upon the two cranes before referred to, taking them without any special preparation, just as found after a day's work, and with varying loads up to their full capacity, indicated very uniform results in the percentage of friction. In one crane, the average was 9% and in the other less than 12% of the loads lifted, including dead load.

As chains are so largely used for carrying the suspended loads in cranes, it may appear to some that substituting wire rope for this purpose is attended with risk. The writer does not think this to be the case, and is further of the opinion there are certain advantages in its use. Wire ropes possess their calculated strength to a degree unattainable in chains, the reason being that in one there is a series of links, each of which must have strength to resist rupture from the entire strain, while the other is made up of a combination of separate pieces, running throughout its length, of iron or steel in its most reliable form. In the question of costs and weights there is a difference in favor of the rope; as, for example, 100 feet of chain which has an estimated ultimate strength of 28,000 lbs., would weigh over 500 lbs. and cost $37.00, while its equal in pliable wire rope would weigh but 60 lbs. and cost less than $11.00. Those who have observed chains in service know in what a sudden and unlooked-for manner they frequently give way, either through defective workmanship in the manipulating and welding, or inferior material, or through what is still an open question, the deterioration of iron fibre from use—all sources of weakness invisible to outward inspection. In ropes, outward evidence of approaching failure appears gradually, by showing broken wires on the outside of strands at points where wear occurs. In making a comparison of the lasting qualities of the two, it should be borne in mind that the wearing away to the danger-point in crane chain by use takes place on the inside ends of the links, when the surfaces are in contact with one another. The wear on the outside surface from contact with sheaves and drum will be found comparatively trifling, even where the use has been such that the ends of links are reduced to one-half of their original area. With wire rope there is only the outside surface to wear; with the groove in wheels turned properly, and with wheels so set that the lead is straight into them, this will be extremely small. There are cases where the large diameter of the sheaves required would be detrimental, but in designing the machine this drawback can be largely avoided. Where the exigencies of the case demand, steel-wire ropes can be used on wheels with diameters as small as 25 times that of the rope for smaller sizes, and 35 diameters for the larger sizes. Such diameters, while much less than the rope manufacturers advise, will be found to give very satisfactory results, though it is good practice to use larger wheels where such can be employed. To give an idea of what may be expected in the way of lasting qualities, the writer would state that in the 15,000-lb. cranes referred to, after over two and one-half years of constant service, the ropes are so little worn that they would appear to be good for an equal further use.

Where the strain is greater than can be carried by one rope of moderate size, or the necessary diameter of wheels inconveniently large, two smaller ropes can be used, the wheels being properly grooved and both ropes passed about them, as though but one was employed, the fixed ends being secured to an equalizing bar, so that they will be strained alike.

The speeds at which cranes can be operated depends largely on the ability to transmit the necessary power to them, and in means for applying power in ways which will not result in sudden shocks. This is a feature in which the hydraulic crane excels. Take, for example, an accumulator with a plunger $13\frac{1}{2}$ inches in diameter, and loaded to give 1,000 lbs. pressure; when up, with a stroke of 14 feet, there is stored 2,003,960 foot pounds of energy, the equivalent of $60\frac{7}{10}$ H. P. acting for one minute. Transmitting power by hydraulic pressure is accomplished with very small loss; the volume of water in this case would be about

104 gallons, an amount which will readily pass through a 2-inch valve and pipe, into a cylinder, in one minute. Allowing a more liberal estimate for frictional loss than indicated by the tests here noted, and assuming that 75% of the energy would be available for lifting at the hook, there would be 1,502,870 foot pounds, which could be exerted in one minute, or 45½ H. P. for the same time, and would result in lifting a load of 30 tons 25 feet. For bridge and trolley motions the necessary power for any desirable speed can be readily communicated. Such result in lifting is probably 4 or 5 times greater than builders of the ordinary form would care to guarantee. To some it may appear that there is no occasion for such a rapid application of energy; leaving it an open question whether there is or not, it is here only pointed out that such results can be obtained, and moreover without any undue strain upon the machinery. Heavy demands for power can only be of short duration, and as the accumulator holds in store energy for such draughts, it follows that the motive power can be of very much less capacity.

There are difficulties inseparably connected with the methods of accomplishing results in the usual form of power-driven travelling cranes to which it may not be out of place to briefly refer. The means of communicating power to a point upon the bridge, whether accomplished by a square shaft or a rope belt, has objectionable features, which in practice give more or less trouble; but this part will be passed by, and the mechanism upon the bridge itself considered. Here the driving power is in continuous and uniform motion—to produce the various motions requires trains of gearing, reversing appliances, friction clutches, safety devices, etc., the whole forming an intricate combination of machinery which must be under the most perfect control. As before pointed out, the variety of work for a crane to do, makes it desirable that there should be different rates of motion. To meet this, manufacturers usually provide two or more speeds; but to accomplish such result with a uniform driving motion necessitates introducing further mechanism for each change.

When motion is to be produced—as, for example, moving the bridge—it is usual to connect with the power by means of a friction clutch, or some device which will permit of a certain amount of slipping, such slipping being necessary to permit the bridge to acquire its motion gradually. Herein is a source of trouble; clutches do not always act as they are designed to do, nor are they applied with uniformity. A powerful clutch, suddenly applied, may produce strain many times greater than that calculated. The writer knows of a case where, with a crane of moderate capacity, and without a load, a 3-inch square driving shaft was wrenched off through the too vigorous application of the bridge-moving lever. Undue strains from this same cause are in a less degree produced, and constantly tending to destructive results, in the trolleying and lifting trains of gearing. It will readily be understood how these increase with the speed.

The intervals of time in which a crane is not in actual motion forms a large proportion of the whole. During this, the driving mechanism has to be in continuous motion, consuming power while no work is being done. From the indirect means which it is necessary to employ to accomplish actual work, the frictional loss is very great, and the motive power must be capable of fulfilling all demands, as there is no ready means of storing power, as with an accumulator. The driving parts, which are in continuous motion with their gearing, are noisy and wear rapidly, particularly when, as is not infrequently the case, the crane has to perform its work in clouds of dust. The care and operation of such a machine require the services of a skilled mechanic.

With the application of hydraulic power to travelling cranes, the writer believes that there will be found the following advantages: General adaptability, moderate cost, and simplicity of construction, freedom from repair, ease, safety, rapidity and noiselessness in action, and economy of operating power.

DISCUSSION.

Prof. John E. Sweet.—In the hope that the discussion on the above paper may be broadened to a consideration of travelling cranes in general, I am led to describe some novel features which we have introduced in a 5-ton hand travelling crane of 30 feet span. To speak of a hand travelling crane may be rather behind the times, but the features described are applicable to power cranes.

The available space between the bottom of the roof trusses and the top of the rails was 5 feet 6 inches, and in this space we have a lattice girder bridge with trusses 5 feet deep, and a trolley with wheels 4 feet in diameter. This is done by putting the trolley inside the bridge, as has been done in the two 45-ton cranes described in the paper, but we avail ourselves of that well-known advantage possessed by a lattice girder over a plate one, of a great reduction in weight. The weight of the girder and trolley does not exceed 2 tons. The lower chord of the trusses, which are of special section (Carnegie 345), forms the rail for the trolley. The trolley is in general design like a bicycle, with the large wheels 4 feet and small ones 10 or 12 inches, nine-tenths of the load being on the main axle. The bridge is also carried on wheels 4 feet in diameter, with the flanges on the outside and the faces of the two coupled wheels turned conical, as is done on car wheels. The wheels are coupled by a gas-pipe across shaft and two small pinions working in internal gears. There is no squaring device whatever, and we have found no use for one.

A travelling crane is such an improvement on nothing that it would be a poor one, indeed, which did not give satisfaction. What pleases us especially in our own is its free action obtained by the large wheels and no useless mechanism. The lifting device is a simple one-ton triple-geared hoist made by the Yale & Towne Co., with the hoist chain strung over multiplying sheaves. This arrangement is simple; it gives very nearly, if not quite, as great efficiency as any mechanical rig; its ingenuity will never be overestimated and it is quite satisfactory. With the largest weight we have to handle (about 3 tons), one man can handle it with one hand, and move it easily at the rate of 100 feet a minute in either direction. I believe for our use no power crane could equal it in economy.

Mr. Louis G. Engel.—I want to ask a question about the valve shown on Fig. 45. I have always understood that with as high pressure as 1,000 lbs., and with this construction of cup leathers inside of a bushing pierced with orifices, it was impossible to prevent the cup leathers from being drawn into the openings. Has there been any trouble in that way?

Mr. Henry L. Gantt.—The Midvale Steel Works has placed an order with R. D. Wood & Co. for a 40-ton hydraulic travelling crane, and they have insisted on having two valves, a stop and a throttle valve. With this combination, a much more delicate movement of the crane can be obtained. The cranes which they have in use in Camden seem to work very satisfactorily. They start up very slowly and run very rapidly. I have never seen any crane, whatever, handle a flask in a foundry with more ease and more carefully than those cranes do.

Mr. Douglas G. Moore.—I think I can help to answer a question. We have built at times, and do build now, a great many hydraulic presses up to 20 inches in diameter, in which we use a cup-leather packing made of sole leather, and turned in the ordinary old-fashioned way; and to prevent it in some cases from knocking the heads out, after a good many different arrangements, we conceived the idea of a safety valve by boring a hole at a certain height in the cylinder so that when the leather packing passes that hole the liquid would squirt out and the piston would stop. Now we can bore a hole to $\frac{3}{16}$ of an inch in diameter, and under 1,500 lbs. pressure to the square inch, the leather will never go into that hole so that you can ever find a mark on it. We have never had to bore a larger hole than $\frac{3}{16}$. If I had to bore a quarter hole I would not be afraid to do it. The leather is very firm and hard. It does not get at any time soft enough to force itself into this $\frac{3}{16}$ hole. I have used pressures up to 3,000 pounds pressure, not intentionally, but by carelessness of the man using it. I have never known leather packing to be destroyed in any instance from forcing into the hole.

Mr. Engel.—Do I understand the gentleman to say that the pressure is relieved after the leather packing passes the hole?

Mr. Moore.—The pressure is not fully relieved, but the hole is large enough so that the pump cannot discharge enough liquid to lift the ram and at the same time stop the leak. The hole is put there for the purpose of relieving the pump, with a little pipe attached, running it off where the waste water might go.

Mr. Engel.—Does the packing come back over the hole, under pressure, on the return? I can see readily enough that the cup leather will go past it one way, but will it come back again?

Mr. Moore.—The leather will come back after the pump is stopped. These cylinders were built vertically. The rams settle after the pump is relieved or stopped. At that time there is no pressure under the piston, excepting a light pressure to lift the liquid over into the cylinder.

Mr. Engel.—That is just the point I wanted to make. If there is no pressure under the piston, the cup leather will come back, because the cup leather is not expanded; but should there be pressure under the cylinder with the cup leather expanded, would the cup leather come back?

Mr. Moore.—I should think it would, because we build them double head; that is, they are packed two ways sometimes, one leather turned up and the other turned down, and I have known a back pressure to be on one side up to 500 lbs., and that leather would never be forced into the hole.

The President.—Do you have a row of the holes so as to make sufficient area to relieve the pressure, or is there only one? If so, in what sized cylinder is a $\frac{3}{16}$-inch hole used?

Mr. Moore.—We build presses up to 20 inches diameter of cylinder, and an eighth of an inch hole will stop that piston at almost any point after the packing passes the hole. The pump plungers are small, perhaps $\frac{3}{4}$ to $1\frac{1}{4}$ inches in diameter. What surprised me was that it stopped the piston so quickly with such a small hole. In the first one which we built, we turned the bent pipe so that the stream struck the man working the press just about the eye when it went off. But the fellow got tired of that and he turned the pipe down a bit. But it answered the purpose all right. The piston seemed to stop as soon as the packing passed the small hole.

The President.—I suppose if you wanted to relieve the press more rapidly it would be easy to put in several holes instead of one. Is the hole countersunk at all on the inside—to prevent scratching of the leather?

Mr. Moore.—Not at all. We bore a large hole almost through the cylinder to within perhaps a quarter of an inch of the inside, and then bore this very small hole the balance of the way. Three-sixteenths is the very largest we ever put in. One-eighth is about our practice.

Mr. Huston.—Is that leather subject to any particular treatment to keep it from softening with the water?

Mr. Moore.—Not at all. We buy good oak-tanned leather. We have made a 30-inch packing, but we had so much difficulty in getting the leather that we abandoned it. We build the same presses particularly for oil refineries, and they use in the pumps paraffine oil at about 150° fire test, with a cold test of 18° or 20°, and the leathers will work in the oil 12 months before they will give out if you get good leather to start with. The trouble is to get leathers of even thickness. You have to get down into the belly of the hide, where the leather becomes thin. There the leather is softer, and does not seem to stand as well as the back. I think it lasts as long with oil as it does with water.

Mr. Engle.—In this connection I wish to call attention to a valve for the same purpose in my paper, Fig. 49. This valve, of

English origin, is superior to any valve with orifices, because the passages in the ends of the piston itself are tapering, and the cup leathers are not obliged to pass holes into which their edges might be pushed by the pressure.

Mr. L. R. Lemoine.—In the larger illustration, showing two travelling cranes on the same run-way in a machine shop, is also shown a gallery and series of groups of operating levers (of which bare mention is made) arranged so that the cranes may be operated for any of their motions at any of the stations on the gallery, which is usually placed so as to run along the line of bridge travel, and thus occupies about the same space as would a cage suspended from a crane for its clearance. This combination of gallery and operating levers, etc., affords ready control of the valves, which are usually located on the shop floor near the cylinders, whether by positive connection, such as ropes, rods, etc., or by air or electricity, and was originally patented and designed by the writer.* The crane, as originally built, was operated from a central pulpit, which limited its use to comparatively short lengths of bridge travel, as no practicable means had *then* been devised to control the valves on the floor from the suspended cage on a moving crane bridge, although an electrical device has been since worked out by the writer, though not as yet tried.

The gallery and combination of levers—simple arrangement that it is—while making it possible to use the crane for long bridge travels, has a very decided advantage, in many instances, over the ordinary suspended cage, in that the operator can stand at any desired point, one side or the other of the crane, and thus at all times can see the work on the floor to be handled; and may, for that matter, stand at one end or at any station along the line of the gallery, and control the crane in all its motions, starting the bridge travel, for instance, with one set of levers, and stopping it with another; while, with the cage attached to the travelling crane bridge, the operator must always maintain relatively the same view of the hook, which view is often interfered with, as when, in an erecting shop, for instance, it is desired to handle something which lies beyond a large piece of work.

Another decided advantage is that one man may also control two cranes in ordinary service, whether they be on the same or on adjoining parallel run-ways.

* See patent No. 431,000, June, 1890.

Of course, this arrangement is merely a detail of the crane described in the paper referred to, but it has been considered of sufficient value to have been adopted in connection with eight or nine of the cranes we have received orders for.

*Mr. Erwin Graves.**—Replying to the remarks of Prof. Sweet, I would say that my paper refers only to the methods of operating a travelling crane by hydraulic power, through the means of wire rope from rams at fixed points. The particular form of bridge is not treated of, except to point out that it can be of the simplest construction, and that the operating mechanism on the crane itself does not require any complication or special design.

If desired, a latticed one can be employed. It is true that with lattice work and girder of greater depth, a bridge of given span and capacity can be constructed with less weight of metal than can a plate girder. This extra depth, though, is in most cases an objection; it is usually desired to have the greatest possible lift with a crane of this sort. The hook might, it is true, be arranged to rise between the lower chords, but such is not altogether desirable. In the matter of cost, it is quite probable that the increased mechanical work would about offset the saving of material, and that the total costs would not differ materially, while in appearance the plate girder is much more pleasing and the consumption of power in moving this trifling extra weight is a matter of no moment, when accomplished by other than hand power. The large wheels are certainly a desirable fixture in all parts where such can be readily employed.

With a hydraulic travelling crane the whole question of form of bridge and mountings can be left to the circumstances which surround each individual case.

Replying to the general remarks about valves, I would say that the one referred to is the well-known "Critchlow" form, and which has, in one form or another, satisfactorily filled the requirements for hydraulic use, particularly in the cranes about the iron and steel-making plants, and when the service is severe.

The perforations in the brass casing, which also serve for the ports, are but $\frac{3}{32}$ inch diameter each, and over this the cup leathers pass freely and without trouble. In the case of the "four way" valves, which control two rams acting against each other as described, the pressure does not act in the direction to force the leather into the holes at all—the action is all in the opposite

* Author's Closure, under the Rules.

direction, as a leather passes over the port. By making a sketch one will readily see how this is, and that it comes from the fac that all throttling of pressure is as the ports are exhausting.

With the "three-way" valve, as used in hoisting and lowering the case is different. In hoisting, the pressure is in the directio of forcing one of the two leathers into the holes as it passes alon the port. It acts with a force equal to the difference between th accumulator pressure and that in the lifting cylinder. In lower ing, the case is same as with the "four-way" valve—it acts th opposite way from pressing leathers in.

Owing to the smallness of those holes there is scarcely an trouble with these valves in that particular, though the leather do give out in time. In use these valves give the least trouble o any I know.

CCCCXXV.*

AN INTERESTING EXPERIMENT WITH A LUBRICANT.

BY G. W. BISSELL, ITHACA, N. Y.
(Junior Member of the Society.)

THE experiment referred to in the title of this paper was undertaken for the purpose of determining the law governing the variation of the *coefficient* of *friction* of a lubricant, and the *rate* of *feed* of the same to the journal, all other conditions being constant. It was conducted in the Mechanical Laboratory of Sibley College by an advanced student, who worked under the supervision of the writer.

The Thurston Railroad Lubricant Tester was used for the determination of the coefficient of friction. A practically constant source of power was furnished by the line-shaft in the laboratory, so that a uniform velocity of rubbing was assured. The journal was hardened steel and the bearings were fitted with "water brasses," by which the operator was enabled to maintain nearly constant temperature of the rubbing surfaces. The accessory apparatus needed was a device for feeding the oil as required.

The following plan was adopted:

A graduated burette of 50 c.c. capacity was secured to the upright of a ring-stand and the whole placed so that the top of the burette was about 15 inches above and 6 inches in front of the testing journal. The burette was filled with the lubricant (a mineral engine oil), and a siphon of very fine glass tube, with the end of the longer leg drawn down still finer, was placed in the burette. When so placed, the point of the siphon hung over and about 4 inches above the oil-hole in the housing of the testing machine pendulum. In the oil-hole was placed a copper rod, considerably smaller than the hole itself. This rod rested on the journal. A piece of candle-wicking depended from the point of the siphon, and was tied to the upper end of the rod. It is evident that the rate of flow of the oil from the siphon to the journal must decrease gradually as the level of oil in the burette is lowered, and will cease altogether when the level is reduced to that of the point of the siphon.

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

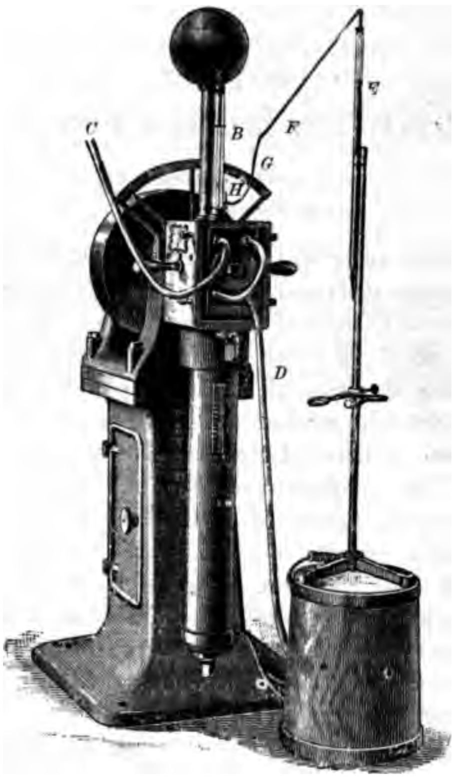


FIG. 51.

The final arrangement of the apparatus was as shown in Fig. 51, to which the following references are given:

B, Thermometer, giving temperature of bearings.
C, *D*, Rubber tubes for circulation of water through the bearings.
E, Burette, furnishing supply of oil.
F, Siphon, controlling supply of oil.
G, Candle-wicking.
H, Copper rod.

The continuous counter was not used during the experiment; but a tachymeter was belted to the machine and occasional readings of a hand counter were taken as a check.

In the preliminary experiments with the apparatus some difficulty was met with in keeping the oil from running off the journal.

This was finally overcome by following the suggestion by Professor Denton in his recent paper before this Society.

The first test was made with a total load of 2,000 lbs. on the journal, corresponding to about 70 lbs. per square inch.

Every ten minutes, observations were made and recorded of the following data: time, speed, burette reading, temperature, and total friction.

This test was not wholly satisfactory, and two others, at 3,000 and 4,000 lbs. total respectively, were made, with better results, as shown in the appended log and curves.

Three sets of curves were plotted—the first with time as

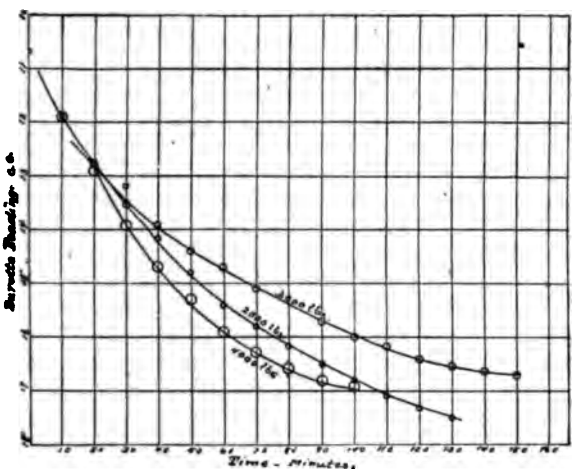


FIG. 52.

abscissæ and burette readings as ordinates (Fig. 52); the second with time as abscissæ and the rate of flow of the lubricant in c.c. per square inch of projected area and per minute as ordinates (Fig. 53); and the third with the rate of flow as above as ordinates, and the coefficients of friction as abscissæ (Fig. 54). The ordinates for the second and third sets were obtained from the first set of curves.

The curves of the third set exhibit the relation between rate of feed and the coefficient of friction, thus giving the information sought. Inspection of these curves shows that at low rates of feed the effect of pressure on the coefficient of friction is practically *nil*, which would tend to prove that, under such conditions, lubricated rubbing surfaces follow the laws of solid or "immediate" friction. But it should be noted also (see curves) that, at the rate of feed at which this state of affairs begins to be apparent, the augmentation of the coefficient of friction is dangerous to the continuance of the smooth running of the journal, as witness the irregularities in the curve for 3,000 lbs., as the rate of feed is reduced to a small amount. For this reason it is not advisable to reduce the rate of feed so far as to approximate to this limit. The question then arises, What is the safe limit in this direction? and how may this laboratory experiment be given a practical bearing?

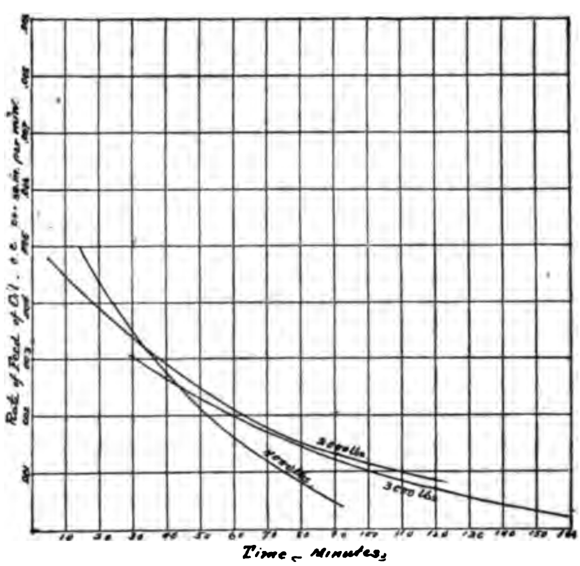


FIG. 53.

The rate of feed should in every case be so high, at least, that the coefficient of friction will not vary. In the curves, irregularities appear between the ordinates .003 c.c. and .002 c.c. Therefore for this case it is unsafe to reduce the feed below .003 c.c. per square inch projected area and per minute. If a series of such curves could be drawn for each of the various lubricants in common use, it is the opinion of the writer that a glance at that curve which fits the problem in lubrication which the designer may have in hand, will enable him to settle for himself the minimum safe rate of supply of lubricant.

It is the intention of the writer to continue the investigation in

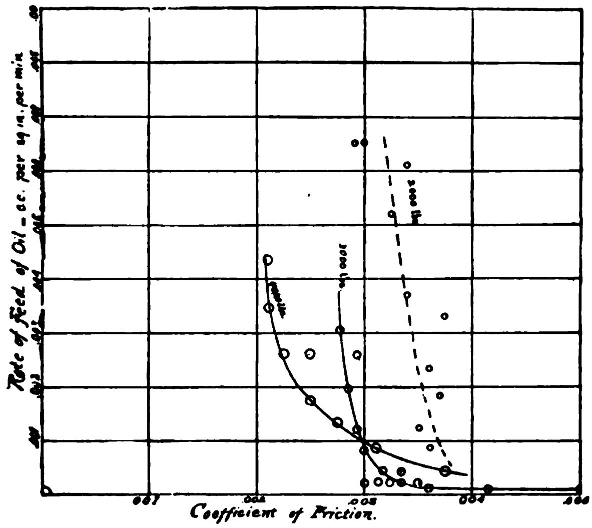


FIG. 54.

the line indicated above, and to publish the results in a later paper.

DATA OF TEST No. 1.

Total pressure on journal, 2,000 lbs.

1	2	3	4	5	6	7	8
Time.	Revs.	Burette Reading.	Temp. of Brasses.	Total Friction.	Coefficient of Friction.	Corrected Burette Reading.	c.c. oil per □″ area of journal per minute.
10.00	318	3.	59°F	62	.0031	3.	......
10	....	5.1	59	63	.00315	5.1	.00924
20	318	6.6	59	58	.0029	6.6	.00660
30	....	8.1	60	60	.0030	8.1	.00660
40	317	9.5	59	68	.0034	9.5	.00616
50	....	10.7	60	65	.00325	10.7	.00528
11.00	327	11.9	60	65	.00325	11.9	.00528
10	....	12.7	60	68	.0034	12.75	.00374
20	....	13.2	60	75	.00375	13.55	.00352
30	...	14.2	61	75	.00375	14.27	.00316
40	...	14.8	..	..	......	14.86	.00259
50	328	15.4	60	72	.0036	15.40	.00237
12.00	....	15.8	60	74	.0037	15.83	.00189
10	....	16.2	60	70	.0035	16.20	.00162
20	....	16.5	60	70	.0035	16.50	.00132
30	320	16.8	60	72	.0036	16.80	.00132
40	....	17.1	60	70	.0035	17.10	.00132
50	....	17.3	60	70	.0035	17.30	.00088
1.00	320	17.5	61	72	.0036	17.50	.00088

No. 2.

3,000 lbs.

1	2	3	4	5	6	7	8
Time.	Revs.	Burette Reading.	Temp.	Friction.	Coefficient of Friction.	Corrected Burette Reading.	c.c. of oil per min. and per □″.
10.05	319	12.8	60°	83	.00276	12.65	......
15	....	13.5	62	83	.00276	13.35	.00308
25	....	13.9	62	88	.00293	13.95	.00264
35	322	14.4	62	85	.00283	14.50	.00242
45	....	14.7	62	86	.00286	14.95	.00198
55	....	15.1	62	87	.00290	15.37	00185
11.05	....	......	....	..	......	.....	......
1.15	....	15.7	61.5	87	.00290	15.72	......
25	....	16.0	61.5	88	.00293	16.00	.00123
35	328	16.2	61.5	90	.00300	16.20	.00088
45	....	16.4	62.5	93	.00310	16.40	.00088
55	....	16.5	63	95	.00316	16.50	.00044
12.05	310	16.6	62	98	.00326	16.60	.00044
15	....	16.7	62	100	.00333	16.70	.00044
25	....	17.0	62	100	.00333	17.00	.00132
35	....	17.1	62	90	.00300	17.10	.00044
45	....	17.15	62	90	.00300	17.15	.00022
55	....	17.2	62	90	.00300	17.20	.00022
1.05	....	17.25	62.5	94	.00313	17.25	.00022
1.15	....	17.3	62	95	.00316	17.30	.00022
1.25	....	17.35	62	97	.00323	17.35	.00022
35	....	17.40	62	100	.00333	17.40	.00022
45	....	17.45	62	105	.00350	17.45	.00022
55	....	17.50	62	105	.00350	17.50	.00022
2.05	....	17.53	62	108	.00360	17.53	.00013
15	....	17.56	62	110	.00366	17.56	.00013
25	....	17.58	62	125	.00416	17.58	.00008
35	328	17.6	62	150	.00500	17.60	.00008

No. 3.

4,000 lbs.

1	2	3	4	5	6	7	8
Time.	Revs.	Burette Reading.	Temp.	Friction.	Coefficient of Friction.	Corrected Burette Reading.	c.c. of oil per min. and per □″.
12.20	330	11.9	68°	85	.00212	11.90	.....
30	....	12.9	69	87	.00217	12.90	.00440
40	....	13.9	70	87	.00217	13.90	.00440
50	336	14.7	70	87	.00217	14.70	.00352
1.00	....	15.3	70	90	.00225	15.30	.00264
10	....	15.9	70	100	.00250	15.90	.00264
20	327	16.3	70	100	.00250	16.30	.00176
30	....	16.6	71	110	.00275	16.60	.00132
40	....	16.8	71	125	.00312	16.80	.00088
50	....	16.9	72	150	.00375	16.90	.00044

DISCUSSION.

Prof. R. H. Thurston.—The investigation of which this paper may be taken as an incident, planned by me long since, has been on the programme for a long time awaiting a good opportunity to take it up formally and to do the work thoroughly; and we hope that we may see results after a while which may be taken as more than simply illustrative. But the work here reported, as the initial tentative research, is at least interesting as exhibiting the general facts of a very important and practically useful discussion. It is sufficiently well understood that we may have, in any given case of lubrication, an illustration of fluid friction or of the friction of solids, accordingly as we secure more or less thorough lubrication with a good friction-reducing material. With flooded journals under light pressures, the resistance may be that of fluid on fluid; or with heavy pressures, that of solid on solid. The two sorts are of radically different kind and of entirely different method of variation with varying pressures, temperatures, and speeds of rubbing. For the former, the well-known laws of fluid friction, resistances varying as the square of the speed of rubbing and independently of pressures on a given area of surface, and, for the latter, friction varying as a function of the pressure and independently of the speed or areas of surface up to the point of approximate abrasion, may be taken as well established and to-day universally admitted laws. As shown by Hirn many years ago, intermediate conditions between those of free lubrication and of greatly restricted supply of the lubricant give rise to intermediate variations of the frictional resistance. It has not hitherto been possible to say, in any case, just when and how the one form became transmuted into the other. These experiments show perfectly that method, extent, and rate of transformation for the cases taken, and illustrate satisfactorily the general law which I have desired to prove and to illustrate. The precise value of its constants is a matter of comparatively small importance. Those may be obtained later, and, we may hope, in ample quantity, and for all the usual lubricants. The main point is here established of the gradual transformation of the friction from that of fluid to that of solid friction, and the corresponding alteration of the laws of resistance.

In the light of these facts, and those of common experience, it becomes sufficiently obvious that the real problem of lubrication is not that of endeavoring to find ways of safely using oils at low rates of feed; but rather that of *finding ways of insuring such rates of feed as will make it certain that the lubrication shall be efficient in the sense that it shall insure fluid friction*, and its ordinarily comparatively insignificant resistance.

It is easy to see that our usual methods of lubrication are decidedly faulty, and our task is not to find a lubricating substance which, under these faulty systems of employment, will not permit a journal to heat, so much as to find ways of changing the methods of application, with a view to the introduction of and making standard better methods, and securing a better result in reduction of waste of energy and of lost work by this means. When friction, under ordinarily heavy loads, as in engine practice, is found to amount, as is often the case, to such figures as Morin obtained under the condition of his experiments, it is made our task to find satisfactory methods of reducing the figure to such as are given in railway practice on the best lubricated car-axle journals, or under the best conditions attainable in the use of the testing machine. True progress means steady approximation toward the best and the ideal conditions of most successful practice. The experiments here described show how far are the usual conditions of practice in some departments of engineering from the ideal, and it may perhaps be also said the practicable; and also how we are to go about the improvement of the case, within the limits of safety.

Prof. J. E. Denton.—It would be interesting to know what results would be obtained with the feeding device here shown if the siphonic head was maintained sufficiently constant to secure a constant rate of supply for several hours, and the pressure varied for intervals during this time. With the constantly varying rate of supply shown, it is doubtful if any condition of practice is copied. The rate of feed does not measure the oil present on the surface, unless it is shown that there is an escape of oil constantly equal to the supply. I understand from the paper that all escape of oil was prevented. If such was the case, the irregularities of friction, attributed to reduction of rate of feed, are probably due simply to differences of distribution of the oil, whereby the condition of the lubricating film changes from that yielding friction due purely to viscosity toward that due principally to metallic contact. The latter condition could be wholly realized only by maintaining a constant feed at the minimum rate for several hours. Morin's laws would then practically prevail. A rate of feed of .003 c.c. per minute per square inch of projected area is a liberal feed in many cases of practice. It corresponds to the use of three-quarters of a cupful of oil on a locomotive crank-pin per about 100 miles of service.

This is a greater amount than is necessary so far as satisfactory lubrication is concerned, and the conditions of pressure are probably more severe than in Mr. Bissell's experiments. It should not be concluded, therefore, that the irregularities of friction recorded in the experiments signify any dangerous condition of lubrication. I doubt the possibility of defining the rate of supply independently of the speed of rubbing.

Mr. Bissell.—It was not the intention to imitate any given practice in the experiments described, but to deduce the law governing the change from fluid to solid friction, and to demonstrate, by comparison of the results of those deductions with various practices, that Morin's laws do not prevail in efficient machines

It is evidently highly desirable that the laws referred to should not govern the action of machines when it is possible to reduce their friction to comparatively small amounts by rendering active those other laws of fluid friction.

CCCCXXVI.*

ROPE-DRIVING.

BY CHARLES W. HUNT, NEW YORK CITY.
(Life Member of the Society.)

THE transmission of power by cotton or manila ropes was a few years ago an experiment. It passed the experimental stage, and its use has spread with remarkable rapidity, both in England and in the United States, and promises, at the present rate of increase, to become a formidable competitor with gearing and leather belting for use where the amount of power is large, or the distance between the power and the work is comparatively great. The present article will not attempt to make a comparison with other methods of driving, but will endeavor to give some of the limitations of this method of conveying power, which will enable the engineer to decide at once upon the relative advantage or disadvantage of any proposed speed, with a close approximation to the true value.

The most prominent questions which the engineer wishes to have answered who proposes to make an application of rope-driving are those relating to the—

Horse-power,
Wear of rope,
First cost of rope,
Catenary.

These questions cannot be answered with precision in a general article, but it is the purpose of this paper to give the general limitation of this method of transmitting energy in order that an engineer may make a close approximation, and then, in the case of a large installation, make such supplementary calculations as the importance of the case demands. In this paper it will be assumed that the conditions are favorable to this mode of driving, leaving abnormal cases for such special treatment as the engineer may at the time think best to use. The subject is so new that but few accurate data are available, arising from the long

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

period required in each experiment, a rope lasting from three to six years. In many of the early applications so great a strain was put upon the rope that the wear was rapid, and success only came when the work required of the rope was greatly reduced. The strain upon the rope has been decreased until it is approximately known what it should be to secure reasonable durability. Installations which have been successful, as well as those in which the wear of the rope was destructive, indicate that 200 lbs. on a rope one inch in diameter is a safe and economical working strain. When the strain is materially increased, the wear is rapid. For the purpose of this paper, this strain of 200 lbs. for the normal fibre stress of a manila rope one inch in diameter will be assumed to be the one which it is advisable to use, and the horse-power and relative wear at different speeds will be estimated. Farther experience may modify this assumption, but at the present time it is the best obtainable, and the changes which may be made hereafter will affect but slightly the estimates here given.

To ascertain the average breaking strength of commercial rope, I lately purchased four pieces made by different cordage works, and had them tested on the Fairbanks' testing machine in New York. They varied slightly in diameter, but when reduced to the equivalent of a rope one inch in diameter, had an average breaking strength of 7,140 lbs. Expressed algebraically, the breaking strength, weight per foot, and the working strains are:

$$W = 720\ C^2 \qquad (1)$$

$$P = .032\ C^2 \qquad (2)$$

$$w = 20\ C^2 \qquad (3)$$

in these and the following equations:

C = Circumference of rope in inches.
D = Sag of the rope in inches.
F = Centrifugal force in pounds.
g = Gravity.
H = Horse-power.
L = Distance between pulleys in feet.
P = Pounds per foot of rope.
R = Force in pounds doing useful work.
S = Strain in pounds on the rope at the pulley.
T = Tension in pounds on driving side of the rope.
t = Tension in pounds on slack side of the rope.

v = Velocity of the rope in feet per second.
w = Working strain in pounds.
W = Ultimate breaking strain in pounds.

This makes the normal working strain equal to one thirty-sixth of the breaking strength, and about one twenty-fifth of the strength at the splice. The actual strains are ordinarily much greater, owing to the vibrations in running, as well as from imperfectly adjusted tension mechanism.

For this investigation we will assume that the strain on the driving side of a rope is equal to 200 lbs. on a rope one inch in diameter, and that the rope is in motion at various velocities of from 10 to 140 feet per second. Under this assumption, we will have in all cases a fibre strain of 200 lbs. on the driving side of a one-inch rope, and an equivalent strain for other sizes.

The centrifugal force of the rope in running over the pulley will reduce the amount of force available for the transmission of power. The centrifugal force of the rope is computed by the formula:

$$F = \frac{P v^2}{g} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (4)$$

At a speed of about 80 feet per second, the centrifugal force increases faster than the power from increased velocity of the rope, and about 140 feet per second equals the assumed allowable tension of the rope. Computing this force at various speeds and then subtracting it from the assumed maximum tension, we have the force available for the transmission of power. The whole of this force cannot be used, because a certain amount of tension on the slack side of the rope is needed to give adhesion to the pulley. What tension should be given to the rope for this purpose is uncertain, as there are no experiments which give accurate data, and at the present time a decision must be made partly from analogy and partly from experience.

If the rope be considered as a belt on a plain pulley, the friction would be substantially the same as a leather belt at the same tension, but as ropes are frequently lubricated to reduce the wear, the coefficient of friction must be materially reduced. There have been no experiments to decide with accuracy what this reduction is, but it is known from considerable experience that when the rope runs in a groove whose sides are inclined toward

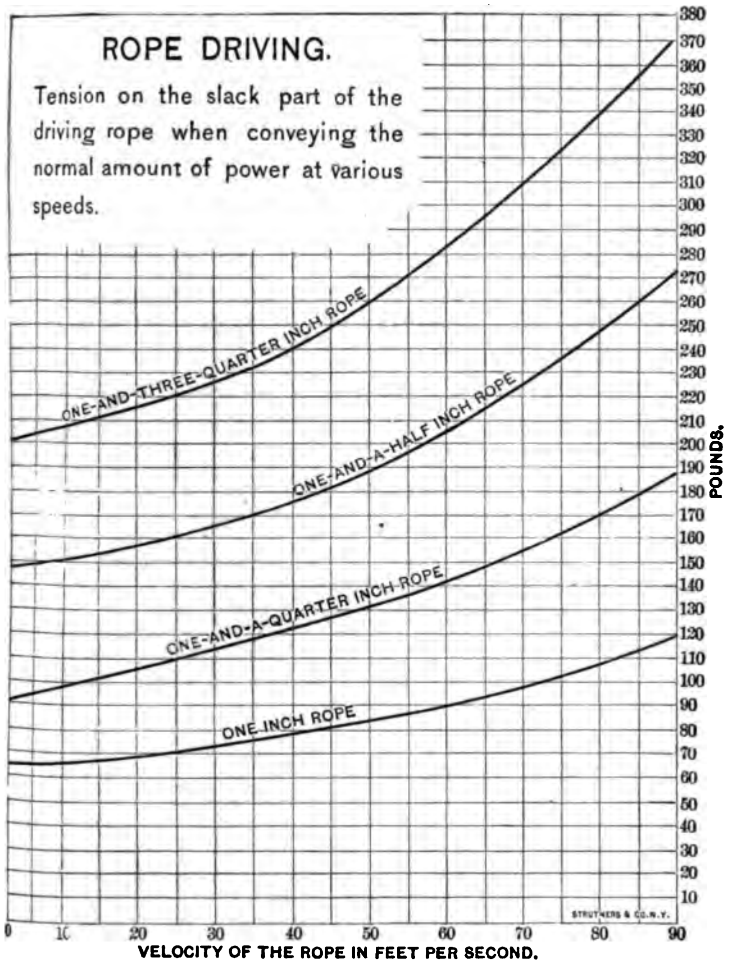


FIG. 56.

each other at an angle of 45° there is sufficient adhesion when the ratio of the tension is

$$\frac{T}{t} = 2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

For the present purpose, T can be divided into three parts :

Tension doing useful work.
Tension from centrifugal force.
Tension to balance the strain for adhesion.

The tension t can be divided into two parts:

Tension for adhesion.

Tension from centrifugal force.

It is evident, however, that the tension required to do a given work should not be materially exceeded during the life of the rope.

There are two methods of putting ropes on the pulleys; one in which the ropes are single and spliced on, being made very taut at first, and less so as the rope lengthens, stretching until it slips, when it is respliced. The other method is to wind a single rope over the pulley as many turns as needed to obtain the necessary horse-power and put a tension pulley to give the necessary adhesion and also take up the wear. The tension t required to transmit the normal horse-power for the ordinary speeds and sizes of rope is given in Fig. 56, computed by formula (8). The total tension T on the driving side of the rope is assumed to be the same at all speeds. The centrifugal force, as well as an amount equal to the tension for adhesion on the slack side of the rope, must be taken from the total tension T to ascertain the amount of force available for the transmission of power.

I have assumed that the tension on the slack side necessary for giving adhesion is equal to one-half the force doing useful work on the driving side of the rope; hence the force for useful work is:

$$R = \frac{2\,(T - F)}{3} \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

and the tension on the slack side to give the required adhesion is

$$\frac{(T - F)}{3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

Hence,
$$t = \frac{(T - F)}{3} + F. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

The sum of the tensions T and t is not the same at different speeds, as the equation (8) indicates.

As F varies as the square of the velocity, there is, with an increasing speed of the rope, a decreasing useful force, and an increasing total tension, t, on the slack side.

With these assumptions of allowable strains, the horse-power will be:

$$H. = \frac{2\,v\,(T - F)}{3 \times 550} \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

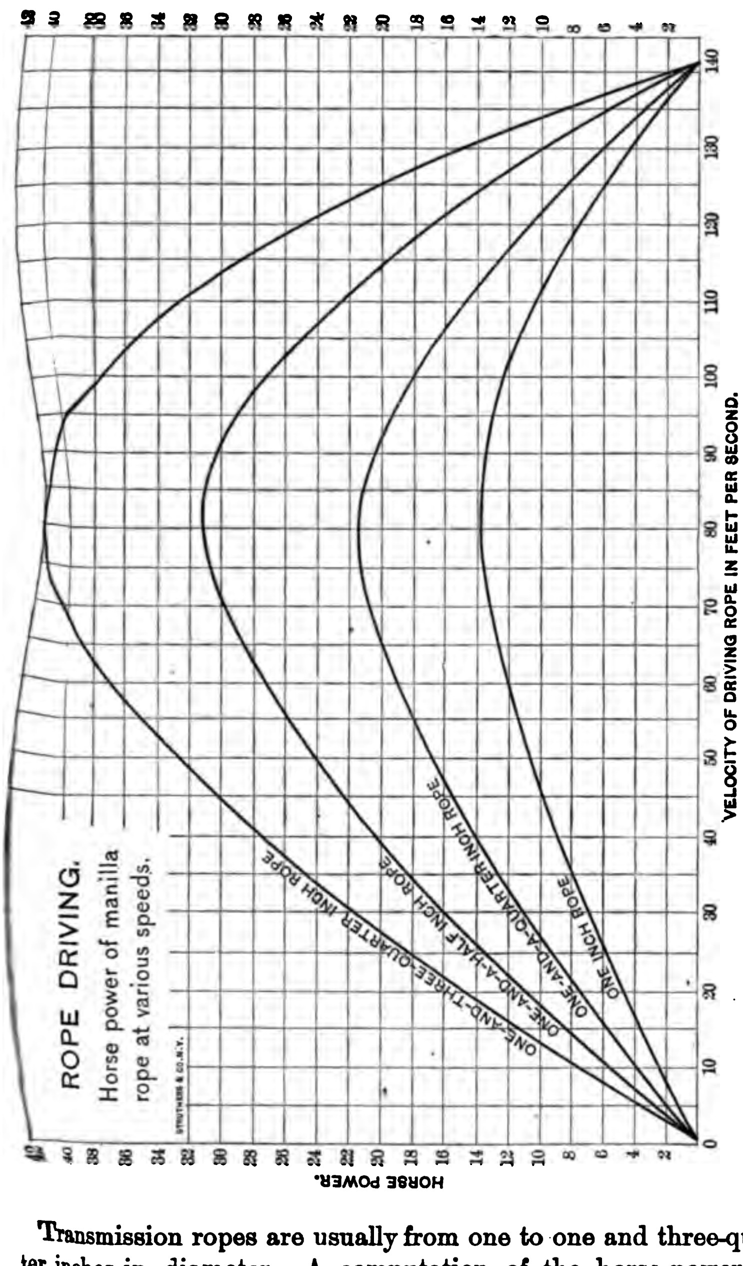


FIG. 55.

Transmission ropes are usually from one to one and three-quarter inches in diameter. A computation of the horse-power for four sizes at various speeds and under ordinary conditions, based on a maximum strain equivalent to 200 lbs. for a rope one inch in diameter, is given in Fig. 55. The horse-power of other sizes is readily obtained from these. The maximum power is transmitted, under the assumed conditions, at a speed of about 80 feet per second.

The first cost of the rope will be smallest when the power transmitted by it is greatest, and, under the assumed conditions, will be a minimum for a given power when the velocity of the rope is about 80 feet per second. The ratio of the first cost of the rope running at any other speed will be:

$$\text{Ratio of first cost} = \frac{\text{H. at 80 feet per second}}{\text{H. at required speed.}} \quad . \quad . \quad (10)$$

The curve in Fig. 59 shows the relative first cost of ropes to transmit a definite power, calculated for various speeds, with the coefficient of the cost assumed to be of such value that the cost of the rope running at 80 feet per second is 100. The first cost of the ropes running at different speeds is in proportion to the ordinates at those velocities.

The wear of the rope is both internal and external; the internal is caused by the movement of the fibres on each other, under pressure in bending over the sheaves, and the external is caused by the slipping and the wedging in the grooves of the pulley. Both of these causes of wear are, within the limits of ordinary practice, assumed to be directly proportional to the speed. Hence, if we assume the coefficient of the wear to be k, the wear will be kv, in which the wear increases directly as the velocity, but the horse-power which can be transmitted, as equation (9) shows, will not vary at the same rate.

If we divide the value for wear at a given speed by the horse-power which the same rope will transmit at other speeds, we get the relative wear of the rope in transmitting one horse-power. For this purpose, assume for a basis of comparison such value for the coefficient as will make the wear of a rope running 10 feet per second equal to 100. Fig. 58 shows the relative wear for transmitting the same horse-power computed by this method. The wear of a rope running at different speeds is in proportion to the ordinates of the curve at those velocities. The higher the speed, up to about 80 feet per second, the more power will be transmitted, but it is accompanied by a more than equivalent wear.

The rope is supposed to have the strain T constant at all

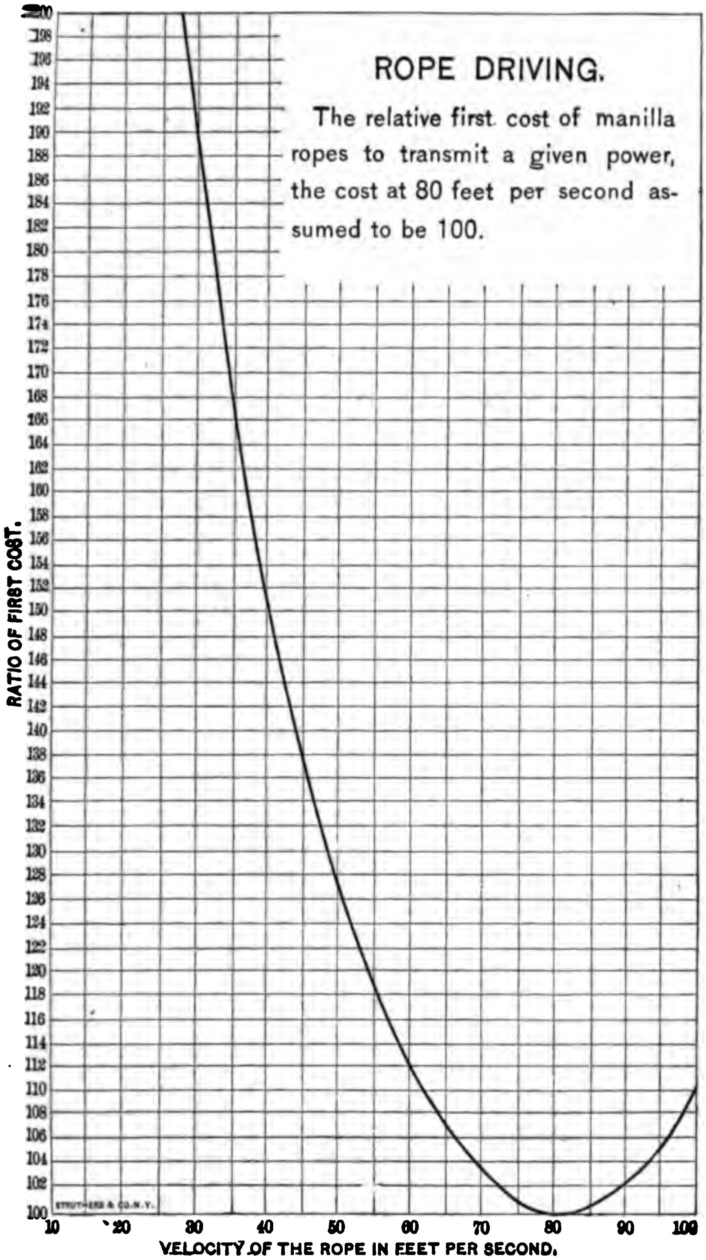

ROPE DRIVING.
The relative first cost of manilla ropes to transmit a given power, the cost at 80 feet per second assumed to be 100.
RATIO OF FIRST COST.
VELOCITY OF THE ROPE IN FEET PER SECOND.
200
198
196
194
192
190
188
186
184
182
180
178
176
174
172
170
168
166
164
162
160
158
156
154
152
150
148
146
144
142
140
138
136
134
132
130
128
126
124
122
120
118
116
114
112
110
108
106
104
102
100
10
20
30
40
50
60
70
80
90
100

FIG. 59.

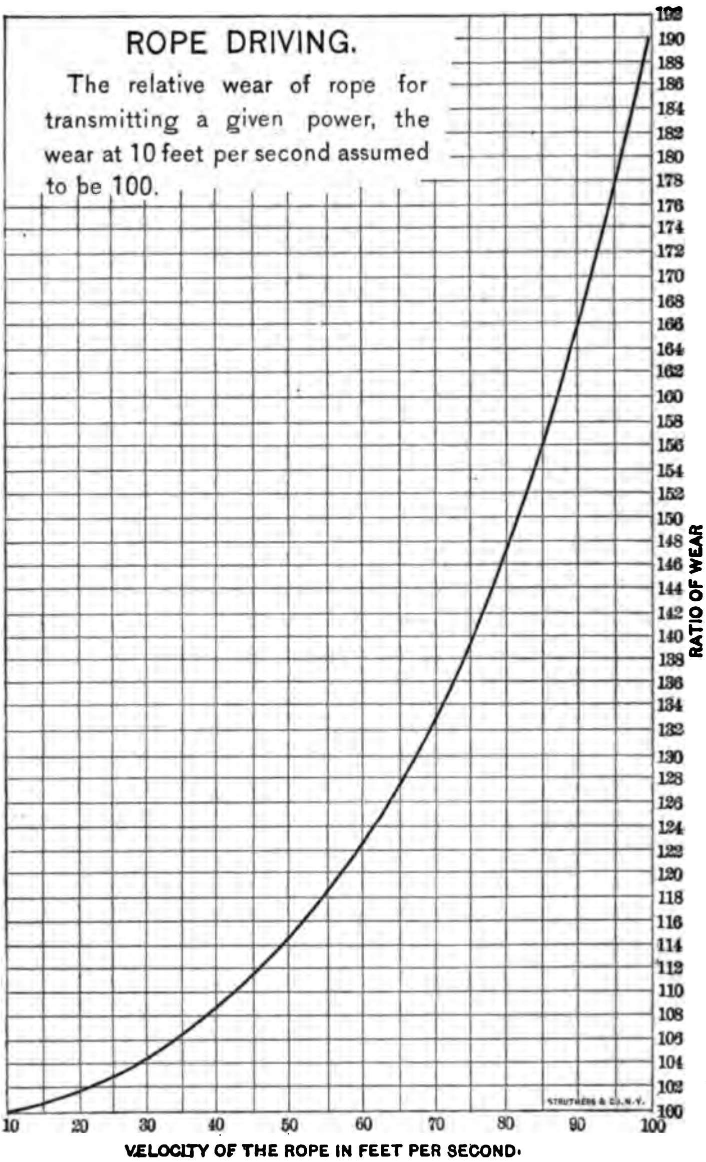


FIG. 58.

speeds on the driving side, and in direct proportion to the area of the cross section; hence the catenary of the driving side is not affected by the speed or by the diameter of the rope.

The deflection of the rope between the pulleys on the slack

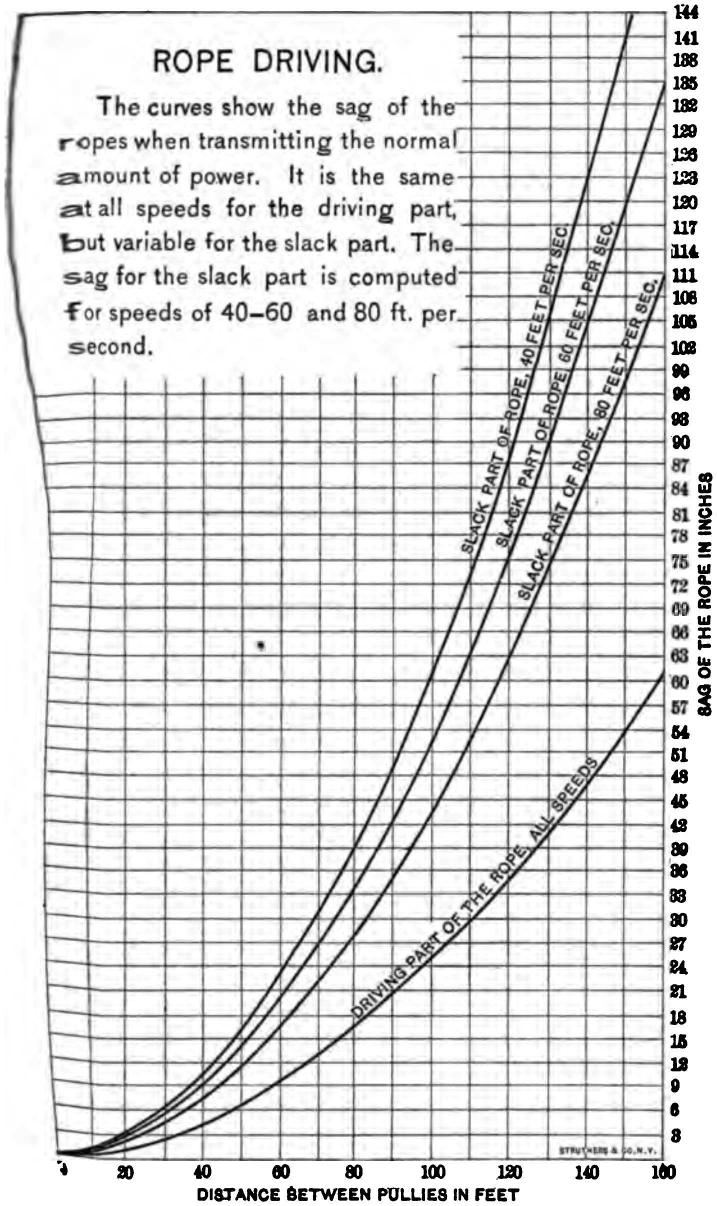


FIG. 57.

side varies with each change of the load or change of the speed, as the tension equation (8) indicates.

The curves in Fig. 57, giving the deflection of the rope, were computed for the assumed value of T and t by the parabolic formula:

$$S = \frac{PL^2}{8D} + PD, \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

S being the assumed strain T on the driving side, and t, calculated by equation (8), on the slack side. The tension t varies with the speed, and the curves, showing the sag of the rope in inches, are calculated for speeds of 40, 60, and 80 feet per second, and for spans commonly used in rope-driving.

It is to be regretted that accurate data are not available to determine the constants needed in the equations for wear and for friction on the pulley. No estimate has been made, as this paper is intended to state the general limitations of this mode of transmitting energy rather than to give details of its application. Enough is known, however, to enable the following points to be determined with reasonable accuracy for ordinary cases of rope-driving.

The horse power for all sizes of rope and all practicable velocities can be obtained from Fig. 55.

The tension on the slack side of the driving rope is given in Fig. 56 for usual sizes and speeds, from which the amount of counter weight, to give the necessary adhesion, can be deduced.

The sag of the rope between the pulleys is given in Fig. 57, for both the driving and the slack parts of the rope.

The comparative first cost of ropes running at different velocities, but transmitting the same horse-power, is given in Fig. 59.

WORKING TABLE OF THE HORSE-POWER OF TRANSMISSION ROPE.*

Computed by formula 9, Fig. 59, which makes the total strain on the driving side of the rope; when transmitting the normal power, the same at all speeds, and takes into consideration the effect of the centrifugal force in reducing the driving power of the rope.

---

* Added after the adjournment.

Diam. of Rope.	Speed of the rope in feet per minute.											Diam. of smallest pulley or idler in inches.
	1 500	2 000	2 500	3 000	3 500	4 000	4 500	5 000	6 000	7 000	8 400	
½	1.45	1.9	2.3	2.7	3.0	3.2	3.4	3.4	3.1	2.2	0	20
⅝	2.3	3.2	3.6	4.2	4.6	5.0	5.3	5.3	4.9	3.4	0	24
¾	3.3	4.3	5.2	5.8	6.7	7.2	7.7	7.7	7.1	4.9	0	30
⅞	4.5	5.9	7.0	8.2	9.1	9.8	10.8	10.7	9.3	6.9	0	36
1	5.8	7.7	9.2	10.7	11.9	12.8	13.6	13.7	12.5	8.8	0	42
1¼	9.2	12.1	14.3	16.8	18.6	20.0	21.2	21.4	19.5	13.8	0	54
1½	13.1	17.4	20.7	23.1	26.8	28.8	30.6	30.8	28.2	19.8	0	60
1¾	18.	23.7	28.2	32.8	36.4	39.2	41.5	41.8	37.4	27.6	0	72
2	23.2	30.8	36.8	42.8	47.6	51.2	54.4	54.8	50.	35.2	0	84

DISCUSSION.

Mr. Samuel Webber.— I agree fully with Mr. Hunt on the value of "rope-driving" in certain cases, especially when a large amount of power is to be transmitted a long distance, and more especially still for conditions which he does not mention; *viz.*, where the plane of motion of the pulley or shaft on which the power is to be received varies from that of the one from which it is delivered. It is well known that a "quarter-twist" flat belt loses part of its hold on both pulleys, while ropes in a grooved sheave lose no contact whatever, and by the use of proper idlers, or carriers, will transmit power perfectly to a line at right angles with the one from which it was delivered. Herein I conceive to be one of the greatest advantages of rope-driving. I am not yet convinced of its ultimate economy in cases where the power is to be transmitted in the same plane, and where belts can be used to advantage, for although the first cost of ropes is undoubtedly less than that of leather belts, their durability is less also; and as I know many instances of leather belts which have run perfectly more than thirty years, I have great doubts whether the renewal of ropes in that time will not bring their cost fully up to that of leather.

There is one point, however, on which I differ from Mr. Hunt. I do not think the angle of 45° the best for the grooves in the sheaves. Theoretically, an angle of 60°, or that of an equilateral triangle, should give the best holding *surface* for a circular rope, although it would not *grip* the rope as severely as a groove of a less angle. On the other hand, it would deliver it more freely from the side on which it leaves the pulley after passing the centre; and I have seen grooved pulleys at work where the rope wedged very tightly in the groove, and followed the sheave down three-quarters of a circle before it was released. This I think wears and chafes the rope unnecessarily. The best practical groove I have seen was an English one, a tracing from which gives me an angle of 53°, or just intermediate between 45° and 60°. Another fault which I find with rope-driving as introduced in this country is the use of manila rope.

I was taught in early life, by a brace of old sailor uncles, one of whom had circumnavigated the globe several times, "never to use manila rope for running rigging." "It was all right," they said, "for shrouds and stays; but for anything that was to go through blocks, use Riga hemp." A pretty long study of fibres has shown me the correctness of this advice.

The fibre of the "*Musa textilis*," or manila hemp, as used in ropes, is but a bundle, or *congeries*, of very fine fibres, held together by a vegetable glue, which by constant flexure cracks and disintegrates, and causes the rope submitted to such usage to give out very rapidly. Flax and hemp, and other similar vegetable fibres, are prepared for use by "setting," or rotting, in the first place to remove this glue; and fine flax for linen is ultimately spun through hot water, which thus reduces the fibre to its ultimate constituent. The fibre of cotton, which is a unit in the first place, is really one of the strongest fibres known when compared with its actual area, and has the advantage of great flexibility, enabling it to be used on smaller pulleys or sheaves than is possible with manila rope, and a cotton rope is therefore used in England for this purpose; and I have been recently told of one of the largest plants in the country, which started up two years ago with manila, taking it out and substituting cotton. Another trouble with the use of manila rope is the attempt to use it on too small pulleys, or idlers.

Our colleague, Mr. Cooper, in his book on the "Use of Belting," says that six feet is the minimum size of pulleys admissible; but, of course, this must bear some proportion to the diameter of the rope.

The English rule for cotton rope is from 30 to 36 diameters.

I was shown a six-foot leather belt a few weeks since, which I was told was transmitting 136 H. P. at 1¾ miles per minute, or about 9,000 feet. This agreed exactly with my published rule for the capacity of a double belt, and did not surprise me so much as the statement that the owners had gone back to leather after trying ropes unsatisfactorily.

An examination of the situation soon showed me the trouble. The power was transmitted from the ground floor of one mill to the second floor of the other one, and in so going the belt was bent twice at right angles, over two-foot diameter idlers. They told me that the first set of ropes lasted *one day*. They put on a second set themselves, which held ten days, and then went back to leather! One word more: Mr. Hunt's diagram, Fig. 55, gives the horse-power which a manila rope will transmit at different velocities, and I submit the following comparison with an English table I have of the power transmissible by a cotton rope at 50 feet per second, or 3,000 feet per minute.

	Manila.	Cotton.
1-inch Rope	10.75	10.50
1¼ "	17.50	19.50
1½ "	24.	30.
1¾ "	32.50	42.

I have seen some very admirable specimens of English cotton rope made especially for this purpose, and know that it can be readily made in this country, although I do not happen to have seen any samples. I consider it very far preferable to manila.

Mr. T. Spencer Miller.—Mr. Hunt's paper is a valuable addition to the literature on the subject of rope transmission, and there is certainly little room for contradiction.

However, Mr. Hunt, on page 231 of the paper, gives a formula for the strength of rope as 720 × Circ.^{2}.

The result of tests made by the government lead the writer to believe that while 720 is about the right *multiplier* for rope of 1 inch in diameter, that for 2 inches in diameter would be much smaller, and of ½ inch in diameter much larger.

The writer regrets his inability at this time to be more exact.

In England, hemp and manila ropes have been largely superceded by ropes of cotton, and I am satisfied one reason therefor is that *dry* manila ropes wear out too fast, while the *lubricated* ones give too low a coefficient of friction.

The angle of the groove at 45° has been used for 33 *years*, being first introduced by James Combe in Belfast, Ireland.

If we are to use tallow, lard, or other lubricated ropes, we certainly should use a sharper angle in the groove, especially in the

American system—that employing a continuous rope, and making many wraps.

Mr. Hunt's formula (No. 5) implies a coefficient of friction of .10. I have obtained a coefficient of .26, and have found one authority giving .28.

Reuleaux advises for single line transmission 30° angle of groove. Ramsbottom, an English engineer, and Yale & Towne use a 30° groove in transmission wheels on travelling cranes, and the writer hopes to see the best American practice use 30° or 35° as a standard angle for groove.

The effort to pull out a greasy manila rope from a groove of 30° is not worth consideration, although we hear a great cry about the loss of power on this account.

The writer is strongly in favor of using the *continuous rope system*, and of using *smaller ropes* than implied in this paper.

A factor of safety of 36 or 25 seems to me a *factor for blunders in engineering* rather than one for safety.

The most perfect small transmission the writer ever saw (about 20 H. P.) employed $\frac{5}{16}$ manila ropes on wheels 30 inches in diameter, using a tension carriage.

Rather than use so large ropes, the writer thinks it wiser to replace small ones oftener, for by so doing a great gain may be made in efficiency and a saving of fuel.

A large majority of the failures in the continuous rope plan has been when the driving and driven sheaves were of largely differing diameters, such, for example, as driving dynamos, and from engine fly-wheels to line shafts.

As ordinarily installed, the ropes will not pull alike, and by calculation and experiment we may find one strand of rope pulling twice or even three times as much as other strands. It is no wonder, if this cannot be cured, that engineers advise a factor of safety of 25 or 30.

An installation designed by the writer some two years ago, for J. K. Russell of Chicago, employs an engine driving sheaves about three times the diameter of the driven sheave. In order to cure the unequal pull on the strands of the rope, the grooves of the large driver were made 60°, and of the driven 45°.

This change of groove has proven a satisfactory cure for this particular plant, and is arrived at from the following course of reasoning:

It has been observed that in sheaves of the same diameter, by the use of a proper tension weight, the ropes may *pull alike*, while, where the sheaves are of unequal diameter, *they do not.*

The only difference of conditions in the two cases lies in the difference of arc of contact of the rope on the wheels, that of the large wheel being much greater than that of the smaller.

To produce the same conditions as obtained in transmission having sheaves of same diameter, we must either sharpen the angle of the small sheave or reduce the angle of the larger.

In dynamo-drives the writer would advise using 45° for the driving sheave, and as sharp as 30° in driven; of course the proper angle of driver must be arrived at by calculation, assuming a coefficient of friction of .20 as being about a proper medium.

The above improvement is the subject matter of a patent recently allowed the writer.

Mr. Scott A. Smith.—On his sixth page Mr. Hunt gives a diagram showing the power which a $1\frac{3}{4}$-inch rope would transmit, which amounts to 42 H. P., at a speed of 4,800 feet per minute. Now, a $1\frac{3}{4}$-inch rope occupies a space of $2\frac{1}{2}$ inches, including one-half of each flange, and that for a belt one inch wide would require that the belt should transmit horse-power with 285 feet of velocity per minute. Now, at the Hamilton Woolen Company in Massachusetts they used a two-ply belt for nine months which ran a horse-power with 268 feet of speed per minute. Now, if we take a three-ply belt, it is very safe to say that that belt will drive a horse-power at 285 feet per minute. I have here a letter from the Washburn & Moen Manufacturing Company of Worcester, showing that they are using four-ply belts. I speak of that to show the possibilities in belting. Now, parties wishing to drive a very large amount of power in the same space which ropes would require can use a three-ply belt and get the same amount of power, if I am correct, as I believe I am; and, if they are not satisfied with that, they can easily put a narrow rider on the top of the belt. The Washburn & Moen Company are using a belt nearly one inch thick. It is four-ply and 46 inches wide. Another belt which they are using is one inch thick and 38 inches wide, and that belt has run eight years, and they say it appears to be in good condition. This last mentioned belt drives a receiving pulley 10 feet in diameter. Within a few years a good deal of attention has been given to making better belting and also to seeing that a belt has full contact with the face of a pulley. In my own experience I have found that formerly, for instance, the fly-wheel of a pulley had altogether too much crown, and also would be frequently turned so that there would be ridges extending the whole width of the face on which the belt would rest. The practice now is to have the face of the pulley perfectly smooth. In reference to the low crown of pulleys in order to get the best tractive force, Charles T. Porter's custom was to make the face of his driving pulleys perfectly flat, and James Moore, of the Bush Hill Iron Works, Philadelphia, always ordered his pulleys to be made flat face. I have had a good deal of experience with quarter-twist belts, and if the belt were shaped especially for the purpose, I think it would work well. That shape is determined by some belt-makers by taking a belt which has been running a good many years driving quarter-twist, laying it out on the floor and seeing the shape the belt has taken by use, and then to make the new belt of that shape. They run very satisfactorily.

The President.—In what shape do you find them after use?

Mr. Smith.—That I cannot give satisfactorily. We leave it to the belt-makers to determine that. The belt tends to shape itself to the form which it should have.

Mr. Jas. McBride.—I would like to ask Mr. Hunt if he knows of any cases where power has been transmitted using the ropes crossed.

The President.—I want to ask if Mr. Hunt has had any experience with making the sides of the grooves concave rather than straight; and whether there is any advantage in making them so, and not leaving a space at the bottom for the rope to clear.

Prof. Jas. E. Denton.—Mr. Hunt has said a useful word regarding where rope comes in compared with belting. The rope comes in where belting works badly. Can he similarly sum up the relative importance of manila rope and wire rope? How does the transmission by manila rope have any advantage or disadvantage regarding wire rope in the same place?

Mr. Paul H. Grimm.—Having used both systems of rope driving, manila rope and wire rope, I think I can answer Mr. Denton's question more or less satisfactorily. A manila rope may be used to turn corners. You can turn at right angles or any other angle with it by using suitable wheels (idlers), and thereby avoiding the application of gearing or anything of that kind. I have never found a wire rope run satisfactorily over idlers, and therefore think that it is not practicable to turn a corner with a wire rope; and right there is where the application of manila rope comes in more particularly over wire rope. I have used manila rope for out-door driving, and have found it as durable as wire rope, doing the same amount of work under the same conditions as to speed.

*Mr. Hunt.**—I can make but a partial reply to the many points brought out in the discussion, for the lack of reliable data. Rope-driving is not a direct competitor with belting under the ordinary conditions of mill-work, as belting is by far the best and most economical material that engineers have yet found for the purpose. It is when belting does not work well, or when it becomes troublesome, that rope-driving comes in; and the more troublesome the belting is, the more splendid does the rope-driving seem. It is especially satisfactory where the power transmitted is large, otherwise requiring very wide or double belts, where the driving and driven shafts are at an angle with each other, where a vertical drive is required, when the distance is comparatively great, and when the speed of the rope is high.

In regard to the durability of manila rope as compared with hemp or cotton, I cannot speak from my own experience in rope-driving, but in coal hoisting there has never been a rope used which is at all to be compared with manila for durability, and, from analogy, I would suppose that for the similar, but lighter, work of rope-driving it would be equal, if not superior, to either hemp or cotton.

The strength of rope assumed in this paper is less than usually given in published tests; but they are a fair average of the strength of rope sold in the market, and such as a purchaser must expect to use.

In the discussion on the paper by Mr. Miller there is a slight misunderstanding when he says: "A factor of safety of 36 or 25 seems to me a factor for blunders in engineering rather than one for safety." In the paper under discussion I believe there was no mention of a factor of safety, for the advisable working strain to use depends largely upon other considerations. If a coefficient of safety could be ascertained of exactly the right amount when the rope is put on, the first day's use would so weaken it by wear that it should be removed. The rope must

* Author's Closure, under the Rules.

have a large surplus of strength for wear in use beyond the strength needed for safety; what this should be is a question upon which engineers differ. A high tension means a short life. If we had constants for the rate of wear, the first cost, the rate of interest, and cost of renewals, we could easily equate their factors and select the most economical size. While carefully watching for more light on these points, I believe the strains assumed in this paper are the advisable ones to use except for temporary installations.

In reply to Mr. McBride's question, I have seen cross ropes in use, and they work very much like cross leather belts; that is, very badly. The ropes were put in every other groove, and theoretically they would not touch, but practically they do chafe more or less.

Replying to the query about a concave groove, if it is concave, the largest part of the rope, when it is spliced, wedges in where the angle is the most acute and puts a very great pressure on it; and, as the failure of a rope is usually in the splice, I think that will increase the wear.

As to the comparison between manila rope and wire, manila rope transmission works well when exposed to the weather, and, what was unexpected, generally lasts longer and runs more smoothly than wire rope. I have not investigated the matter enough to give an opinion for which I would be willing to be held responsible in the future, but my present way of accounting for the fact is that there are two causes of abnormal wear with wire rope which are absent in manila. Wire rope used for transmission is not, in practice, lubricated in such a way as to have the lubricant reach the wires where they rub against each other in the rope, and consequently they are chafing each other dry. The second cause of wear is on account of the sheaves being filled with a soft substance to prevent wearing the rope. This frequently gets uneven and causes short bends in the rope when in contact with the pulley and vibration in the rope between the pulleys.

Manila rope is usually lubricated internally, and the sheaves are turned true and always remain so.

CCCCXXVII.*

ACCIDENT-PREVENTING DEVICES APPLIED TO MACHINES.

BY JOHN H. COOPER, PHILADELPHIA, PA.

(Member of the Society.)

When considering the possibility of damage to any machine while in service—an event likely to happen, as all experience teaches—the application of preventives would seem to be desirable. To provide machines with safeguards at every danger point, having for their object the protection of the attendant against injury, would seem equally desirable. When the dangers incident to rupture are very great, extraordinary precautions become necessary. The law-makers, not satisfied with the security afforded by one safety-valve on each independent steam-boiler, which may become inoperative by defect or neglect, have ordered *two* of good make, and that they shall be tested each working day, thus making surety doubly sure.

A well-planned legal act may have good effect; as, for instance, greater safety in the use of steam-boilers and economy of pump repairs are gained by the use of independently driven feed-pumps; and the passage of this part of the Philadelphia Boiler Ordinance has wiped out the excuse of engine attendants, that low water was caused by their inability to keep the feed-pump in good working order, because it was attached directly to the engine, and therefore was constantly running and as rapidly wearing out its packing and itself.

Every imperfect machine is liable to accident, which must happen sooner or later. It is like a strong chain with one weak link, whose strength cannot be counted as more than that of its weakest part. The nearer the approach is made to uniformity of strength and capacity for wear in all parts of a machine, the further away is accident removed from that machine. In the use of materials for any machine or structure, care must always be taken

* Presented at the Richmond Meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

to not exceed, by the stresses applied, the " elastic limit ;" safety is only to be secured and preserved by keeping fully within this boundary.

In cases where breakage is imminent from the nature of the work to be done and the circumstances of use, a certain part of the machine is designed to receive a " breaking piece " of trifling cost, which will hold the parts in place for all regular work, but will yield to unexpected strain and give warning of its failure, as, for instance, the delivery spouts of grain drills, which may at any time strike a hidden root or stone, when a frail wooden pin yields to the pressure and saves the machine.

Instantaneously-acting stop-motions may be mentioned as live precautions, such as are applied to card-making machines, to weaving looms, and the like, for effecting a sudden stoppage when the card wire has gone astray, or when the tardy shuttle has failed to pass the web. Such also are relief-valves applied to the cylinders and pipes of water pumps and to steam-motor cylinders, for sudden relief from destructive strains due to confined water.

The safety pinion of Mr. Fogg, of Waltham, Mass., is of this character, and consists in attaching one of the pinions of the train to its arbor by means of a screw-thread, so that, when it is driven in the direction it is intended to run, it will be down in its place and in gear, but in the event of the breakage of the mainspring, the force of the recoil will cause it to revolve in the opposite direction, when it will rise on its arbor out of gear with the wheel into which it takes, thereby avoiding all liability of derangement of the train of gearing.

Some machines are destined to early destruction through lack of parts which should be designed for yielding to sudden strains and blows.

The writer, some years ago, perfected a machine for beating gold, in which all previous attempts had succeeded only in beating itself to pieces. The solution of the problem was completely accomplished by judiciously connecting together, with elastic intermedia, all the parts of the machine concerned with the reciprocating movements of the hammer, and these interposed cushions were made in simple form, of such common material as wood and leather.

All engineers who have grappled with the problem of large steam and power hammers know the value of the knowledge which enables them to apportion and locate aright the elastic and the rigid parts.

With the lack of precision in certain moving parts of a machine and of means for adjustment of the same when worn or deranged, there results injuries to delicate and costly parts which are beyond repair.

For machines in which a punch shall unerringly enter a number of dies in swift succession and never strike their edges while entering (which would irreparably injure them), the movements being necessarily intermittent, with measured intervals of absolute rest between each movement, during which the die is immovably locked in correct position for receiving the punch, the mechanical movement shown in Figs. 60, 61, 62, and 63 was designed and built by the writer as long ago as 1871. This represents a type

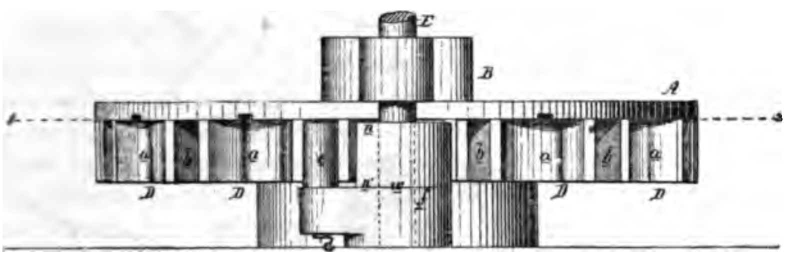


FIG. 60.

of a perfection of contrivance which secures immunity from self-destruction while in action.

It consists of a wheel or disk, *A*, arranged to turn upon a shaft, *B*, as shown in Figs. 60 and 61. To the under side of this disk is attached, by screws, a series of blocks, *D*, all alike in shape, and arranged in a circle concentric with the disk, and at equal distances apart from each other, the blocks being so formed that there shall be between them spaces *b*, each space having opposite parallel sides *xx* (Fig. 61), which are also parallel with a radial line, *g*, drawn from the centre of the disk midway between the said sides *xx*. *E* is the driving shaft, to which is secured an arm, *G*, having near its outer end a pin, *d*, carrying a cylindrical roller, *e*, which is of such a diameter as to fit snugly in but pass freely through any of the spaces *b*. The hub *f* of the arm *G* is of the peculiar segmental form illustrated in the drawing, being cut away on the radial lines *mm'*, Fig. 61, and from the upper end *n*, Fig. 60, of the hub to the line *n'*; the periphery of this segmental hub is adapted to a segmental recess, *a*, formed in each of the blocks, and each recess is cut away at *p*. The arm *G* is supposed to be turning continuously in the direction of the arrows ; and, as seen in Fig. 61, the roller of the arm is on the point of entering the space *b'* ; on the further turning of the arm the disk *A* must necessarily be turned in the direction of its arrow, the corner *i* of the segmental hub *f* at the same time receding, so as to offer no impediment to the free turning of the disk. By the time the arm *G* has reached the position shown in Fig. 62 the disk will have completed one-half of one movement, and the corner *i* of the segmental hub will

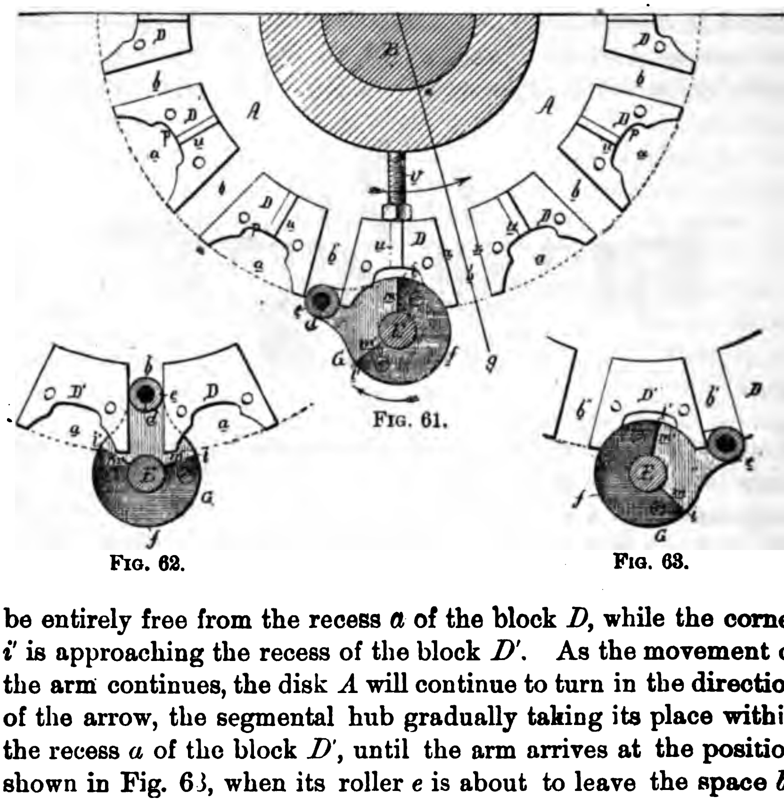


FIG. 61.

FIG. 62.

FIG. 63.

be entirely free from the recess *a* of the block *D*, while the corner *i'* is approaching the recess of the block *D'*. As the movement of the arm continues, the disk *A* will continue to turn in the direction of the arrow, the segmental hub gradually taking its place within the recess *a* of the block *D'*, until the arm arrives at the position shown in Fig. 63, when its roller *e* is about to leave the space *b'*, after which it can have no further control of the disk, which is locked by the segmental hub, owing to a portion of the latter fitting the recess *a* of the block *D'* ; and the hub, as the arm continues to turn, will continue to retain and lock the disk until its roller *e* enters the next space b^2, Fig. 63, when the disk, freed from the control of the hub, will commence a second movement. It will be seen that the disk is controlled in its movements by the roller of the arm G, and that the locking of the disk is effected by the segmental hub, the several parts being so accurately arranged in respect to each other that the disk is never free from the control of the hub until the roller is in a position to turn the disk; in other words, as the arm G revolves the disk is either being moved by the roller of the arm or is locked by the hub of the same. The recesses p in the recesses a of the blocks are necessary for permitting the corners i and i' of the hub to take the course pointed out in the said recesses a. Although the segmental portion of the hub may form a part of the arm, it may be made separate, as shown by the line w, and to secure it to the arm by suitable bolts and screws, so that the said segmental portion can be adjusted at pleasure, or renewed when worn, to be replaced by a new segment.

The above-described movement is applicable to many machines, and under many circumstances, but one example of its applicability will suffice to illustrate the importance of the invention. In moulding soap, crude blocks of that material are placed in chambers of the proper size and configuration for imparting the desired form to the block, and the pressure on the latter is accomplished by means of a heavy falling plunger, which must enter and exactly coincide with the chambers. The chambers may be formed on the disk A, and be arranged radially on the same or a circle concentric with the axis of the disk, and one plunger may be raised and permitted to fall successively into every chamber containing the soap. It is of the greatest importance that the disk should be immovably locked and in the proper position during the falling of the plunger, so that the latter may coincide with the chamber which it has to enter; it is also important that, as the plunger is elevated free from the chamber, the disk should be promptly and accurately moved to the desired extent for bringing another chamber directly beneath the plunger. It will be evident that the above-described mechanical movements will meet all these requirements.

The blocks D are made adjustable to wear and precision, as shown in Fig. 61; for instance, where each block has a rib, v, adapted to a radial guiding groove in the disk, and each block can be moved by means of a screw, v, and secured after adjustment by screws. The roller at the end of the arm can be readily removed, and replaced with such a roller as may coincide with the spaces between the blocks, these spaces varying in width in accordance with the adjustment of the blocks.

Many serious accidents occur from motor engines running away when the governor belt slips or slides off its driving pulley. These may be prevented by the duplication of the valve disks, the invention of the writer, and shown in Fig. 64. The ordinary balanced governor valve *A* is provided with two extra disks *BB*, which do not interfere with the passage of steam to the cylinder when the governor balls are revolving in the proper working plane; but when they fall by loss of circular motion, the disks *BB* are

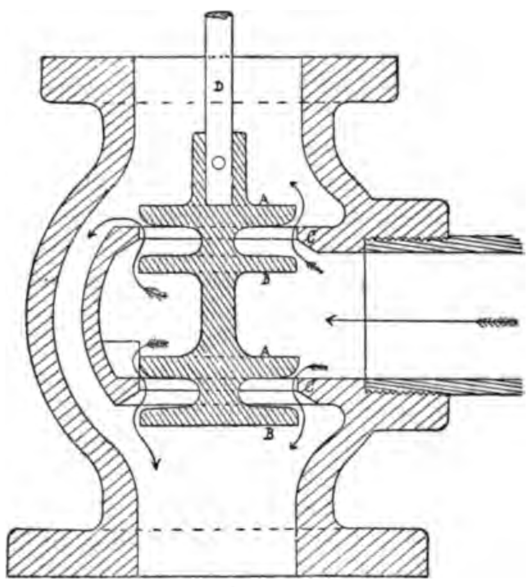


FIG. 64.

drawn into the ways of the steam and effectually stop its flow, whence the engine comes to a standstill.

Loss of life or limb to an operative very often means the loss of a skilled "hand" to an important industry, as well as the loss to a family of its chief means of support. It may add to the burdens of public institutions and may lessen the productive power of our mills. Therefore, the prevention of accidents, in a humanitarian sense as well as in a commercial one, should be a matter of deep interest to us, demanding the exercise of ingenuity in the devising of effective means for securing this end.

The subject is a broad one, and this paper proposes to merely touch it at a few salient points, and there leave it to the humanely wise, to whom a hint is sufficient.

On the well-recognized principle that self-preservation is the first law of nature, the duty of establishing and, maintaining protective appliances wherever necessary about dangerous machinery becomes apparent, and the effort to do so is worthy of praise, since we find that nature has well considered this necessity, having organized in our being ample means and methods of giving warning of approaching danger, and of guarding us against injury, which are at once effective and beneficent.

Almost every machine has places about it which will pinch the flesh, mash a finger, or crush a bone, and it will perform these cruel acts upon any one who thrusts his members into its open jaws.

With the exercise of ingenuity and care for protecting the machine against injury to itself, the attendant may be protected also, and if proper safeguards be thrown around a machine, with ordinary care and with instructions that may be given the attendant can perform all the work necessary about and with the machine with perfect safety to himself, thus preserving machine and man; while the employer, who may be held responsible for accidents occurring from exposed machinery which might have been protected at little cost, will have performed his whole duty.

A great number of accidents is caused through loss of presence of mind by the eagerness of work-people to get through their work expeditiously, under threat of punishment if they do not. The largest contributions to the list of accidents are amputations and maimings of dexter hands, due to this eagerness. If urging must be done, these results of experience call loudly for safeguards and warning mechanism, and the enforcement of stringent rules for safety, which shall be binding alike on employers, on foremen, and on the hands themselves.

Dr. G. C. Swallow (inspector of mines for Colorado), says: "The greatest number of accidents on the hoisting cage happen from neglect of ordinary caution. It is noteworthy that visitors to mines do not come to grief. It is always the men best acquainted with the workings who tumble down winzes or absent-mindedly ring the wrong signal. Careful as a miner may be, there comes a moment of forgetfulness, which is perhaps the critical moment for him." "After all, the great safe-guard against mining accidents is the ever-present realization of danger, and the consequent habit of caution. On this ground some mining men condemn safety-catches to cages and devices to prevent overwinding, thinking it better to trust to the watchfulness of the engineer than to any mechanism which is liable to rust and to fail just at the time—one instant out of hundreds of days—when it should come into play."

Dr. Swallow suggests that mine cages be completely enclosed by strong wire netting of a mesh too small for the hand or foot to pass through, in order to prevent accidents to these members while passing the framing of the cage ways. The top should be well protected by similar means against accidents from objects falling down the shaft upon the cage. Other means of convenience and safety may be noted in this connection, as, for instance, sliding doors of light, open framework under control of an operator, may be fitted where needed for quick movement. A free space of 2 inches or more allowed all around an open hoisting cage has been insisted upon to prevent damage to the inadvertent foot projecting over the platform. Of all automatic mechanism it may be said: Those combinations which are much used need constant looking after so as to avoid accident by the failure or derangement of excessively worn parts; and those which are little used, to avoid accident by non-operation when wanted.

To prevent accidents* happening from belts winding around shafts or couplings of the same, simple and cheap hooks or belt-rests may be devised for and attached in the needed places. They may be located as at C', Fig. 65, to prevent a belt from

* The writer has been favored with the perusal of a copy of a work published by the "Society for the Prevention of Accidents in Factories," illustrated by 43 plates, with tri-lingual text. This assemblage of appliances and apparatus proves on examination to be a veritable encyclopædia of ingenious and practical means for the prevention of accidents in factories, and of injury to the operatives working therein; it embodies the full record so far as completed by this committee, up to the year 1889, covering twenty years of active existence.

The self-appointed task assumed by this committee having already accomplished remarkable results, seeks, by a well-directed public spirit, through this publication, to diffuse them among manufacturers everywhere.

This publication, from which aid has been received in the preparation of this article, is, therefore, recommended to the attention of all those who desire to perfect the safety side of machines, as well as those who take more interest in the manufacturing problems of the day than in the mere mercenary outcome of them. J. H. C.

sliding off the pulley, and others, as *CC*, to hold the belt in safe position when off.

In the belt carriers of Mr. Biedermann, which are formed of a bent flat iron bow, *C*, Figs. 66 and 67, generally concentric with the rim of the pulley and around, say four-fifths of its diameter, and carry four to six hook-bolts which project in a circle a few inches less than the pulley rim, the first bolt being situated at a point where the belt is taken up, and the last where it is paid out. When the belt is dismounted it is supported by these bolts,

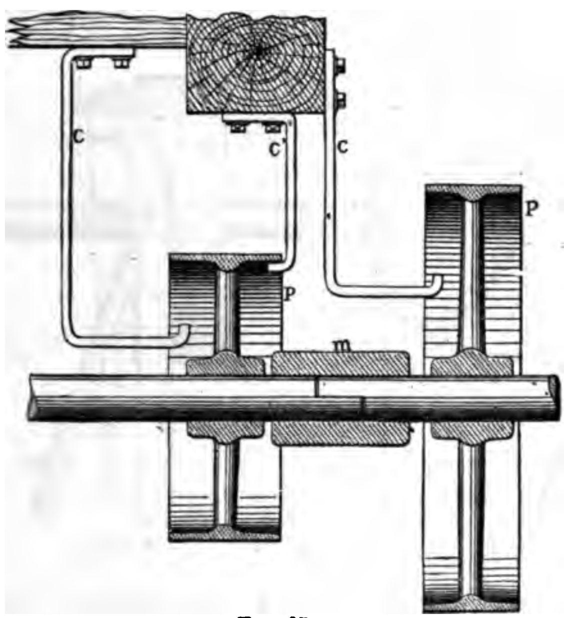


FIG. 65.

retaining a curved position, and is ready to be remounted; it is enough to lift it gently with the long-arm *Q*, surmounted by a curved finger, and bring it in contact with the pulley, when it is immediately carried away upon the pulley *P* without the least exertion.

In the case of a horizontal belt, which is taken up at the top of the pulley, the appliance is fitted rather higher, so that the first bolt may be brought as close as possible to the rim of the pulley; this bolt being removed from *a* to *a'*, to allow the long-arm room to guide the belt and to set it in its real position on the pulley. See Fig. 68.

F is a light fork of hard wood, which prevents the appliance from being bent against the shaft when the belt happens to repose upon it. This fork does not touch the shaft when the belt is on the pulley. When the belt is horizontal and taken up from underneath, the same kind of appliance is used, but in a reverse manner. Oblique belts whose angle does not exceed 45° may retain the appliance in its concentric form; but when it

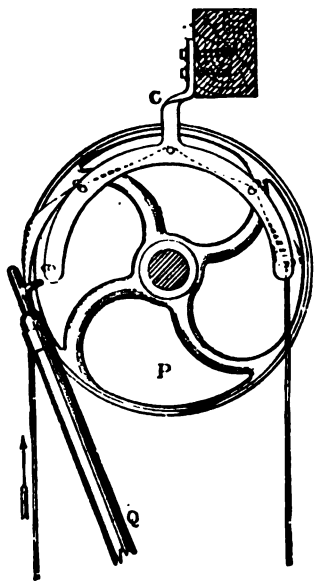


FIG. 66.

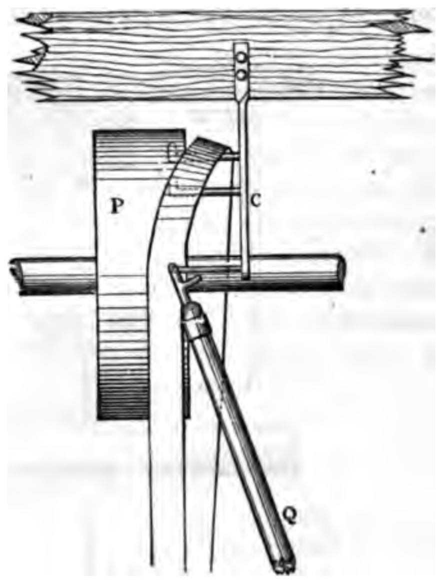


FIG. 67.

exceeds 45° the form adopted for horizontal belts should be used. Workmen frequently indulge in the bad habit of taking both ends of a belt (which is off the pulley and resting on a running shaft) in one hand and of pressing them together, in which case a fold is apt to be formed on the slack part which is liable to be drawn in under the tight part, and the whole be rolled up rapidly on itself on the shaft, with risk of accident to any one who may get caught in its drawing up folds.

Many accidents occur from ropes and chains running off their sheaves; an idea is given in Figs. 69, 70, 71, and 72, of guards applied to such sheaves.

These may be modified in a multitude of ways, according to the purpose to be served, whether as a protection to the hand, where the rope must be taken hold of, or as a safety device to the machine itself, for hoists, cranes, and the like, where dislodged ropes would wreck the machine.

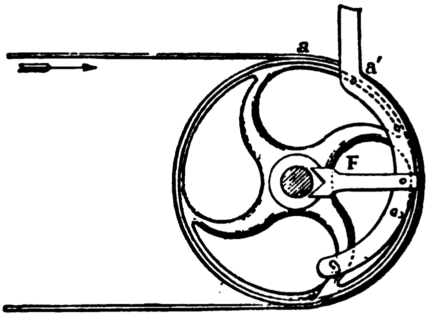


FIG. 68.

To prevent rolling wheels from running over any object in its way, a rail guard, shown by Figs. 73, 74, and 75, is provided and attached.

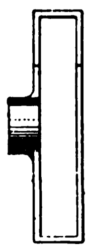


FIG. 69.

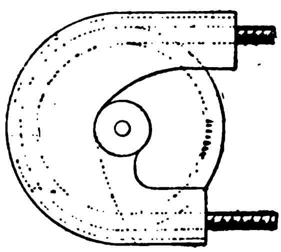


FIG. 70.

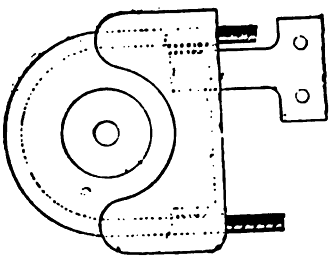


FIG. 71.

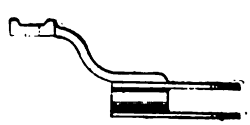


FIG. 72.

Devices such as these for preventing accidents to machines and their attendants might be reproduced almost indefinitely.

As new "hands" are coming into mills all the time, who are presumably ignorant of the dangers surrounding them and of the constant care which is necessary to be exercised over their persons at all times, and so may meet with accident from such causes, it is desirable to have plainly posted rules which, if enforced, might enable them to avoid injury.

A rule forbidding the cleaning of shafts and wheels, while in motion, with "waste" or rags taken in hand, would save many of these all-important members from mutilation or loss.

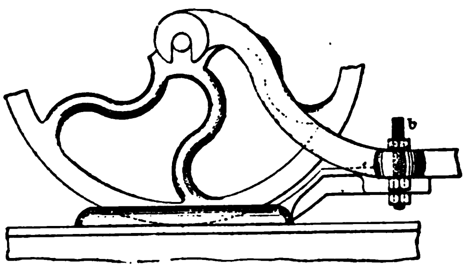


FIG. 73.

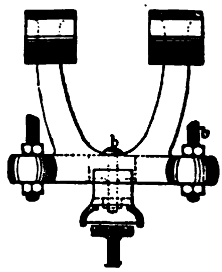


FIG. 74.

Stringent, clearly understood rules in regard to starting the motive engine, which should be done only on well-determined signals in cases where repairs are going on in any part of a mill, that no accident may occur to the men at work upon standing machinery; and, conversely, plainly visible and easily accessible alarm apparatus should be located at proper places in all the

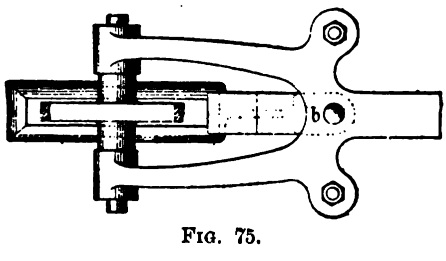


FIG. 75.

work-rooms, which are to be used in cases of accident for signaling the engineer to stop the motive engine at once.

Much has already been done, but much yet remains substantially to protect the attendant against projecting wedges, keys, set-screws, nuts, grooves, and the like of machinery in motion, which are so liable to catch his clothing and scarify his flesh.

Many serious accidents have happened from this source, and the exercise of common ingenuity with uncommon vigilance seems necessary to reduce their number.

The usual methods of protective boxing to belts, ropes, shafts, and pulleys, where they pass through floors, should receive attention, as their importance demands, and there are peculiar cases of these parts of transmitting machinery which need critical looking after.

The engineer on duty must not be permitted to wear garments with loose fluttering ends; the same advice may be extended to all attendants, male and female, and to workmen on machine tools or any rapidly going machines.

The subject is an important one, not only in its humanitarian aspects, but in its promotion of increased production, by insuring full confidence to every operative that there is no such thing as harm in any machine, and that accident to man and machine are alike nearest the impossible.

Much may be said of the grandeur of our industrial systems and machinery, of the excellence of our contrivances, which will do almost everything except think and talk, yet with their development and use comes a long list of distressing accidents, apparently beyond our power to prevent.

Let us hope that from another presentation of this subject, a thought will be added to the many with which the engineer has to deal, and that he will employ it for the safety of machines and of men; that the growing intelligence of workmen will lead them into habits of greater watchfulness while in places of danger and to a more prompt obedience to orders respecting the care of machines as well as of themselves; and that masters will take a "holding turn" in the consideration and application of protective devices wherever necessary.

DISCUSSION.

Mr. J. L. Gobeille.—There is one line of machinery which is certainly more imperfect than anything else made in this country in this matter of danger in use, and that is the tool known as the jointer or buzz planer. The accident which usually happens is that the man, in running a small piece of wood over the knives, drops his left hand into the cutter-slot. That takes off his three last fingers. Or else he puts his thumb or the fore-finger of his right hand into the slot and loses that. So far as I know, there has never been any accident-preventing device for this tool which has been at all practicable. The only preventive which I have found for this class of accidents is to forbid a man from using a machine when he appears to be careless. It not only occurs in the largest sizes but also in the smaller sizes. I know of one 24-inch machine which is responsible for perhaps sixteen or eighteen fingers, and parts of hands, and it has been necessary to stop the use of it because men would get careless after using it for a year or eighteen months, and in an unguarded moment, generally from seven o'clock to eight o'clock in the morning, they would thrust their fingers into the slot. Another wood-working machine which is extremely dangerous is the ordinary circular saw with tilting table. All the accidents which I have observed, or of which I knew the circumstances, arose when the man in sawing put his hand back of the saw. That takes the hand between the fingers and cuts it up through the palm. This kind of accident has occurred several times under my own observation. It is a very serious accident, and difficult to cure without amputating one or more fingers. Another serious accident occurring in wood-working shops is to have a piece of wood caught on the top of the saw while running at perhaps 3,000 revolutions, and which hurls the piece with an impetus of perhaps $2\frac{1}{2}$ miles per minute. It catches a man either in the chest or the abdomen. That means, it knocks him senseless; he is partially paralyzed—his extremities being in a tremor for two or three days. I have had two accidents of that kind. One was the case of a young man nineteen years old, who was in this state a week. This young man was struck by a piece of pine $3\frac{1}{2}$ inches long, about 4 inches wide and $1\frac{1}{4}$ thick. When we showed this piece of wood to our attorneys, anticipating trouble, they laughed at the idea that such a fragment could hurt a man seriously; but it nearly killed this young man, and it did it without any abrasion, simply leaving the mark of the corner of the piece of wood just below the chest.

The President.—I knew of a case very similar to this, where a contractor, a well-known man, a very skilful builder, living near me, attempted to put a piece of ordinary one-inch pine board 10 or 12 feet long against a circular saw to rip it lengthwise. It came back and struck him in the abdomen, knocking him senseless, and so injuring him that he died a few days afterward. In another case a small piece of wood flew back from a saw and cut a man's thumb very badly. I believe indeed that this is a very frequent form of accident.

*Mr. Jno. H. Cooper.**—There can be but one opinion expressed while discussing the subject of this paper regarding accidents to running machines and to the men who operate them.

Every engineer can relate in detail numerous cases which have occurred during his experience, but few care to bring such company home to their feelings, and therefore desire neither retrospection nor repetition of these painful happenings.

It is an easy task to formulate a plan of accident-preventing devices after the harm is done, but the wiser engineer foresees the possible weakness, as well as the dangers ahead, which are involved in his new enterprise, and at once embodies all the necessary means of safety in his original design.

The writing of this article has for its object the furtherance of this latter idea.

Fortunately the American mechanical engineers are not bound by such laws as we are told the ancient Hindoos have written in their books, which had the effect of closing the gates of progress. "Under pretext of care for the creature, their authors imposed the fatal principle, that a man must not address himself to discovery or invention, as Heaven had provided him all things needful."

Our more practical rule—"Take care of yourself"—brings best results when faithfully carried out, and its force is not weakened by the addition of helping to care for others, who, lacking experience, may profit by advice or material methods placed where necessary, especially when we have come to know the meaning of the hard truth expressed in the doggerel of *Hudibras*:

> "Ah, me! what perils do environ
> The man that meddles with cold iron."

To the consideration of the two features referred to, a third may be named, and that is upon whom the responsibility falls when accident occurs, which may happen from want of proper safeguards where machines are dangerous or carelessly constructed.

More than twenty years ago it was said by M. Dollfus, president of the Mulhouse Society for the prevention of accidents

* Author's Closure, under the Rules.

in factories: "The manufacturer owes his workmen other things besides merely their wages." Among these will surely be included their personal safety, secured to them from the moment they enter his service.

To argue the right and need of such care-taking, in the light of existing law, would be as throwing words away, and whatever may be said for or against the objects and aims of any engineer who is devising means and methods for saving the limb or life of a fellow creature while engaged in the service assigned to him, he is without question contributing to an extraordinary and benevolent work.

CCCCXXVIII.

*A NEW PROCESS FOR GENERATING AND CUTTING THE TEETH OF SPUR WHEELS.**

BY AMBROSE SWASEY, CLEVELAND, OHIO.
(Member of the Society.)

THE theory of the interchange system of gearing according to the solution first given by Professor Willis, in his treatise on the *Principles of Mechanism* is, "that in a set of wheels of the same pitch, having a constant generating circle for the flanks and faces of the teeth, any two wheels of the set will work correctly together;" and as a rack is a gear so infinitely large that its periphery forms a straight line, it follows that, if the rack teeth are also described with the same circle as that used for the wheels, any one of the set will run correctly with the rack.

The diameter of the generating or describing circle which gives one of the best forms of teeth for a set of wheels is equal to the radius of the pitch diameter of a 15-tooth pinion, making a 7½-inch generating circle for one diametral pitch. The flanks of a 15-tooth pinion being radial, a 12-tooth pinion, which is the smallest generally used in practice, will have the flank of one tooth nearly parallel to that of the tooth following, thus allowing the space between the teeth to be cut with the regular gear cutter.

It was thought at the time this system was first brought out that its use would be very limited, as in the case of change gears for lathes, and where it was found necessary to construct a train of gear wheels; but time has shown that it has many advantages over other equally correct systems not on the interchangeable plan, and it has been constantly growing in favor, until it is now almost universally adopted, especially among those using cut gearing.

Prof. Edward Sang has also solved the same problem in

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

another way, by taking the rack for the foundation of all curves for a set of interchange gear wheels. In the Sang theory, instead of using a constant describing circle, the rack acts as a constant generator, and the faces and flanks of all teeth throughout the whole set of wheels are described by it; therefore it is evident that, if the constant generating circle which is used for a set of wheels according to the Willis theory is also used to form the teeth of a rack, this same rack can be used in accordance with the Sang theory to generate another set of wheels, and any gear of one set will work correctly with the other set.

For many years, one of the foremost mechanical problems in gearing has been to reduce these well-known theories to practice, and produce gear teeth having contours which correspond as nearly as possible to the theoretical curves. At first the teeth were drawn on paper or sheet metal, using arcs of circles which approximated the true curves, and then templets were made to these drawings by which the cutters were shaped; but it was found that cutters made in this way would not give as good results as desired, and afterward curves which were very much nearer the theoretical lines than arcs of circles were laid out by means of Professor Robinson's Templet Odontograph, and cutters made from them with much better results. Later, the curves of coarse pitch teeth were laid out, and sheet-steel formers shaped to the lines as nearly as possible. These formers were then placed in a Pantagraph Machine, and cutters of the different pitches made from them; but it was found impossible, even with the large drawings, to make the formers sufficiently accurate. The Epicycloidal Milling Engine was then constructed for generating and milling the curves of one diametral pitch upon a steel plate, and, with these formers as a basis, cutters of any pitch were made. Thus step by step the process of making gearing with special cutters for each wheel, according to the Willis theory of the interchange system, has been brought to great mechanical perfection.

In the new process, which is the subject of this paper, instead of making all gears so that they will run into a rack, the rack is transformed into a cutting tool, and by it the teeth of wheels of any diameter are generated and cut at the same time.

Fig. 76 illustrates a gear generating and cutting engine, designed and constructed by the writer, for the purpose of reducing to practice the principles of this process. The cutters are shown in position as they appear in the machine when the teeth are cut partly across the face of the wheel.

The cutting spindle and the main spindle which carries the wheel are connected by means of change gears, the number of teeth to be cut in the wheel determining their proportion, on a

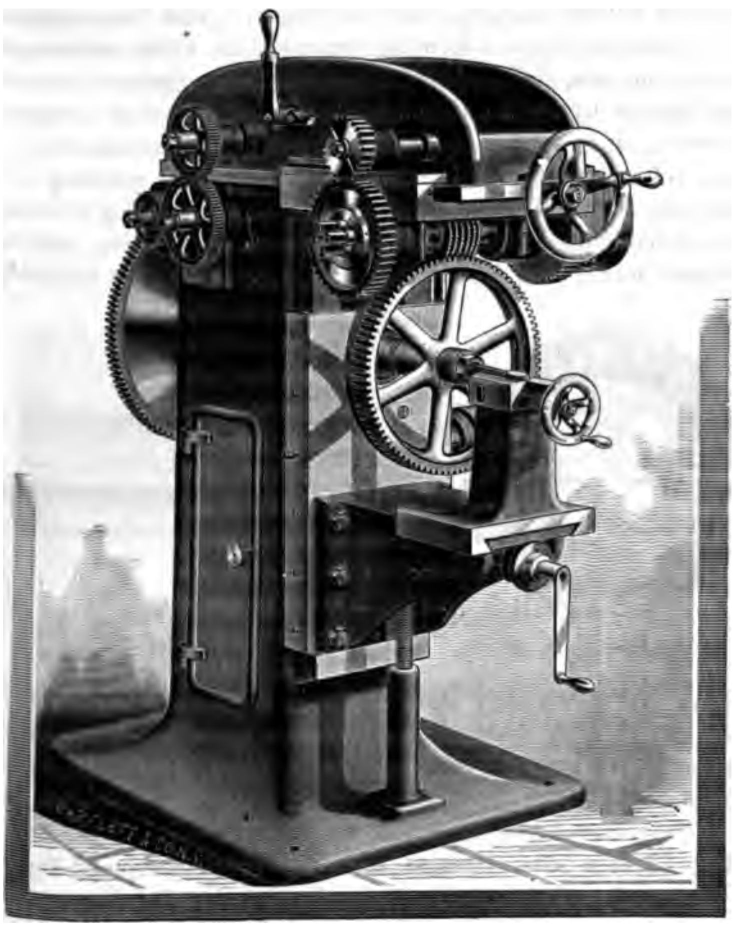


FIG. 76.

similar principle as the change gears of an engine lathe, thereby causing the cutting spindle to make as many revolutions as there are teeth required in the wheel, while the main spindle makes one revolution.

The cutting tool, which is composed of a series of cutters rigidly connected, revolves, and at the same time moves longitudinally, or endwise, at right angles to the axis of the wheel to be cut; and at the same speed, it is continually revolving at the pitch line, the motions being the same as in the case of a rack engaging with a revolving gear.

As it would be impracticable to continue moving the whole series of cutters endwise, they are bisected, and these segments are connected in series forming two sections, which revolve upon a common axis, and each section is given an independent endwise motion by means of a cam. When one section is engaged in cutting, it is carried endwise in the same direction and at the same velocity that the pitch line of the wheel is revolving, until disengaged from it, when the cutters, while continuing to revolve, are carried back by the cam to their original position, ready for the next tooth. By means of both sections, as they continually

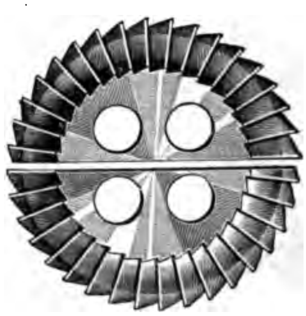


FIG. 77.

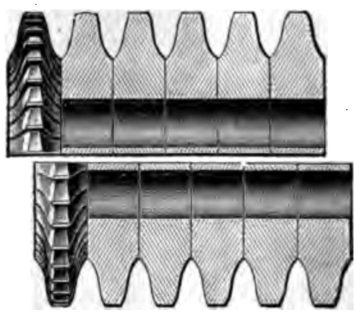


FIG. 78.

revolve and alternately slide forward while cutting, and back when disengaged, there is a continuous cutting and generating process of the teeth in the revolving wheel. The head carrying the cutters is automatically fed across the face of the wheel, and when the cutters have proceeded once across the gear is completed.

Fig. 77 is a side elevation of a bisected cutter, and Fig. 78 shows a series of six cutters, the end one being in elevation and the others in cross-section—these having cutting portions, which in cross section represent the teeth of a rack, with the addition to the diameter of a given proportion of the pitch by which the clearance and fillets at the bottom of the teeth are made. If their cutting portions are formed of cycloids, then the whole set of gear wheels cut with them will be of the epicycloidal or double-curve system. If they are formed simply of straight sides, then a set of involute or single-curve gears will be generated and cut, or their cutting portions may be composed of both straight lines and cycloids and produce Professor McCord's recent system of gearing, which has composite teeth with the contours partly involute and partly epicycloidal.

All the cutters in a series are made exactly alike and interchangeable, the thickness of each or the distance from the centre of one to the centre of that adjoining being equal to the pitch of the gear to be cut. As indicated in Fig. 77, the two segments of a cutter are first made whole, with four holes an equal distance from the centre, through which the rods pass that fasten them

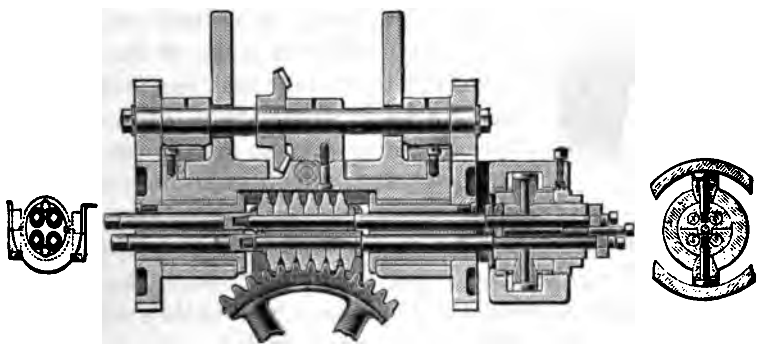


FIG. 79.

together. After the cutters are nearly completed, they are bisected with a narrow tool, leaving two holes in each segment.

Fig. 79 is a cross-section of the head, showing the mechanism for revolving and reciprocating the cutters. The rods which extend through the cutters serve not only to hold them firmly together but to revolve them, and at the same time act as slides for the reciprocating motion. The spindles on either side of the cutters, through which the rods extend, are revolved independently and at the same speed by means of a parallel shaft, having a pinion at each end, which engages with a large gear on each spindle. By this means the four rods carrying the two cutter sections are revolved from each end, thus avoiding the torsional strain which would result if driven from one end only. The pair of rods for each section, after passing through one of the spindles, terminates in semi-cylindrical blocks. From each of these blocks a stud extends, on which is journaled a roll, engaging with a cam attached rigidly to the head. This cam is shown in Fig. 80, the working portions being made in the form of a screw thread, which, if extended all the way around, would have a lead equal to th thickness or pitch of the cutter. As each section of the cutter engages with the wheel but three-fourths of a revolution, th thread portion of the cam which carries the cutters forwar extends only three-fourths of its circumference, leaving th other one-fourth for the reverse curves of the cam to bring th cutters back to their starting point. Pro vision is made for adjusting one section o cutters so as exactly to coincide with th other. The variation in the spacing fron one tooth to another is reduced to a mini mum, as the series of cutters act upon botl sides of a number of teeth at the same time and serve to average and eliminate any loca inaccuracies in the division of the index an driving gears; also to obviate any tendenc to crowd the wheel from one side to th other.

FIG. 80.

The endwise motion of the cutters and the velocity of th wheel at the pitch line being exactly the same, the process c generating and cutting the teeth goes on continuously and un formly around its entire periphery, so that one part is not heate more than another, but all the teeth are cut under exactly th same conditions, and when the revolving cutters have onc passed across the face all the teeth in the gear are complete and given the correct form for each diameter of wheel; and as b the Willis theory all gears are cut to run into a rack, so by th process the Sang theory is put into practice and a rack is mad to cut correctly all gears.

DISCUSSION.

Mr. A. H. Campbell.—I would like to inquire of Mr. Swasey gear-cutting on this self-forming principle has ever been su cessfully accomplished, to his knowledge by means of a rac shaped cutting tool moving to and fro, as in a shaping machine Of course, we recognize the novelty of the system he has s beautifully perfected, but I have had in my mind for some tim that the same thing has been accomplished in a more or les perfect degree by the moving rack. I would like just a littl information on that point, as the subject is one in which I a very much interested.

Mr. Oberlin Smith.—I hope there will be more questions asked and facts given in regard to this paper, for it certainly records a revolution in gear-cutting, and a very novel and interesting process. I would like to ask Mr. Swasey regarding the practical efficiency of this system compared with the old methods, especially as respects speed. Of course, we all recognize the economy of having so few cutters—that is, of having one set of cutters answer for all possible diameters of wheels. But, in actual practice in running the machines, about what economy is there over the other method in turning out commercial gears? It seems from what Mr. Swasey says that this machine must be much more rapid than those working by the old processes on account of the greater number of teeth doing the work.

In regard to the question asked by Mr. Campbell, I think that the idea of which he speaks is analogous to the action of the Bilgram machine for planing bevelled gears. This machine can be adjusted so that it works upon a cylinder instead of a cone. I think if it were adjusted to that position that it would be just the process in question—generating the tooth curves as it goes along with a shaper tool made to the same contour as an involute bevel rack-tooth would be.

Mr. Jno. T. Hawkins.—I do not know that I would like to undertake to express my admiration of this new method of gear-cutting fully, or that I could do so. I think it is one of the most admirable devices which I have ever seen in connection with gear-cutting, and I rise now merely to say, if it effected nothing further than that advantage which it has in preserving the diameter of the wheel constant while being cut, as affected by expansion from heating, or obviating the wear of cutters in going around a large wheel, that this consideration alone would justify the adoption of this method. In my experience in gear-cutting, there is nothing more difficult to overcome than the disparity obtained in the first and last teeth of wheels from expansion due to heating of the wheel, especially where the material is of a somewhat refractory nature, such as phosphor-bronze, which is a quite difficult material to operate upon with revolving cutters such as are used in cutting gears, particularly in cutting epicycloidal teeth. This principle of continuing the partial cutting around the wheel and gradually working across the face for every rotation of the wheel maintains whatever change in diameter is effected constant for all parts of the wheel. This is a very important matter, and completely overcomes that difficulty of having a large or a small tooth left at the end in getting around a large wheel, which we so often encounter.

I would like to ask Mr. Swasey a question which has just occurred to me. In cutting a gear like this, if it rotates pretty slowly and the cross-feed is considerable, it would produce a very similar effect on the tooth to the ordinary tool-cut mark in a planer or lathe. I would like to ask him about what proportion the feed bears to the rate of rotation; about how long it takes to cut such a gear as that largest one shown.

Prof. Jas. E. Denton.—I am afflicted with the same feeling of which Mr. Hawkins has spoken in regard to this paper, but I do not wish it to pass without saying a word of praise regarding it. Certainly, it is a piece of mechanism for American engineers to be proud of. It is very natural to expect that it should arise from the source from which it has, since we know of Mr. Swasey's work in former years at Pratt & Whitney's, where he was the first, I believe, to generate teeth entirely by mechanical processes; and now he has gone further in giving us this beautiful principle. I want to say that while I admire the mechanical accomplishment in the highest degree, I think that the modesty of Mr. Swasey in waiting three years until after the apparatus was a practical success before making it public is a still greater and rarer thing.

Mr. S. Tompkins.—I had the pleasure of going through Mr. Swasey's shop two or three years ago, and he showed me his machine, and I became very much interested in it, and have thought about it a great deal since. It has occurred to me that, by arranging the teeth spirally around the cutter, like an ordinary hob, and setting the cutter at an angle, it might be possible to obtain the same results without the cost and difficulty of making the cam and making the cutters in halves. I never had an opportunity to make a complete machine on this principle, but I made such a cutter and put it on an ordinary milling machine and cut some wheels of widely differing diameters which seemed to work very well. I have never had occasion to use this plan for any practical purpose. I would like to ask Mr. Swasey if he has ever tried it, and if so, with what results?

Mr. Jas. McBride.—I would like to ask Mr. Swasey if he finds any more work on these cutters in the middle than on the ends. It appears to me there is more work done on the middle cutters.

Mr. Ambrose Swasey.—In answer to Mr. Campbell, I will say that I studied for a long time to find some practical way of making the rack into a cutting tool which would cut the teeth of wheels to the best advantage, and at the same time correct in theory. I found several plans which would approximate the theoretical shape, and others which were correct, but did not seem to me practicable. The idea of using the rack as a planing tool was among the latter, and also that of using a series of rotating milling cutters representing a rack, which I found was also open to some objections; but when I struck upon the idea of bisecting the milling cutters, that solved the problem, and is practically the fundamental principle of this machine, there being nothing specially new in the idea of having a rack to do the work, but simply the way in which the rack is constructed so as to form a cutting tool, and its application to the wheel to be cut.

As to the president's question about the efficiency of the machine, at first I thought of making a model, and afterward concluded, since I was in a position to make a practical working machine, that would be by far the best. I decided to make a machine large enough to cut a gear 20″ in diameter, 4 diametral pitch. This machine is now practically as first constructed, and has been used a great deal in our factory for cutting our regular gearing, with very satisfactory results; but it was made to demonstrate the practical solution of a principle, and therefore I have not studied its efficiency or commercial value.

Answering Mr. Hawkins, the velocity of the gear depends upon the number of teeth. Assuming the cutters make 40 revolutions per minute, if a gear has 80 teeth, it will revolve once in two minutes, and proportionately faster or slower as the gear has a less or greater number of teeth.

The feeding mechanism of the machine is governed by the spindle which carries the gear, so that for every revolution of the gear the cutters are fed a given distance across the face whatever the diameter may be, leaving cutting marks on the teeth similar to those produced by the regular milling process.

As to the time of cutting gears, I have compared it with our regular gear-cutter, and found that a gear could be cut in very much less time, but I cannot give you any definite data on that point.

Mr. McBride asks whether wear is equal on the different parts of the cutter.

Most of the work comes on the middle cutters, the outside ones having very little to do, and then only in case the gear to be cut has a very large number of teeth. The smaller the number of teeth in the gear the less number of cutters are brought into action. As the cutters are made entirely interchangeable, the outside ones can be put in the middle or changed about in any way, so as to make the whole set wear out uniformly.

The system of cutting spur wheels to which Mr. Tompkins refers has been tried several times. I understand such a machine was made in Germany more than twenty-five years ago, and several attempts have been made in this country to cut gears on the same plan. There are, however, some difficulties which are very hard to overcome on account of the hob being spiral. In order that the teeth of the wheel may have correct contours, it is necessary to have the hob large in diameter, with the angle of the thread very great, which has made the system not altogether practicable.

Prof. Denton.—Does this new machine cut any slower than the ordinary process?

Mr. Swasey.—No, sir; for the reason that when cutting a gear with 80 teeth there are perhaps 5 or 6 cutters each doing its part of the work at the same time instead of one, and while the cutters on the outside of the set are commencing to form the teeth, those in the middle are finishing.

There is no time lost in backward motions, but a continuous milling process from the time the gear is commenced until it is completed, and it is only a question of how fast a set of milling cutters can be fed across the face of the gear. That I will leave to your experience and judgment. I will say, however, that I find in practice that it cuts the gears very quickly, and the machine is so thoroughly automatic that a man can go about other work.

CCCCXXIX.*

A SINGLE-ACTING COMPOUND ENGINE.

BY WILLIAM A. BOLE, PITTSBURG, PA.
(Member of the Society.)

THE single-acting compound engine of which it is desired to speak in this paper, has been built up upon the plan of the single-acting high-speed engine which has been before the American public for some years.

The advantages offered by the self-contained construction have been elsewhere reviewed, so that it is scarcely necessary to repeat them. The working parts are impervious to dirt and grit, and copious lubrication is secured, and accurate maintenance of foundation and alignment is rendered less necessary. By a compound single-acting high-speed engine, an increased amount of power at a reduced cost can be developed in close quarters where ground room is costly; a simplicity of design is secured, so that any ordinary engineer can operate it safely and successfully; and by having a single valve controlled automatically for both cylinders, under all ranges of pressure and load, simplicity is secured to a very great degree.

The cylinders of the compound engine are provided with trunk pistons, each of which is coupled by a connecting-rod to its own crank-pin, these latter being opposite each other, or 180° apart, so that as one piston descends the other ascends, and at the same rate of speed. Above and across the top of these two cylinders is the steam-chest containing the valve by which steam is admitted to and discharged from the high-pressure cylinder into the low-pressure cylinder, and thence into the condenser or the atmosphere. This valve is actuated through a bell crank rocker arm by a shifting eccentric on the shaft outside of the engine frame.

The sectional views, Figs. 81 and 82, show clearly the construction of the engine.

A by-pass valve allows steam to be admitted from the boiler to

* Presented at the Richmond meeting of the American Society Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

the clearance space, and thence into the low-pressure cylinder, to facilitate starting up the engine. This valve, of course, is kept closed while the engine is running. Relief valves allow any

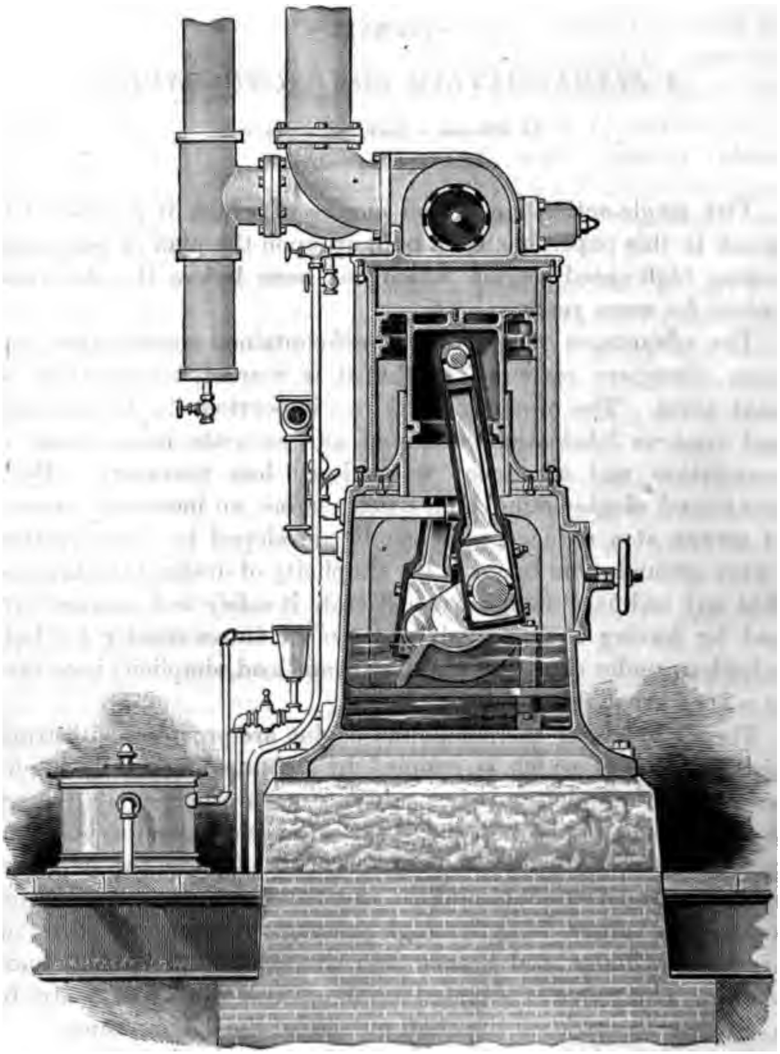


FIG. 81.
Section through Low-pressure Cylinder.

entrapped water to escape from the cylinders by this means, and lessen the danger from breakage by water. It will be seen that there is no receiver space between the high-pressure cylinder and the low-pressure cylinder. The space around the central body of the valve is always in connection with the high-pressure cylinder, and forms part of the high-pressure clearance.

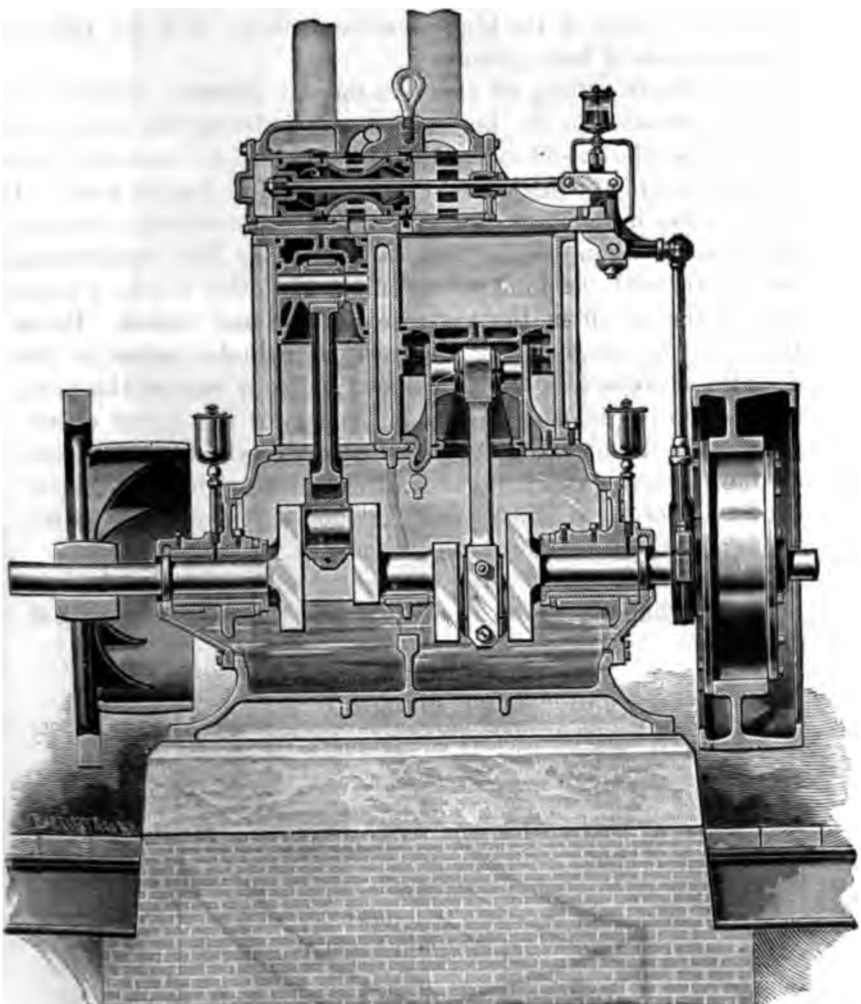


FIG. 82.
Longitudinal Section.

The use of a shifting eccentric and a single valve involves, of course, a variable point of compression in both cylinders. It is of great importance to bring final compression in the high-pressure cylinder up to the initial pressure, and neither above nor below that point, no matter what the engine's load may be or where the point at which cut-off occurs. This end is accomplished in the single-acting compound engine by carefully proportioning the clearance volume of the high-pressure cylinder and the relative displacements of both pistons.

The valve in cutting off steam in the low-pressure cylinder begins compression in the high-pressure cylinder by the same act; and thus, as the cut-off in the low-pressure cylinder becomes later, compression in the high-pressure cylinder also begins later. If the pressure in the high-pressure cylinder were always constant when compression began, then, of course, the final compression would vary with the point of beginning. In other words, it would vary as the cut-off in the low-pressure cylinder varied. But as this *point of cut-off* in the low-pressure cylinder varies, so also does the *pressure* in the low-pressure cylinder vary at this point, and that in the inverse direction. The later this point of compression is, the higher is the pressure in the cylinder at the time, so that the result is a uniform final compression in the high-pressure cylinder up to boiler pressure; and it is to this feature that much of the economy of the engine is due.

This variable cut-off with constant compression in the high-pressure cylinder is shown graphically by the following series of diagrams:

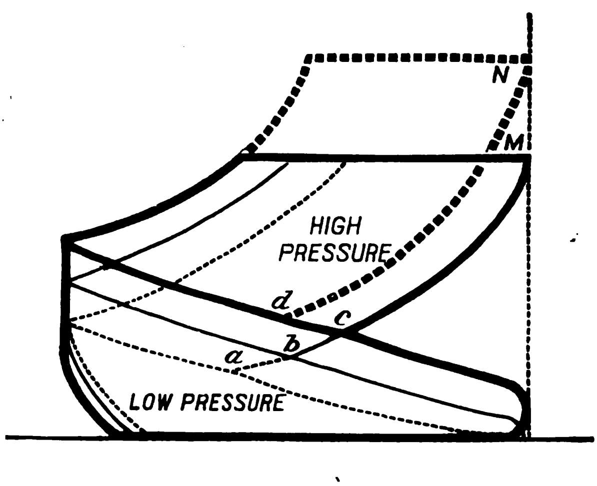


FIG. 88.

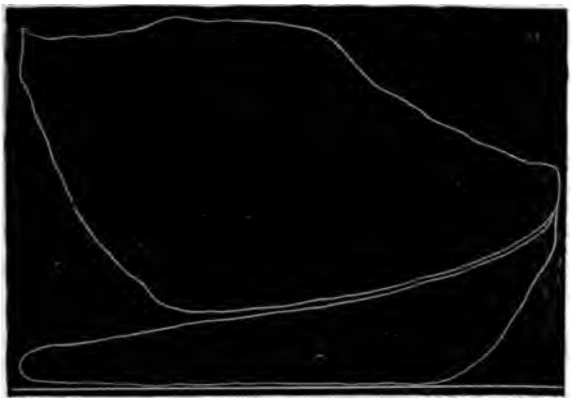


FIG. 84.

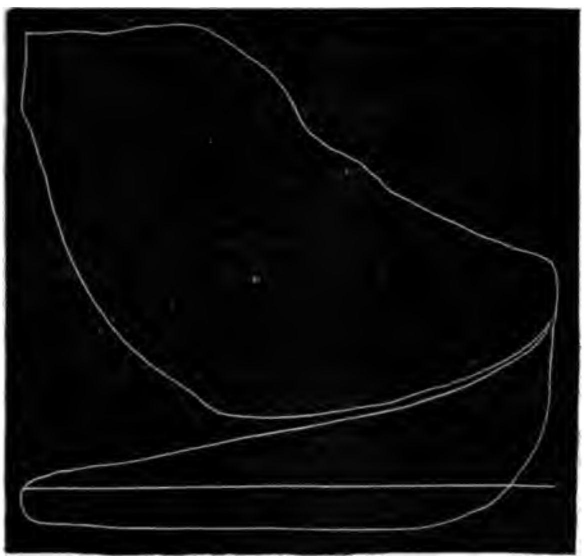


FIG. 85.
Sample Cards taken in Test Room.

Of course the compression in the *low-pressure cylinder* varies with the point of cut-off as in any engine having this type of valve gear. In this cylinder, clearance is kept at the lowest possible point.

The diagrams shown, Figs. 84 and 85, illustrate the efficiency of this construction. These were taken from a 14-inch and 24 × 14 inch engine. It will be noticed that the fall of pressure between the high-pressure cylinder and the low-pressure cylinder is very small, as would be expected in view of the directness of the passage from one cylinder to the other and the absence of radiating surfaces between the two.

The construction of the low-pressure piston is peculiar and worthy of note. The lower end of this cylinder is closed by a sleeve, or internal cylinder, which is bored to the same diameter as the high-pressure cylinder. The low-pressure piston has therefore two diameters; its upper end being turned to fill the low-pressure cylinder, and its lower end to fill this sleeve, or internal cylinder. This construction is primarily intended to equalize the displacement of air in the crank chamber of the engine and to prevent the pumping in and out of air as the low-pressure piston ascends and descends. This efflux of air would carry with it particles of oil and water held in suspension, and would be objectionable. This arrangement, however, has another use, and a quite important one. At the bottom of this interior chamber is an outlet provided with a check valve, opening outward, and closed by a light spring so arranged as to lift at slightly more than atmospheric pressure. As the low-pressure piston descends, it expels from the interior any pressure of air or steam, and at the bottom of its stroke there will be scarcely more than atmospheric pressure in the chamber when the check valve exhausts into the air. As the low-pressure piston ascends there is a partial vacuum produced under it, which helps to hold down the reciprocating parts of the low-pressure side, and maintain the single-acting effect which distinguishes this engine from others. This action assists compression, in bringing the reciprocating parts to a stop at the upward limit of the stroke.

Two indicator diagrams which were taken from this chamber are shown in Figs. 86 and 87.

The governor is enclosed in an oil-filled case keyed to the engine shaft, and is a modified form of the old and well-tried centrifugal shaft governor with shifting eccentric. Motion is taken to the valve by means of a single rocker arm. The eccentric strap has a spherical joint to provide for the movement in two planes, as has also the upper end of the eccentric rod where it is attached to the rocker arm. The case containing the governor is completely filled with a heavy oil, and thus the latter is lubricated for long continued runs. The presence of this oil in the case effects a change in the speed of nearly 10%, the speed increasing when the oil is added. This, of course, is due to the fact that the governor weights lose part of their centrifugal effect when immersed in the oil, which latter is also revolving with the case and at the same speed. On account of the copious lubrication of the governor, no matter what the running periods are, the regulation is uniform and reliable, and is not, as in many cases, dependent on the period of time which must elapse between stops. All

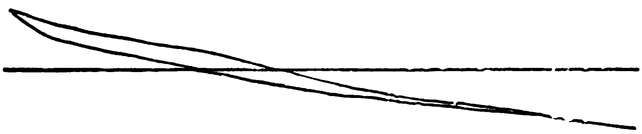


FIG. 86.

Diagram from Annular Space, Non-Condensing. 20 lb. Spring.

other forms of shaft governors, so far as is known, require to be oiled while the engine is standing idle.

The method of testing these engines is interesting, and the results are of value. After each engine is put together in the erecting shop, it is lifted bodily, and carried into the testing room, where it is bolted to a foundation, and connected up to steam and exhaust pipes, and regularly run for a period of from 20 to 30 hours. This test reveals any latent defects in the cast-

FIG. 87.

Diagram from Annular Space, Condensing. 20 lb. Spring.

ings, which might otherwise escape detection. It assures the adjustment of all parts, the strength of all the engine's members, and prepares it for immediate service.

The valve is adjusted to obtain proper indicator diagrams. A band wheel, having internal flanges on its rim, is keyed on the engine shaft, and a Prony brake encircles the same. By running a continuous stream of water into the rim of this wheel, it is an easy matter to carry a brake load of 250 H. P., when necessary, for hours at a time. The surplus water is taken off by an overflow scoop. The brake band is lubricated with raw fat of pork. The elongated arm of the brake band rests on a pair of accurate scales, and the load is kept constant during a test, by the tension screw in the band, which is adjusted by the attendant from time to time, so as to keep the scale beam at all times in poise. The exhaust pipe of the engine is connected to a surface condenser of improved construction; the injection water is supplied from the city mains or from a large tank containing several hundred barrels, and is kept circulating through the condenser by a pump. The indicator is applied to the engine and diagrams taken frequently. The brake is tightened up to secure the brake load which is desired. The indicator diagram shows the horse-power developed in the cylinders. The difference between the indicated and brake horse-power is the engine's friction under load. The steam, after escaping from the engine, is converted into water in the condenser, and is drawn off at the bottom of the same into tanks which rest on platform scales, and is accurately weighed and charged to the engine.

When it is desired to run a test with vacuum, the air pump which is shown under the condenser in Fig. 88 is started; the free outlet of the condenser is closed and the water and air are pumped out and discharged through a second system of pipes into the tanks, and there weighed as before. The steam used to operate the air pump is generally discharged into the atmosphere, and no account taken of it. In cases where the air pump is driven by the engine under test, the indicated horse-power is simply increased by the amount of power necessary for the pump; so that where the more wasteful direct-acting air pump is used, it should not properly be charged against the engine. Such test is made on every engine of this type built; and all the data, with the indicator diagrams taken during the test, are filed away in a record book kept for the purpose, and remain with the builders as a permanent record of the engine's performance. This method of testing makes no deductions for water entrained in the steam; and if the steam-pipes be not properly drained, or the condenser should leak, it all shows up to the disadvantage of the engine. The method is extremely desirable, as it affords a correct test in a brief period of time, and also admits of using steam from the same boiler for other purposes while the test is going on.

The general arrangement of engine, brake, condenser, etc., is shown in Fig. 88.

A facsimile of the testers' record blank is given herewith:

Test No. . *Copy.* **Date.**

OFFICE RECORD.

Number of engine,
Class of engine,
Size of engine, ***10 and 18*** × ***10.***
Maker's name,
Engine used for
Tested at

Tested for
Address
Jacketing, ***None.***
Steam lap, ***⅞ in.***
Exhaust lap, ***0 in.***

TESTERS' RECORD.

	1		2		3		4		5		6	
Number of engine	*88*											
Boiler pressure	*100*											
Speed	*322*											
Brake load	*480*											
Dead weight on scales	*25*											
Time of start	*11*		*11*	*20*	*11*	*40*						
	11	*10*	*11*	*30*	*11*	*50*						
Time of stop	*11*	*20*	*11*	*40*	*12*							
Duration of test		*20*		*20*		*20*						
Full barrel, "A"	*366*		*365*		*365*							
Empty barrel, "A"	*100*		*100*		*100*							
Full barrel, "B"	*361*		*365*		*365*							
Empty barrel, "B"	*96*		*100*		*100*							
Water used	*531*		*530*		*530*							
Water per hour	*1591*											
Vacuum												
Temperature of discharge												
Leakage *per hour.*	*None*											
Brake radius	*27*	*¾''*										
Spring	*40*											
Initial pressure	*98*											
Terminal pressure	*8*											
Ratio of expansion	*4*	*91*										
High pressure M. E. P.	*51*	*5*										
Low pressure M. E. P.	*17*	*7*										
Indicated horse-power	*69*	*38*										
Brake horse-power	*64*	*51*										
Loss or friction	*4*	*88*										
Percentage of loss	*7*	*03*										
Gross indicated water rate	*22*	*93*										
Gross brake water rate	*24*	*66*										

REMARKS.	SIGNATURES OF TESTERS.
Indicated by	________
Water weighed by	________

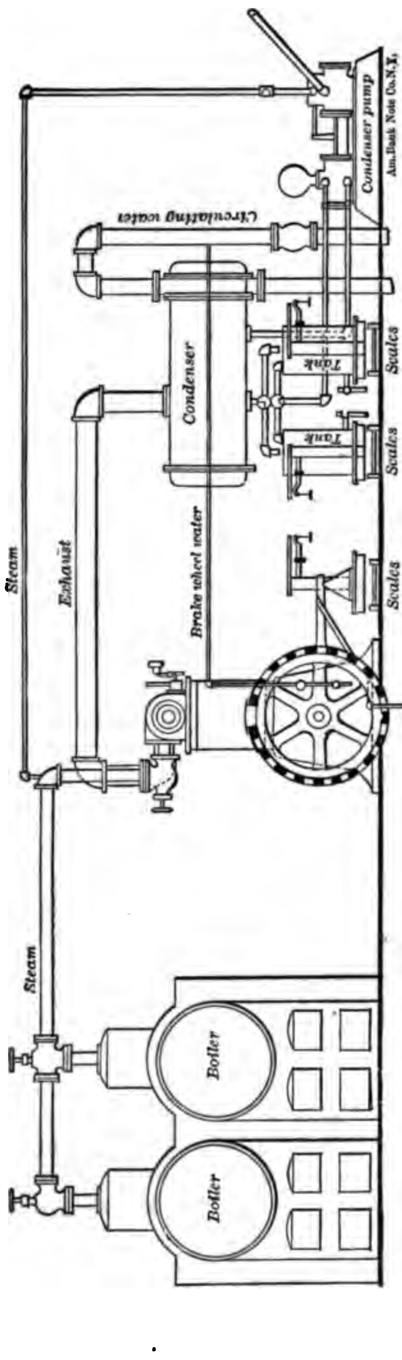

Steam
Steam
Exhaust
Brake wheel water
Condenser
Circulating water
Condenser pump
Tank
Tank
Scales
Scales
Scales
Engine
Boiler
Boiler
Am. Bank Note Co. N.Y.

FIG. 88.

An engine cannot be expected to do its best, of course, after only two or three days' running; and it will, with good attention, keep on improving in tightness, and the friction will also be reduced as the surfaces of the pistons, cylinders, shaft bearings, etc., become polished and smooth by wear.

In every case where an engine has been tested after being put into service, it has shown this expected improvement to a marked degree. As an instance: A 12 and 20 × 12 compound engine, No. 24, which was tested in the engine shop in the foregoing manner on September 6, 1888, and showed a consumption of steam of $25\frac{42}{100}$ lbs. per indicated horse-power, was afterwards tested in service, by Mr. George H. Barrus, a member of this Society, on April 6, 1889, and the consumption of steam per indicated horse-power was then $22\frac{35}{100}$ lbs. non-condensing.

CCCCXXX.*

HYDRAULIC HOISTING PLANT FOR PIER OF BROOKLYN SUGAR REFINING COMPANY.

BY LOUIS G. ENGEL, BROOKLYN, N. Y.
(Member of the Society.)

THE work to be accomplished by a hoisting plant on the pier in question was the unloading of packages varying in weight from 1,200 lbs. to about 2,600 lbs. from vessels lying alongside, including the drag along the intermediate decks of said vessels to the hatches, and the transfer of a part of these packages to the second floor of the pier for storage.

If we except the use of electricity, which for hoisting had not been extensively developed when the matter was under consideration, three general methods would naturally suggest themselves, viz.:

1. Separate steam-engines with drums.
2. Shafting driven by an engine on shore, with isolated friction drums, actuated either by friction gear or a belt.
3. A hydraulic system.

It was the extensive use of water pressure in steel-mills in this country and on docks abroad, and its great handiness in those places, which led me to suggest a hydraulic system, as well as the fact that elevators were decided upon as best adapted for hoisting to the second floor.

Notwithstanding the simplicity of separate engines, they have their disadvantages, especially when a number are required; thus:

They occupy valuable floor space on a pier, and require housing in the winter.

The condensation in long steam-pipes, the getting rid of the exhaust steam without nuisance, and the freezing of drips and connections, as well as the great excess of power usually provided, which is of no use except to damage and tear packages

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

under hatchways, and in breaking out the cargo, are all disadvantages which may be urged against this method of hoisting.

As for the second system, it will be easily seen that the use of two out of six hoists (which is not uncommon) must be attended by a great loss of power; for an engine large enough for all the work must be run, as well as the shafting belonging thereto. Such a system can only be economical where a majority of the hoists are in constant operation, as on piers used by regular lines of steamers. The system is, moreover, difficult to apply to isolated hoists, and as the plant is intended to stand when completed, would have required the oiling and keeping in line of about 500 feet of shafting, with some of it on rather uncertain supports.

The principal disadvantages to be urged against a hydraulic system are first cost, and danger of freezing in this climate. The latter we expect to prevent by using a non-freezing mixture, but I can give no figures as to whether a greater economy will balance the former. One fact is certain, that any man of ordinary intelligence can learn to hoist with a hydraulic machine in far less time than with a steam-engine, for the motion is positive, without any expansion, and under perfect control.

Of the superiority of a hydraulic over a steam-elevator there can be no question.

Again, with large mains and an excess of pump capacity, additional hoists may be added as required.

Lastly, it is quite probable that the system is an economical one, especially with a compound pump, which refinement we omitted, because the exhaust is immediately used for heating.

The plant consists of the following parts:

1. Pump for pressure, and drip pump.
2. Two weighted accumulators.
3. Pipes: High-pressure main, low-pressure return main, and drip-pipe.
4. Tanks: Suction-tank and drip-tank.
5. Four whip-hoists for unloading vessels on to the first floor of pier.
6. Two hydraulic platform elevators.

I. The Pump.

The pump is the Worthington horizontal duplex plunger pressure pump, $20\frac{1}{2}'' \times 4\frac{1}{2}'' \times 15''$, and will run at about 55 lbs., against 800 lbs. water pressure, and our ordinary exhaust pressure of 4 lbs. The pump is calculated for 800 gallons per minute. The water glands, plungers, valves, and seats are of bronze. The drip-pump is a Rumsey rotary, driven by a hand-crank.

II. The Accumulators.

The two accumulators are placed side by side in a brick tank, the bottom of which is a large block of concrete, resting on square spruce piles driven through the mud into the beach. Two of moderate size were selected rather than a single large accumulator, because one may be repaired without stopping the plant, as either can regulate the pump. In operation, one is actually weighted heavier than the other, and controls the pump steam-valve, while the light accumulator remains full in reserve for any excessive draft on the system.

The construction of the accumulators is shown by Fig. 89. The plungers are 10 inches diameter by 8 feet stroke. The weighted form was adopted in preference to a steam-balanced accumulator on account of too great a variation in steam pressure. The wrought-iron tub, *A*, with cast-iron bottom, loaded with 30,000 lbs. of pig iron, is suspended from a cross-head, *B*, on the plunger, *C*, which slides in the cylinder, *D*, with proper guide and stuffing-box. The cylinder rests on the brick foundation on the brackets, *E*, and is held down by anchor bolts. A steadiment for the weight is provided by two guides of rails on wooden posts at *F*. The tub is prevented from rising too high by two rods with buffers, as at *G*, which pass up through the cross-head from the base brackets.

The stuffing-boxes are deep, and are arranged for seven-eighths square flax packing, which has also been used on all the large plungers and pistons of the system, and has thus far given excellent results. The glands are of bronze, and have drip-pans like *H*, with pipes to catch any leak from the stuffing-box. Permanent joints are made with lead.

III. The Pipes.

The high-pressure main consists of five, four, and three-inch nominal diameter double extra wrought-iron pipes, with oval male and female flanges, and two bolts to a joint. These are shown, as well as the cast-iron tees, in Fig. 90. All the sepa-

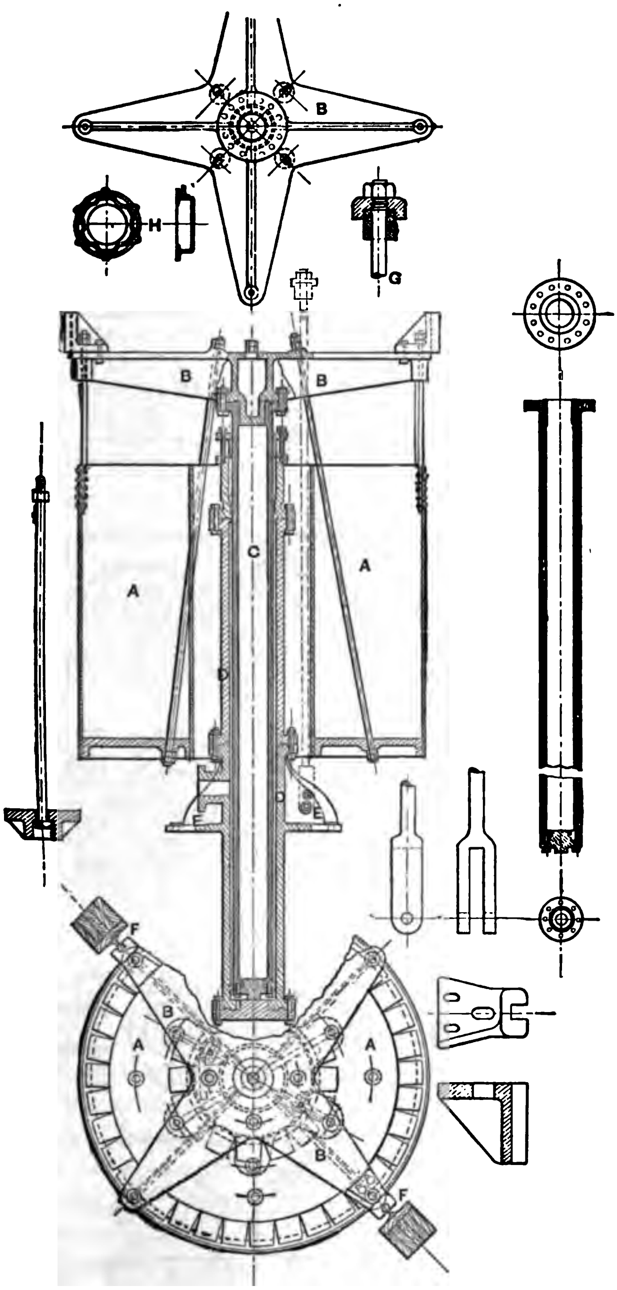


FIG. 89.

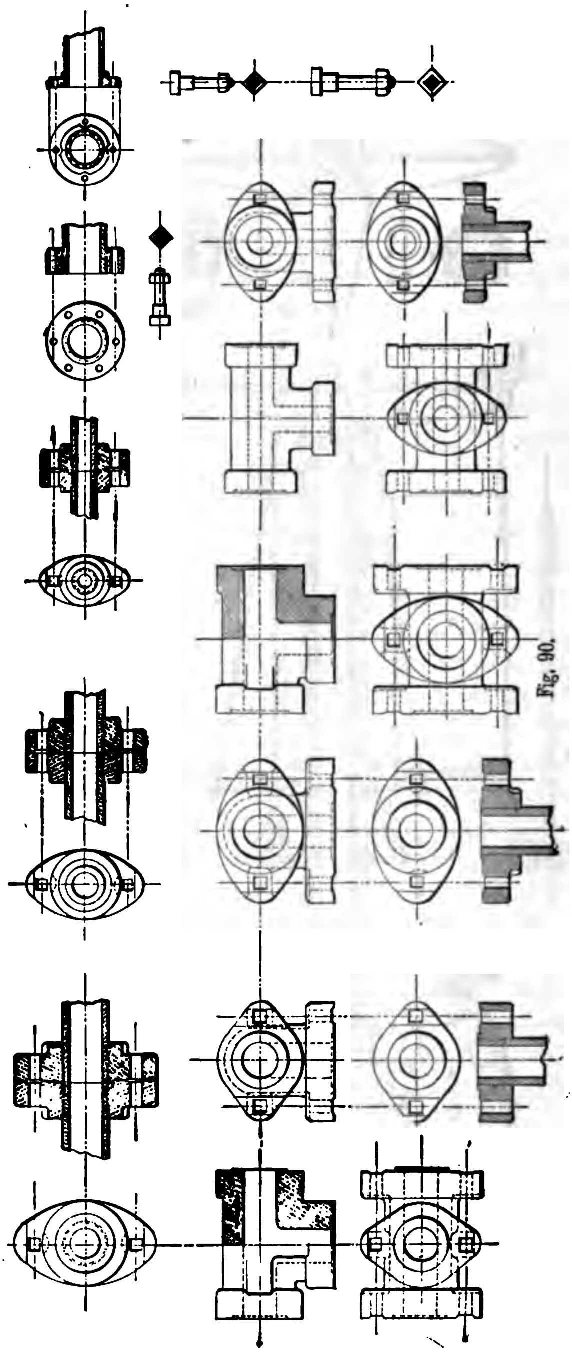


Fig. 90.

rate parts of the system were tested to 2,500 lbs. The threads are straight, and the ends of the pipes are faced fair with the flange. The joint is made between these faced ends by a gasket of "dermatine," a special rubber insoluble in oil or glycerine, which it may be deemed advisable to use in the future.

I will now describe the special appliances on the pipes to prevent a break on the main, and to nullify as far as possible the

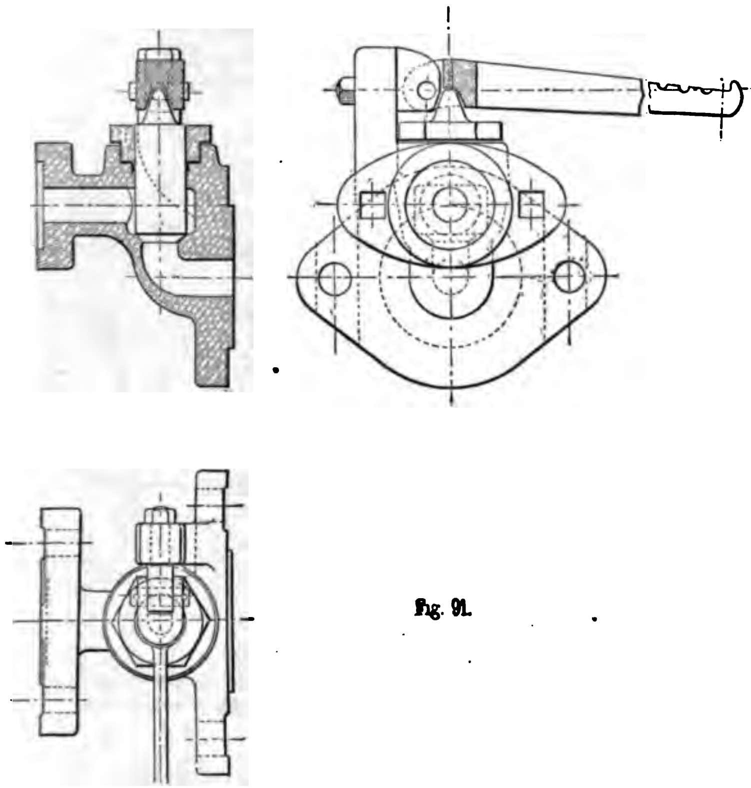


Fig. 91.

evil effects, should one occur. Thus, each connection from the two accumulators, as well as that from the pump to the main, has on it a weighted safety-valve (Fig. 91), consisting of a bronze plug working through a leather cup, and closing on a conical bronze seat, as shown. The safety-valve on the pump is to prevent the latter from bursting the connections under its stop-valve. The function of the accumulator safety-valves will be explained in connection with the automatic stop-valves, one of which follows each safety-valve on the run toward the machines.

The automatic stop-valve (Fig. 93) consists of a bronze sleeve, *A*, seated on a conical seat, *B*, in a tee body, *C*, in such a manner as to leave an annular space, *DD*, around said sleeve in the direction of the run of the tee. Thus, if the valve is closed against the branch of the tee body, *E*, the run of the main, *CC*,

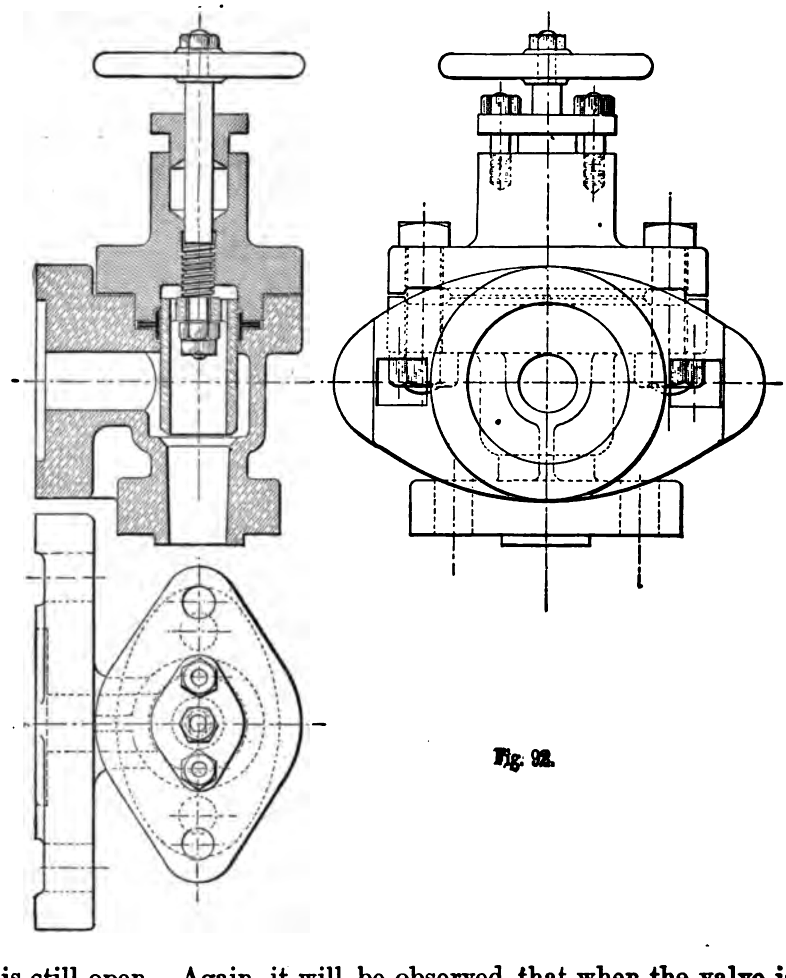


Fig. 92.

is still open. Again, it will be observed that when the valve is open and under pressure, the only unbalanced pressure is that due to the area of the stem *F*, which area is counter-weighted (not shown), so that the valve is in balance at 550 to 600 lbs. pressure. The stem is also provided with a thread, as shown, so that by screwing up the hand-wheel, *G*, against the cross-

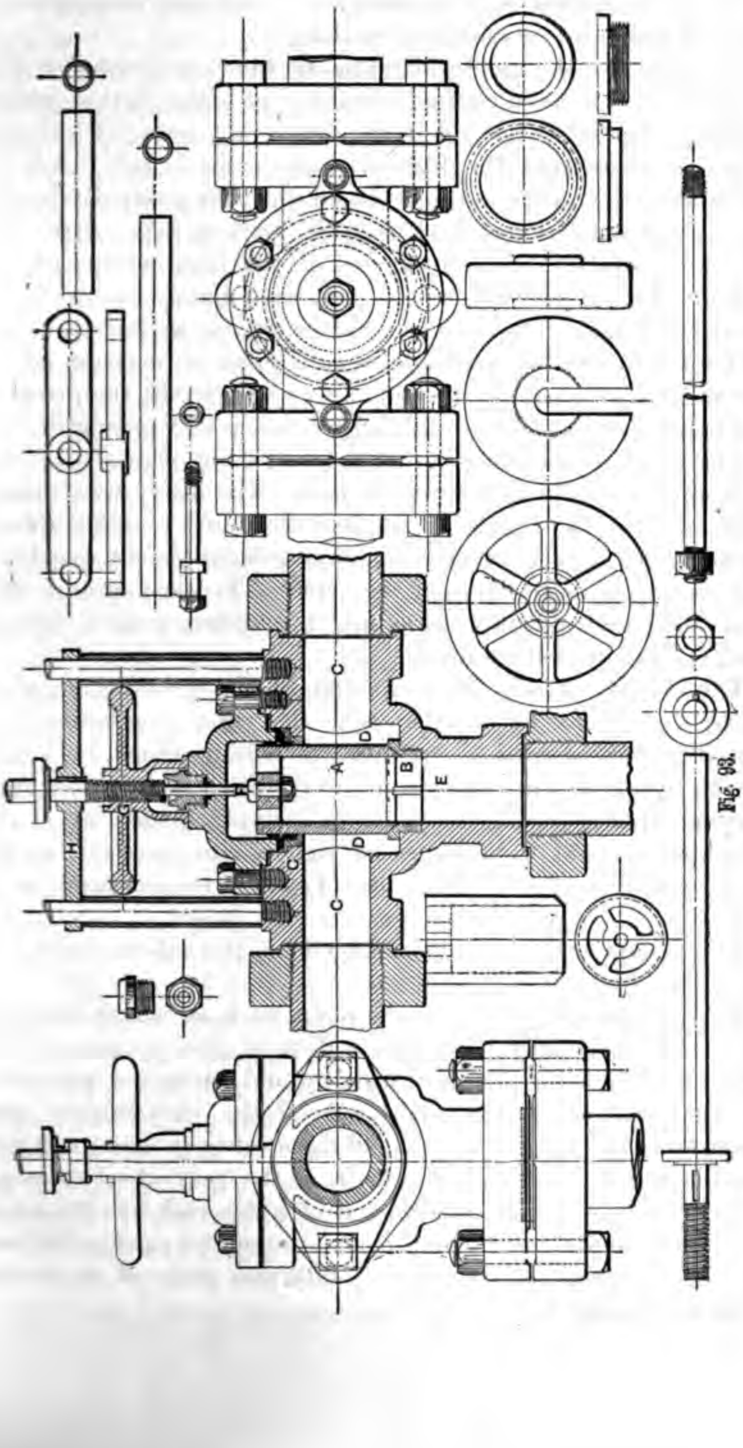


Fig. 93.

head, *H*, the valves may be used as an ordinary stop-valve to shut off either an accumulator or the pump.

In order to understand the automatic function of these valves, I will now explain the method of putting pressure on the system.

The pump is run slowly until certain air-cocks, by showing water, indicate that the high-pressure main is full. All the automatic stop-valves are now closed until the pump discharges through its safety-valve back into the suction tank, when they are again opened wide by means of the hand-wheels on the thread. These hand-wheels are then immediately run up to the cross-head again, releasing the valves, which, as they are now under 750 to 800 lbs. pressure, remain open on account of the unbalanced pressure on the stem area; but should the pressure fall below 550 lbs. they would fall by reason of overweight.

The light accumulator now rises to the top of its stroke, and is stopped there by rods from the base. The heavy accumulator then rises to two-thirds stroke, and there lifts a weight which, by means of a connecting chain, counterbalances the weight on the steam regulator valve of the pump. The last weight then falls and shuts off the steam, and the system is ready for use with the pump under control.

The weight raised by the accumulator has a second chain, slack enough so as not to interfere with the pump regulation, to a heavily weighted vertical cog-rack, in a sliding frame, held from falling by a gear-wheel meshing into it, which is prevented from turning by a projecting tooth on an inclined hinged lever, also weighted to hold it in place. A light chain passes from the lower end of this lever to the stem of each of the automatic stop-valves.

This completes the safety arrangement, the action of which is as follows:

Should a break occur on the main, such as would endanger the accumulators by too rapid a descent, the pressure would probably fall below 550 lbs., which would cause the automatic valves to drop instantly and shut off the accumulators and pump from the break. Any one of these valves in falling will jerk out the pin from between the teeth of the gear-wheel supporting the weighted rack, the descent of which rack lifts the counterbalance of the weight on the steam regulator; and as the last weight is free to act, it instantly falls and shuts off the steam from the pump.

Any further movement of the accumulators or pump which might occur will be relieved by the safety-valves under the automatic stop-valves, so that a sudden shock would probably be avoided.

This arrangement has been repeatedly tested by allowing the pressure to fall below 550 lbs., when the entire preceding action has been observed to take place within a few seconds.

The branches from the accumulators unite in a tee, from which the main runs out on the pier, passing through the automatic stop-valve of the pump on the way.

The main is twice reduced in size on the pier, and is provided with tees at suitable intervals, to which tees are directly bolted the stop-valves for the separate machines. These valves, shown on Fig. 92, are of bronze, similar internally to the automatic valves, and are provided with leather cups. The pipes from them to the machines are 2½ inches nominal diameter, double extra.

IV. The Tanks.

The water discharged from the machines is returned by a main of ordinary pipe, 2, 3, and 4 inches diameter, to the suction-tank, which is covered by an inverted conical strainer of punched sheet brass on a frame, the circumference of the base of which is an angle iron resting on the top of the tank, and of its full diameter. All the pipes from safety-valves, drip-pump, and the exhaust main discharge into this strainer, so that any particle of scale, packing, or other material which might be borne along by the water will be prevented from entering the pump. A second coarse strainer is placed over the suction pipe to hold back bolts or rivets from the first strainer which might become loose. These extra precautions may appear unnecessary, but when it is borne in mind that particles in water at such high pressure will not only cut the valves if caught under them, but will cause the pump to jump and break the high-pressure gauge, the expense is warranted.

A third pipe, 2 inches diameter, is placed under the pier, with a down grade toward the drip-tank in the accumulator pit. Into this tank flow all the leakages from stuffing-boxes, as well as the drains from all the pressure cylinders, from whence they are raised to the suction-tank by a rotary hand-pump.

We will now consider the stopping and starting device for the machines. This is the Ellington valve, is entirely of bronze, and is used both on the elevators and whip-hoists.

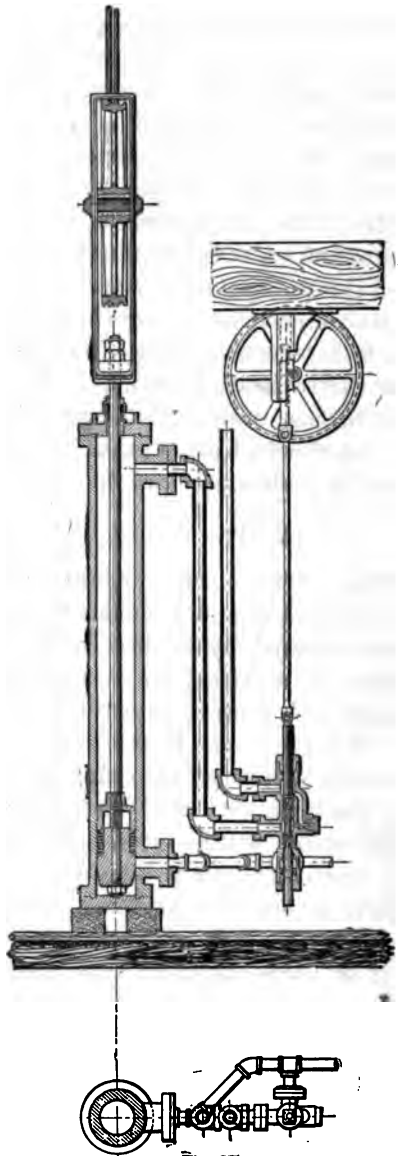


Fig. 97.

The body, Fig. 94, is a cylinder with three branches—*X*, the high-pressure inlet; *Y*, the way to the machine and the return passage from the same, and *Z*, the exhaust outlet to the return main. A cup leather is held at *M*, as shown. The piston is also fitted with a cup leather at *E*, and is turned down to form an annular space, *F*, the passage slots beyond each end of which are cut with a peculiar taper, so that when the piston is moved these slots in passing cup leathers shut off the ports and passages in a gradual manner, and thus sudden shocks on the piping, due to the arrest of moving water under high pressure, are avoided. A small poppet-valve, *H*, at *G*, held down by a spring, gives further relief in action, as follows :

If the piston is pushed to the middle of the stroke, as in the drawing, its annular passage is shut off from the pressure at *X*, and from the exhaust at *Z*, by the cup leathers at *E* and *M*, but is open to *Y*. This movement corresponds to stopping a machine. If the machine has been exhausting, the water, issuing from *Y*, raises the small valve, *H*, at *G*, in which case *H* becomes a relief valve during the time the machine is coming to rest, and balances the inertia of the moving parts against the high pressure at *X*.

It will further be easily seen that by sliding the piston to the left sufficiently, *X* and *Y*, or the pressure and the machine, are in communication by the annular space of the piston, and the exhaust is closed. By sliding to the right, *Y* and *Z*, or the machine and the exhaust, are in communication, and the pressure is shut off.

This will finish the description of the piping, except to mention that all the drips and smaller connections of the machines on the high pressure are closed by asbestos packed cocks. The exhaust pipes are provided with Ludlow gate valves.

V. The Whip-Hoists (Figs. 95 and 96).

Of these there are 4 in the system. Each consists of a massive vertical cylinder with stuffing-box and gland, resting upon a pedestal carrying 4 thirty-inch grooved wheels on a common shaft, the cylinder enclosing a plunger of cast-iron 8¾ inches in diameter, and of a sufficient length for 8 feet stroke. This plunger is surmounted by a cast-iron fork sliding on two 2-inch wrought-iron bars and carrying four wheels, the same as those below. A three-quarter-inch wire rope starts from a socket attached to the base and passes around the sheaves in the manner of a tackle, while the last whip end passes up to the second story of the pier and envelops a drum on a horizontal shaft just under the roof. This drum consists really of two

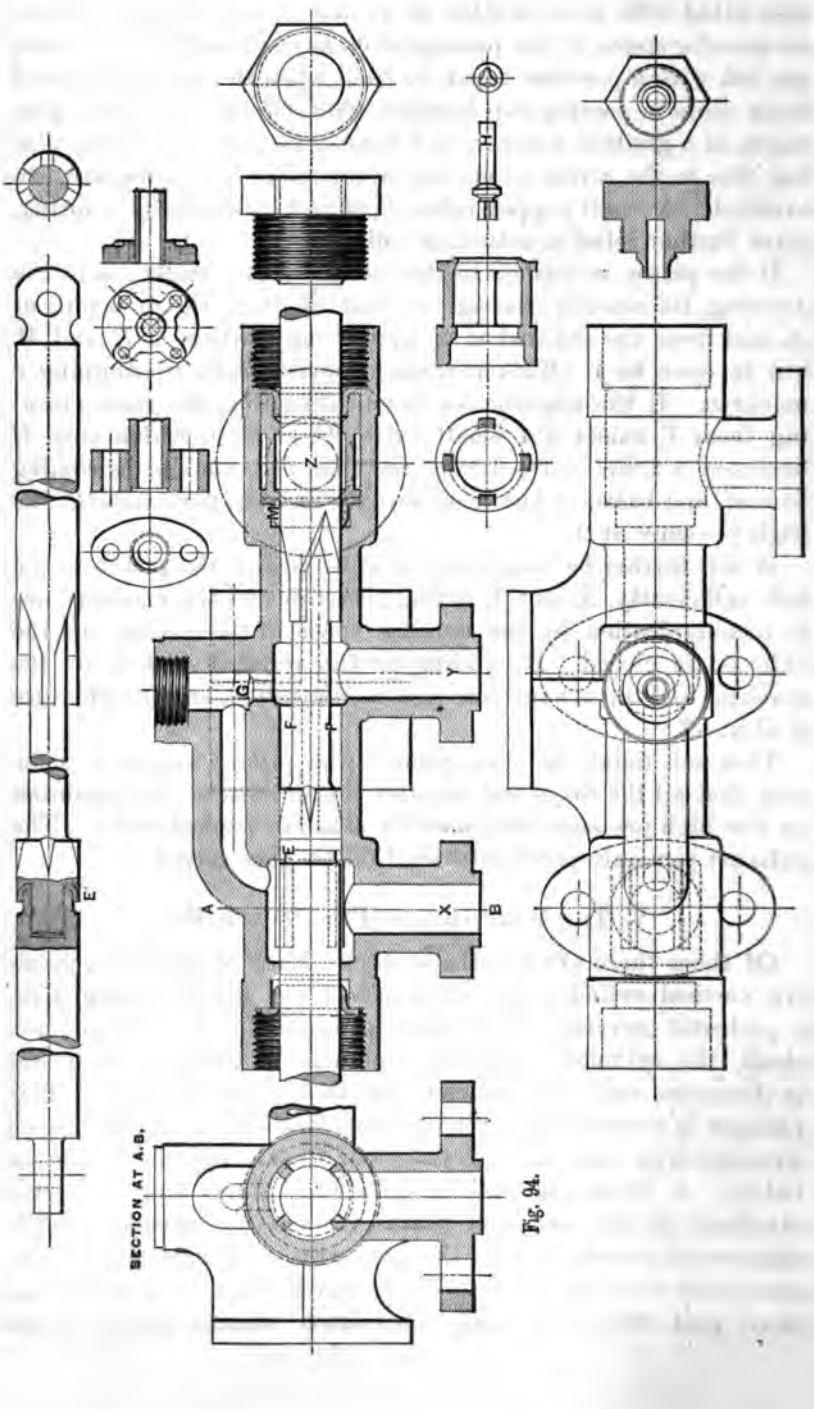


Fig. 94.

drums, one 30 inches diameter, more than twice the width necessary for the wire rope, and a drum 48 inches in diameter alongside. A counterweighted travelling sheave coils its rope into the grooves left vacant by the unwrapping of the whip end. By applying pressure under the plunger, the reverse action of a tackle ensues; the whip end is taken in by eight times the stroke, or 64 feet, and if a manila rope be coiled on the 30-inch

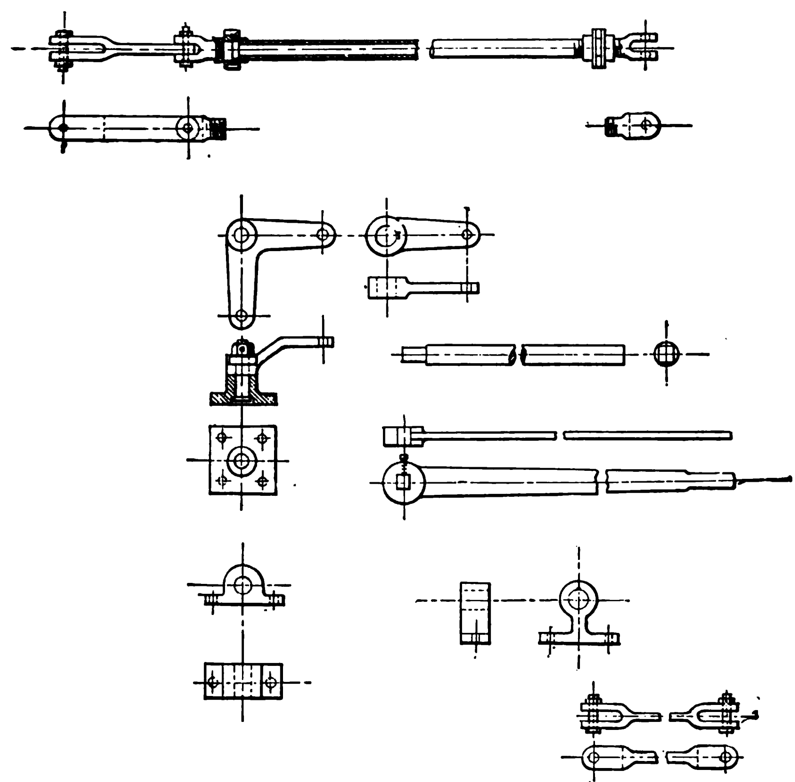


Fig. 96.

drum and run out to a vessel, 64 feet of hoist may be obtained. By exhausting the water, the plunger sinks by its own weight and is assisted by the counterweight, which, furthermore, keeps all the wire ropes taut, while the manila rope has a small counterweight attached to overhaul it.

It is ordinarily intended to hoist with the 30 inch drum one hogshead of from 2,000 to 2,500 lbs., but some allowance had to be made for the friction of the machine itself and the leading blocks. One machine, with a grooved sheave only for the manila rope, on a test with 750 lbs. water pressure, balanced 3,270 lbs.

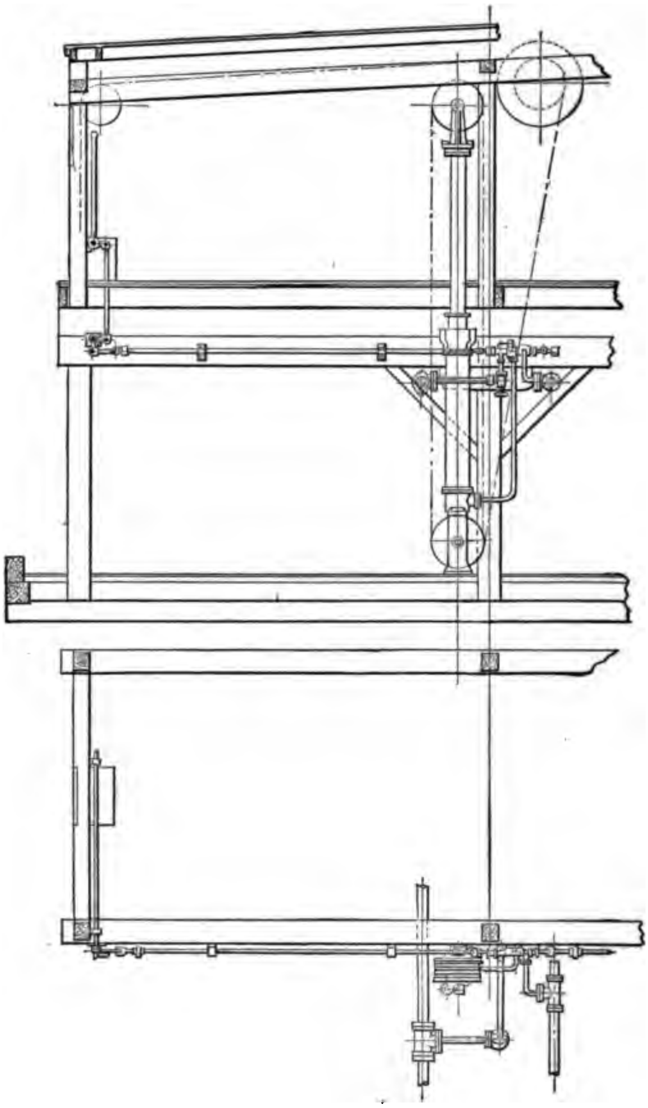


FIG. 95a.

The 48-inch drum is for use in case lighter packages are to be hoisted; for instance, slings of bags or mats, which will vary from 1,000 to 1,500 lbs. The drums are not strictly in this

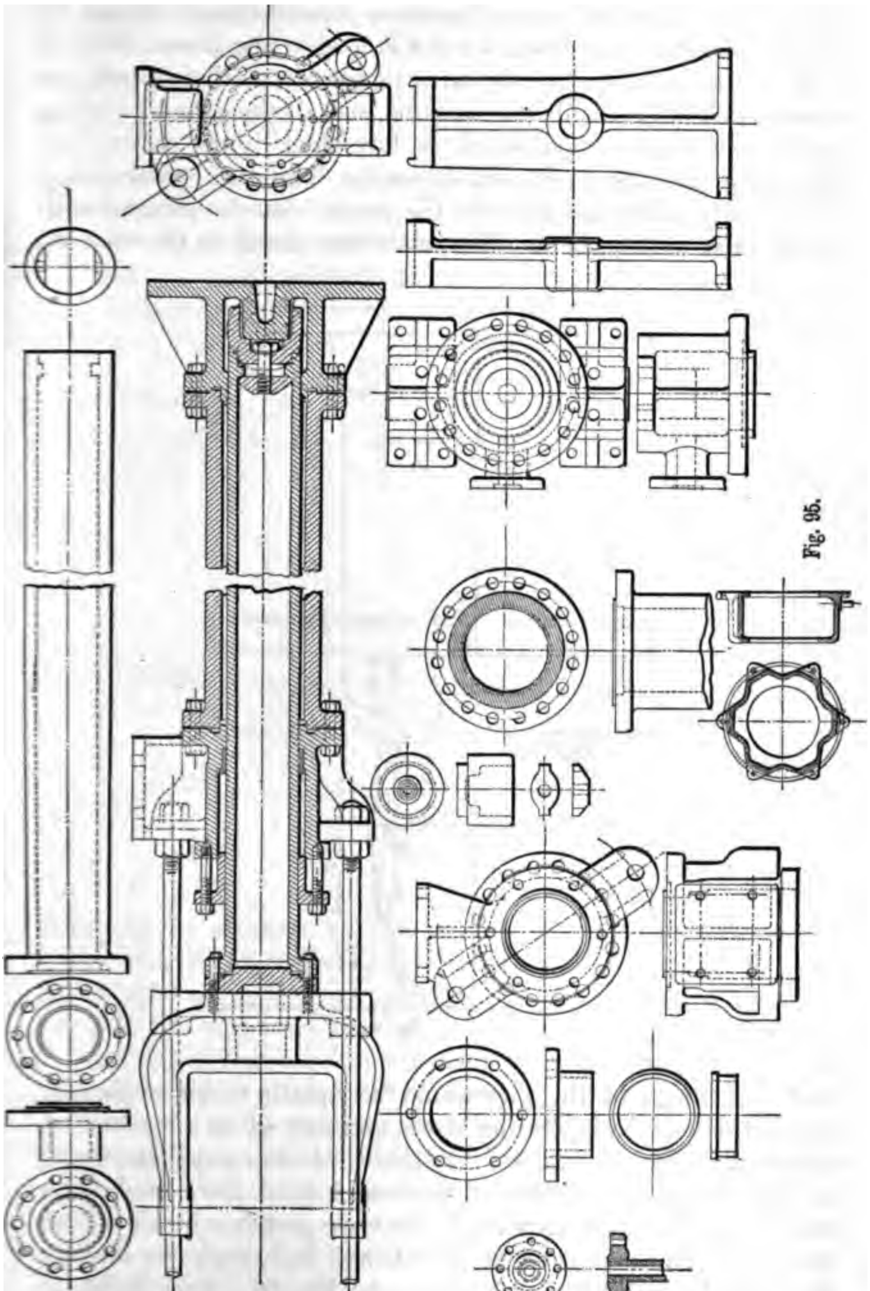


Fig. 95.

proportion; some advantage has been given the larger drum for greater speed, as the possible hoist is longer, viz., about 90 feet.

Thus the wire rope merely turns the drums on their shaft, the counterweight overhauls the machine, and all the direct hoisting is done by manila rope, which is furnished by the stevedores unloading the cargo. Fig. 95 shows the details of construction of the whip-hoist, and Fig. 95*a* the position on the pier, as well as the valve arrangement. The drum was placed in the roof so

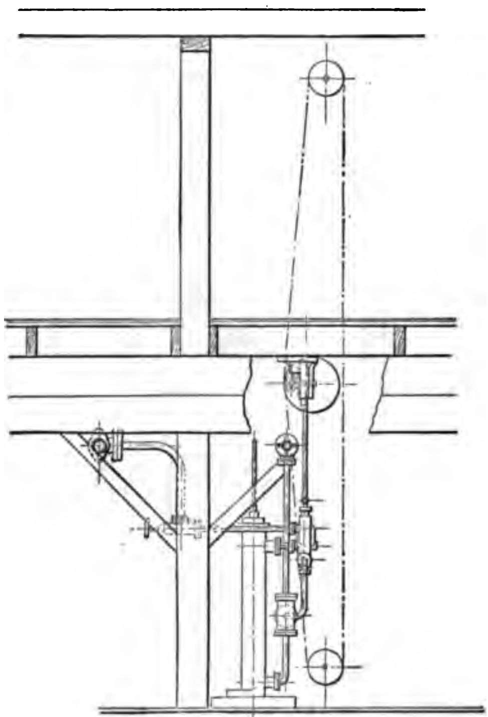


Fig. 96.

that both sides of the pier would be equally under command, and to this end the operating levers and rods go to a window on either side of the second story, so that the hoistman stands where he can view the hatchway of the vessel from above, and thus hoist more intelligently than if he were gauging by a knot on the rope. This arrangement of rods and bell-cranks for sliding the operating valve is clearly shown on Fig. 96. Each hoist is made to close its valve automatically at the upper end of the stroke, and is further restrained from leaving the cylinder by stops on the guide rods. The hoist also cushions on springs at the lower end of the stroke.

VI. The Hydraulic Elevators.

Of these there are two on the system—one on the pier, with platform 10 feet x 10 feet and 15 feet run; the other in the warehouse, with platform 8 feet 6 inches x 10 feet and 52 feet run.

Both are equipped with safety devices and are actuated by the ordinary high-pressure vertical-piston machine. This is shown on Figs. 97, 98, and 99, and differs from the whip-hoist by

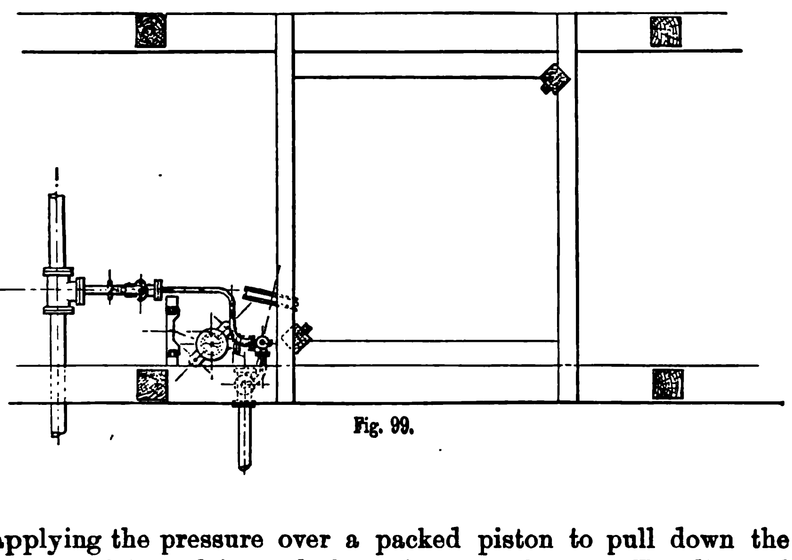


Fig. 99.

applying the pressure over a packed piston to pull down the latter with its rod, instead of pushing up a plunger. The disposal of the exhaust water is also different; thus, instead of discharging into the exhaust main direct from over the piston, it is returned by a circulating pipe under the piston, completely filling this space, to be pushed out by the next stroke of the machine. A deficiency in the quantity of water over the piston caused by the displacement of the piston rod (which is lacking below), is made up by an enlargement of the circulating pipe, which, being full, makes up what is wanting. The same valve as for the whip-hoists controls this circulation, and when this valve is in mid-stroke or closed, the piston is held between two incompressible masses of water, which lock the platform in any given place and resist any unsteadiness of the same while coming to a state of rest.

The 10-foot square platform, on a test with 750 lbs. water-pressure, balanced a load of 5,700 lbs., and will readily lift the 4,500 lbs. for which it was designed.

A direct plunger under the platform was contemplated instead

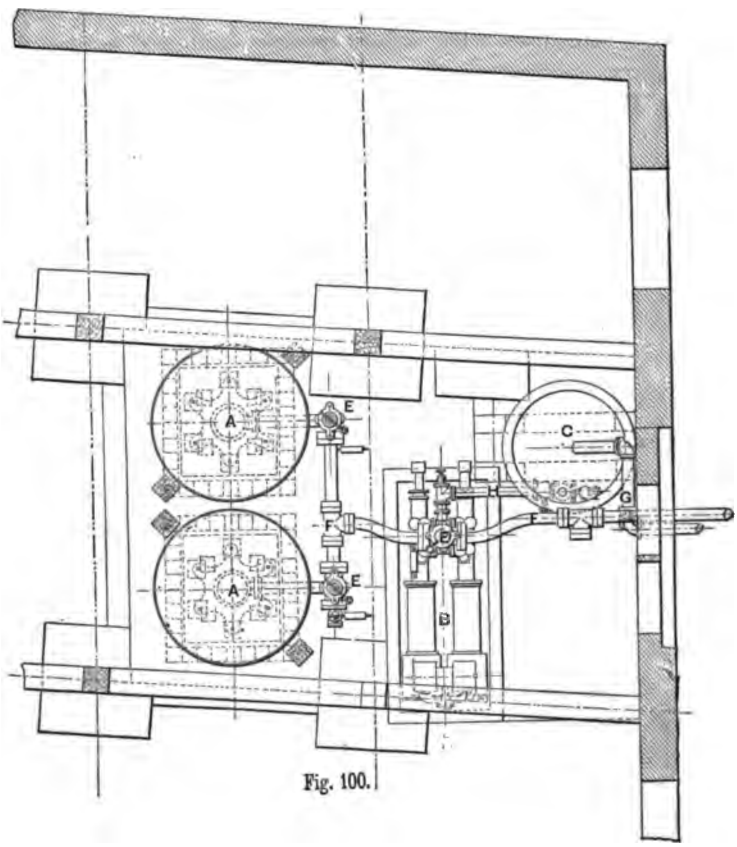


Fig. 100.

of the preceding machine, but was abandoned because it cut the dock timber too much and projected into a strong tide.

Figs. 100 and 101 show the accumulators, pump, and tanks, in which *AA* are the accumulators; *B* is the pump; *C*, the suction-tank; *D*, the safety-valves; *E*, the automatic stop-valves; *F*, the high-pressure main; *G*, the low-pressure return main; and *H*, the suction to pump. The drip-tank and pump are not shown.

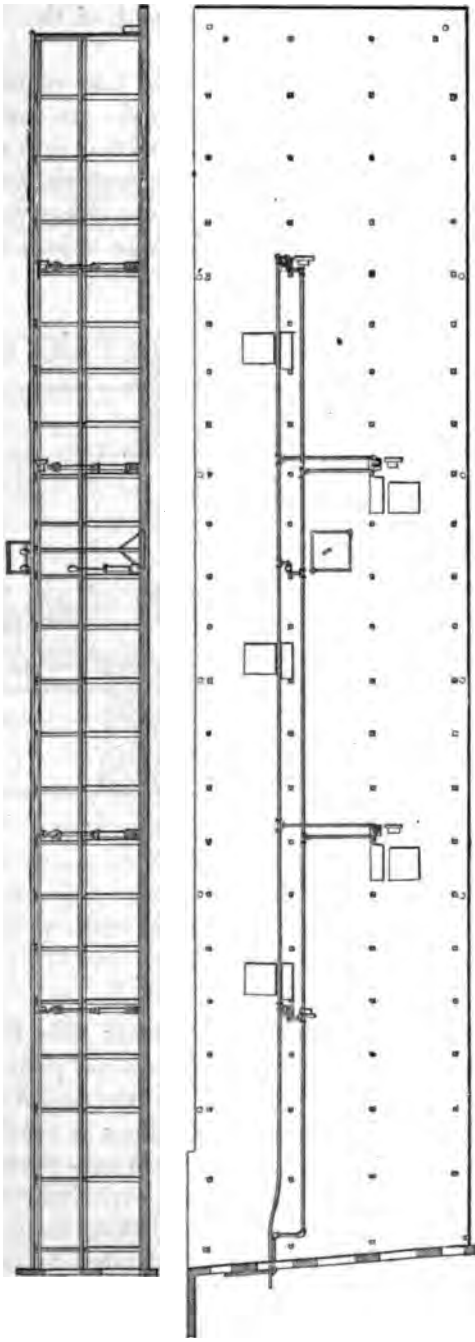


Fig. 102.

Fig. 102 shows the general arrangement of the hoists on the pier, and explains itself.

The choice of high pressure over low may require some explanation. Of course the chances for leaks are vastly increased by the use of high pressure, and the workmanship must be of a superior kind; but, weight for weight of material to do a certain amount of work, there is little to choose either way, while the great bulk of low-pressure machines was objectionable on account of limited space.

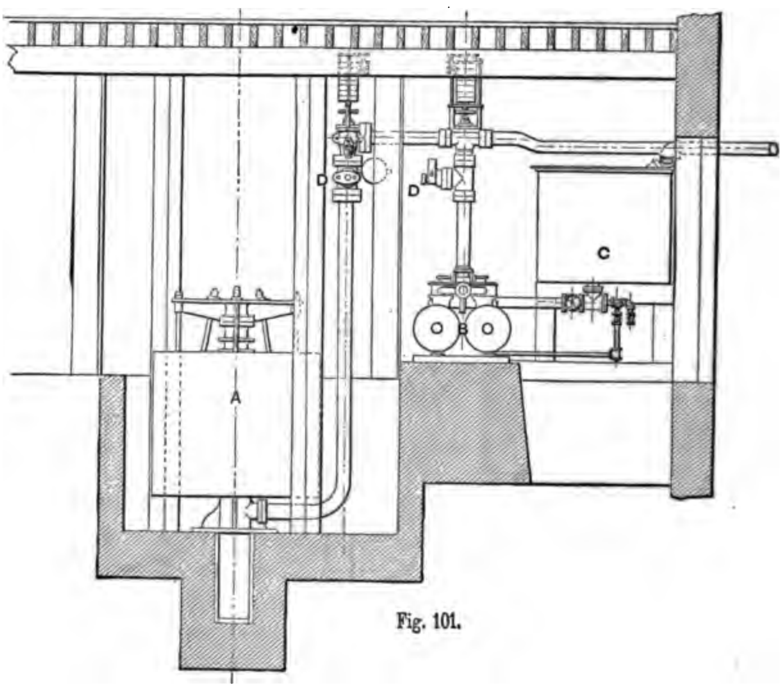


Fig. 101.

The system was constructed by Messrs. Otis Bros. & Co., of New York. It was necessary to make special patterns for every part of the work except the pumps, and the design of the details is due to Mr. Rudolph C. Smith, and follows as nearly as possible the requirements of the London Water Works System. Excepting the delay always experienced with experimental machinery, the plant has given satisfaction, but, as before mentioned, no comparison has been, or is likely to be, instituted by us between the relative economy of operation of this system and one based on separate steam-engines.

CCCCXXXI.*

EXPERIMENTAL DETERMINATIONS OF THE LATENT HEAT OF AMMONIA AND SULPHUR DIOXIDE.

BY D. S. JACOBUS, HOBOKEN, N. J.
(Member of the Society.)

THESE two fluids are at present the most generally relied upon for the production of artificial cold, which, as is well known, now forms an important and increasing field for the application of engineering ability. The latent heat of these fluids is the most important factor employed in calculations regarding the performance of a given plant, and for this reason the writer presents the following experimental figures, together with a description of an apparatus which has been constructed at the Stevens Institute, in order to verify results of experiments already made and to be used in original research on the matter.

The experimental values which have come under the notice of the writer are given below:

AMMONIA.

Experiments made by Regnault † show that the heat required to evaporate ammonia at the temperature of 11.67° C., or 53.01° Fahr., and the corresponding pressure, and to reheat the gas to this temperature after it has been cooled by passing it through an orifice which reduces its pressure to that of the atmosphere, is 294.2 calories per kilogramme.‡ This figure is greater than the latent heat of evaporation by the amount of heat required to reheat the gas. It is not possible to deduce the latent heat from the figure just given without employing theoretical values

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

† The results given are an average of twelve experiments published by Regnault in 1871, for the reference to which we are indebted to Mr. A. Siebert, of the De La Vergne Refrigerating Machine Co. (See *Annales de Chimie et de Physique*, 4th Series, Vol. XXIV., page 375.)

‡ To obtain this figure an assumed value has to be employed for the specific heat of the liquid ammonia. The value employed is .8, which is the same as is

for some of the quantities involved. The specific heat of the liquid ammonia also enters the problem, and, as this has not been determined, it adds another element of uncertainty. The probable latent heat deduced from Regnault's experiments at the above temperature is 289.8 calories per kilogramme. For further details, see the Appendix to this article.

SULPHUR DIOXIDE.

AUTHORITY.	Date at which Experiments were Published.	Conditions in regard to Temperature and Pressure under which the Latent Heat was Determined.	Temperature of Ebullition in degrees Fahr.	Latent Heat of Evaporation.	
				Calories per Kilogramme.	British heat units per Pound.
Favre and Silbermann *....	1853	Atmospheric Pressure....	13.86	94.56	170.21
Favre †....................	1874	" "	13.86	82.22	147.99
Chappuis ‡................	1888	32° Fahrenheit..........	32.00	91.7	165.06
Mathias §.................	1890	5.74° Centigrade	42.39	89.38	160.88
"	"	9.44 "	48.99	88.12	158.62
"	"	10.50 "	50.90	87.32	157.18
"	"	12.23 "	54.01	87.30	157.14
"	"	19.95 "	67.91	84.48	152.06

Regnault made a series of experiments on both ammonia and sulphur dioxide, but unfortunately the greater part of his records were lost during the Commune.‖ A series of twelve experiments on ammonia were, however, preserved, and although the results may not be as regular as some others which were lost,¶ they were made with all the characteristic accuracy of this eminent physicist, and agree among themselves to within about 4½%. The results of each of the experiments are shown in Table I. in the sequel.

An examination of Table I. will show that the average temperature of evaporation varied from 7.8° C. to 16° C. The experiments do not, however, agree well enough with each other to

used by Regnault in his computations. A value is given by Regnault for the amount of heat required to evaporate and afterward heat the expanded gas. This value is not precisely the same as the figure given above, because Regnault included the cooling effect produced by the expansion of gas that is in the calorimeter after the liquid has evaporated, whereas we have deducted this quantity.

* *Annales de Chimie et de Physique.* 3d Series, Vol. XXXVII., p. 470.

† *Ibid.* 5th Series, Vol. I., p. 251.

‡ *Ibid.* 6th Series, Vol. XV., p. 498.

§ *Ibid.* 6th Series, Vol. XXI., p. 116.

‖ *Annales de Chimie et de Physique*, 4th Series, Vol. XXIV., p. 380.

¶ *Annales de Chimie et de Physique*, 4th Series, Vol. XXIV., p. 429, 430.

allow an average to be taken of those made at a high and those made at a low temperature; for example, experiment No. 3, which is made at the highest temperature, should, by theory, give the lowest latent heat, whereas the value given in Table V. is about a mean of the others.

The results of Regnault's experiments are represented graphically in Fig. 103.

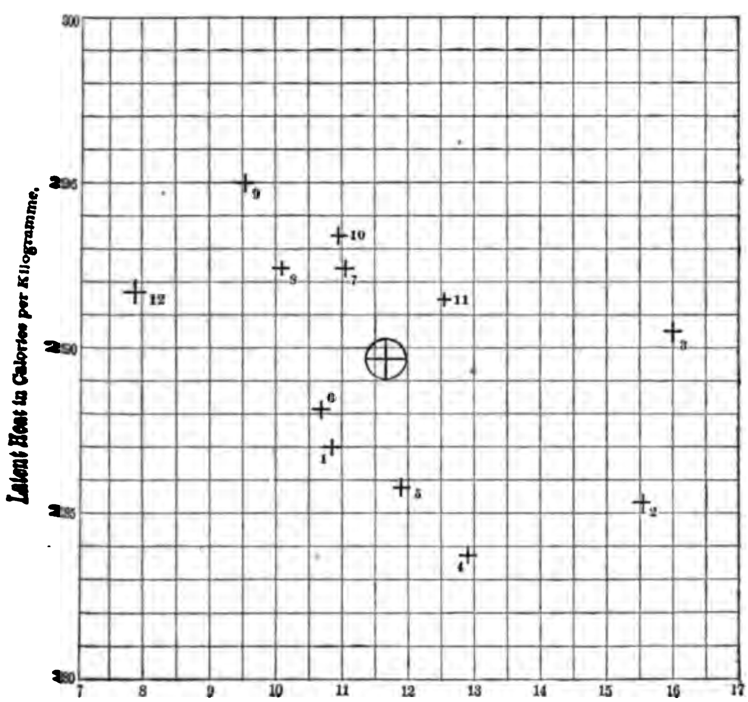


FIG. 103.

Regnault's apparatus consisted of the calorimeter of evaporation, represented in Fig. 104, together with a second calorimeter attached at m, which was used to determine the amount of heat absorbed by the expansion of the gas down to the pressure of the atmosphere. We have not represented the calorimeter of expansion. The apparatus shown in Fig. 104 consists of the vessel ABC, which is partly filled with liquid anhydrous ammonia. DEF and FLJ are chambers having deflecting plates arranged in them so that when the ammonia vapor passes through them it will come in contact with the sides of the chambers and be brought up to the temperature of the water in the vessel *GHIK*. The apparatus is allowed to stand for some time before making the experiments and the rate of radiation determined. On starting the experiment the cock *R'* is opened, which allows the ammonia vapor to pass into the vessels *DEF* and *FLJ*, and thence to a small orifice at *m* that produces a throttling action. The pressure of the vapor before passing through the orifice *m* is shown by a manometer attached at *z*. All the ammonia which is placed into the apparatus is allowed to evaporate during the experiment.

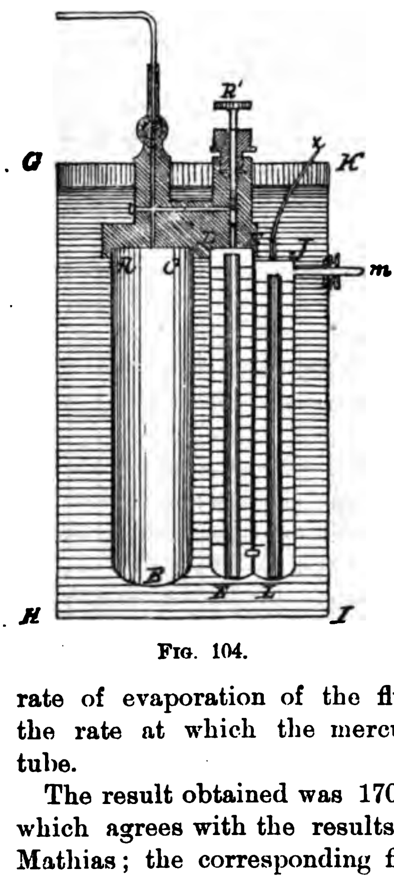


FIG. 104.

We will now consider the experiments made on sulphur dioxide:

The first experiment on the latent heat of sulphur dioxide, the results of which have been already given, were made by Favre and Silbermann in 1853. A mercury calorimeter was used, which may be regarded as a very large thermometer, into the bulb of which an open-ended tube is placed, so as to allow the substance on which the test is made to be enclosed partly by the mercury in the bulb, and thus cool it when it evaporates at the pressure of the atmosphere. The rate of evaporation of the fluid was observed, together with the rate at which the mercury receded in the thermometer tube.

The result obtained was 170.21 British heat units per pound, which agrees with the results obtained later by Chappuis and Mathias; the corresponding figure, calculated by means of an equation, found to agree with these experiments, being 172.28. The value found by Favre in 1874 is lower than it should be according to the experiments of Chappuis, and those made later by Mathias. Only two experiments were made in obtaining this result. As the accuracy of the experiments depended on a constant rate of ebullition, it may be slightly in error on account of some variation in the same, which, as is often remarked by Regnault, is extremely variable for fluids of high volatility. The result found by Chappuis, by means of an ice calorimeter, comes in line with the results of the later experiments made by Mathias.

We find in the experiments of Mathias an excellent and most useful example of physical research. In the article already cited he has reviewed the calorimeters used and methods employed in obtaining the latent heat of carbonic acid and sulphur dioxide, and after stating the reasons for adopting a special form of apparatus, he has indicated in detail all the steps gone through in making the experiments, and finally compared his results with those given by theory.

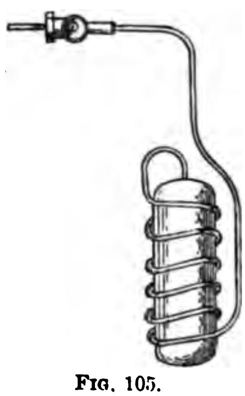


FIG. 105.

In the apparatus used by Mathias the liquid is evaporated at nearly a constant temperature. This is done by adding sulphuric acid to the water which surrounds the vessel in which the liquid is evaporated. A sketch of the evaporating vessel is shown in Fig. 105. The weight of sulphuric acid which is added is determined, so that by knowing its heat of solution the heat imparted to the evaporating vessel may be readily calculated. All the liquid is not evaporated from the vessel. The rate of evaporation is carried on slowly enough to preserve the temperature of the liquid inside of the vessel, at about the same temperature as that of the water which surrounds it; to prove which, a pressure gauge is connected to the apparatus, which remains at nearly a constant reading throughout the experiment.

The objections raised to Regnault's form of apparatus by Mathias are, that the liquid is not evaporated at a constant temperature, and that there are corrections to be applied to the work, which render it complicated.

The error of the work due to the variation of temperature at the beginning and end of the experiment is, however, a refinement that is so small that it will not be appreciated if the proper range is employed. Again, the claim that there are complicated corrections to apply is not intended to depreciate in any way the correctness of Regnault's work, which has often been shown to be worthy of general confidence.

The writer had occasion to review Regnault's work, with a view of checking experiments already made, and of extending the investigation. The apparatus presented in Fig. 106 has been designed and used to determine the latent heat of evaporation of ammonia. The piece *E*, which is a poor conductor of heat, and which was suggested by a similar arrangement in the apparatus used by Mathias, has been added since making the preliminary experiments.

The anhydrous ammonia is contained in the vessel *A*. This

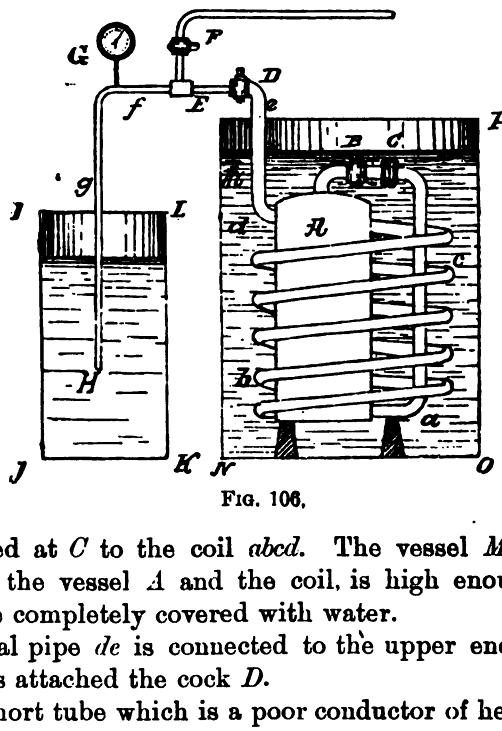


Fig. 106.

is connected at *C* to the coil *abcd*. The vessel *MNOP*, which surrounds the vessel *A* and the coil, is high enough to allow them to be completely covered with water.

A vertical pipe *de* is connected to the upper end of the coil, and to it is attached the cock *D*.

E is a short tube which is a poor conductor of heat. From *E* a tube leads to the vessel *IJKL*, in which the ammonia gas is absorbed when it issues from the apparatus. At the end of this tube is the capillary orifice *H*. *G* is a gauge, and *F* a cock, the use of which will be explained later.

In preparing to make an experiment we proceed as follows: The vessel *A* is nearly filled with liquid ammonia; it is then connected at *C*. The cocks *D* and *F* are closed and *B* opened, which allows the full pressure of the gas to act throughout the length of the coil. The cock *D* is then opened, and the gas allowed to escape for a short time through *H*, in order to expel the air that is contained in the coil.

The cock *D* is then closed, and the ammonia gas expelled from the tube *fg*, by allowing a small amount of air at a slight pressure to pass through the cock *F*. The cock *F* is then closed, and the vessel *IJKL*, containing an absorbent for the ammonia, placed so that the tube *fg*, at the end of which is the small orifice *H*, dips into it.

The vessel *MNOP* is now filled with a known weight of water. There is an agitator in this vessel so that the water may be thoroughly mixed before observing the temperature. After allowing the apparatus to stand for some time in order that all the parts may assume the same temperature, we are ready to commence the experiment.

We observe the temperature of the water in the vessel *MNOP*, and after five minutes have elapsed again note the temperature. The readings of temperature are taken to 0.01 of a degree Fahr.

This is done for two and sometimes three successive five-minute intervals, and the proper correction for radiation thus obtained.

The cock *D* is now opened to its full extent, so that the pressure due to the temperature of the liquid in *A* acts on one side of the orifice *H*, and on the gauge *G*. When the desired range of temperature has been obtained in the water contained in the vessel *MNOP*, we close the cock *D*, and observe the final reading of the temperature.

The cock *F* is now opened, and the small amount of ammonia gas contained in the tube *fg* is forced into the solution contained in the vessel *IJKL*.

The radiation of the vessel *MNOP* is determined at the close of the experiment in the same way that it was at the beginning. The gain of weight of the vessel *IJKL* gives the amount of ammonia expelled from the apparatus, from which, knowing the densities of the vapor and liquid ammonia, we can readily calculate the weight that is evaporated.

In calculating the results, it is necessary to know the total amount of liquid and vapor contained in the apparatus, in order to apply small corrections for the amount of heat imparted to the liquid and vapor which remain in the apparatus. This determined by weighing the flask A, before connecting it at C

The principal reason leading the writer to make so ma modifications in the apparatus devised by Regnault, is the des to remove complications which arise in the use of his appa tus. For example, on starting the experiment and openi the cock R′, Fig. 104, the ammonia gas first expands and fi the chambers DEF and FLJ with gas at a pressure sho by the manometer attached at z. This manometer did n register the pressure due to the temperature of the gas, t its readings were so irregular that they were not includ in the data. It is questionable whether, with such variations pressure, the temperature of the gas entering DEF and F did not fall below that of the water that surrounded it, and t gas become slightly superheated in passing through the t chambers.

Regnault appears to appreciate this fact, for he states that is best not to try and separate the latent heat, as given by t apparatus shown in Fig. 104, and the heat absorbed by exp sion as given by the expansion calorimeter. He states, howev that he considers the variation of the results which he obtain to be caused by the irregularity of the ebullition of the ammon which seems to be one of the characteristics of a very volat fluid. Another correction has to be applied to Regnault's resu on account of the fact that all the ammonia is allowed to esca from the apparatus at the end of the experiment. The meth employed by the writer to correct for the superheating effect given in the appendix.

The writer will give the society at some future time the resu of his experiments on ammonia, the object of the present pap being to precede the one containing the results and data c served, in order to show the necessity of performing the expe ments that will be gone through with.

It is interesting, in connection with this subject, to observe what degree of closeness various theoretical formulæ approac the values given by experiment. The formulæ for ammonia a given in Table II.; for sulphur dioxide, in Table III.

TABLE I.

RESULTS DEDUCED FROM THE EXPERIMENTS OF REGNAULT, ON THE LATENT HEAT OF AMMONIA.

No. of Experiment.	Average Temp. of Ebullition in Deg. Cent. $=\frac{\theta+\theta'}{2}$.	Probable Latent Heat of Evaporation in Calories per Kilogramme.	No. of Experiment.	Average Temp. of Ebullition in Deg. Cent. $=\frac{\theta+\theta'}{2}$.	Probable Latent Heat of Evaporation in Calories per Kilogramme.
1	10.900	287.0	7	11 040	292.5
2	15.529	285.2	8	10.150	292.4
3	16.000	290.5	9	9.520	295.0
4	12.940	283.8	10	10.990	293.3
5	11.900	285.8	11	12.600	291.6
6	10.725	288.1	12	7.800	291.8

TABLE II.

COMPARISON OF THE RESULTS DERIVED BY MEANS OF THEORETICAL FORMULÆ FOR THE LATENT HEAT OF AMMONIA, WITH THE RESULT FOUND BY REGNAULT.

Authority.	Date at which Formula was published.	Formulæ for Latent Heat of Evaporation.	Temperature substituted in Formulæ.		Latent Heat.			
					Probable Value deduced from Regnault's Experiments.		Calculated.	
			Cent.	Fahr.	Calories per Kilogramme	British Heat Units per Pound.	Calories per Kilogramme	British Heat Units per Pound.
Ledoux *...	1878	$309.7 - .8282t + .001398t^2$	11.67	53.01	289.8	521.6	300.2	540.4
Peabody †..	1889	$300-0.8t$	"	"	"	"	290.66	523.19
Wood ‡....	1889	$555.5-0.613t-.000219t^2$	"	"	"	"	290.22	523.39

The formulæ given above by Ledoux and by Peabody are in Centigrade units, and that by Prof. Wood is in British units. The constants in the formulæ given by Ledoux have been slightly altered by Prof. A. Riesenberger and the writer, as numerical mistakes were found in his work (see *Stevens Indicator*, Oct., 1890). In the above formulæ t is the temperature indicated by a thermometer scale, and is not the absolute temperature.

* *Annales des Mines.* July, 1878.

† *Thermodynamics of the Steam Engine.* By Prof. C. H. Peabody, p. 458.

‡ *Transactions of the American Society of Mechanical Engineers.* Vol. X., 1889, p. 641.

the liquid and vapor which remain in the apparatus. This is determined by weighing the flask *A*, before connecting it at *C*.

The principal reason leading the writer to make so many modifications in the apparatus devised by Regnault, is the desire to remove complications which arise in the use of his apparatus. For example, on starting the experiment and opening the cock R′, Fig. 104, the ammonia gas first expands and fills the chambers *DEF* and *FLJ* with gas at a pressure shown by the manometer attached at *z*. This manometer did not register the pressure due to the temperature of the gas, but its readings were so irregular that they were not included in the data. It is questionable whether, with such variations of pressure, the temperature of the gas entering *DEF* and *FLJ* did not fall below that of the water that surrounded it, and the gas become slightly superheated in passing through the two chambers.

Regnault appears to appreciate this fact, for he states that it is best not to try and separate the latent heat, as given by the apparatus shown in Fig. 104, and the heat absorbed by expansion as given by the expansion calorimeter. He states, however, that he considers the variation of the results which he obtained to be caused by the irregularity of the ebullition of the ammonia, which seems to be one of the characteristics of a very volatile fluid. Another correction has to be applied to Regnault's results on account of the fact that all the ammonia is allowed to escape from the apparatus at the end of the experiment. The method employed by the writer to correct for the superheating effect is given in the appendix.

The writer will give the society at some future time the results of his experiments on ammonia, the object of the present paper being to precede the one containing the results and data observed, in order to show the necessity of performing the experiments that will be gone through with.

It is interesting, in connection with this subject, to observe to what degree of closeness various theoretical formulæ approach the values given by experiment. The formulæ for ammonia are given in Table II.; for sulphur dioxide, in Table III.

APPENDIX.

Method of Calculating the Latent Heat of Evaporation of Ammonia from the Experiments Made by Regnault.

Regnault gives for the value of the latent heat,*

$$\frac{Q'}{P'} = \frac{M\Delta\theta - q + P'C\frac{\theta - \theta'}{2}}{P'}; \quad . \quad . \quad . \quad . \quad . \quad (1)$$

in which

Q' = heat imparted to evaporate the weight P' of liquid contained in the vessel ABC before opening the cock R' (Fig. 104), or the latent heat of evaporation multiplied by P';

M = the weight of water in first calorimeter plus the calorific equivalent of the metal of which the calorimeter is composed;

$\Delta\theta = \theta - \theta'$ = difference of temperature of the water in the first calorimeter at the beginning and end of the experiment;

q = heat absorbed by the weight of gas $P - P'$, contained in the vessel ABC before the cock R' is opened, on expanding it to the pressure of the atmosphere; and

C = specific heat of the ammonia liquid.

$P'C\frac{\theta - \theta'}{2}$ is the heat that the weight P' of liquid gives upon being cooled from the initial temperature θ to the average temperature $\frac{\theta + \theta'}{2}$.

To find q, Regnault assumes that the effect of the expansion of the weight of gas $P - P'$ will be the same in the actual experiment as if it were contained in a separate vessel and allowed to escape at a varying pressure until the pressure in the vessel becomes that of the atmosphere; the temperature of the vessel being maintained constant and the proper amount of heat

* *Annales de Chimie et de Physique*, Series IV., Vol. XXIV., p. 420.

TABLE III.

COMPARISON OF THE RESULTS DERIVED BY THEORETICAL FORMULÆ FOR THE LATENT HEAT OF SULPHUR DIOXIDE, WITH THE RESULTS FOUND BY EXPERIMENT.

Temperature substituted in Formulæ.		Latent Heat by Experiment.		Calculated Latent Heat.									
				Ledoux, 1878.* $91.54-.2339t+.0002357t^2$.		Cailletet and Mathias, 1887,† $\frac{T}{E}(u'-u)\frac{dp}{dt}$.		Peabody, 1889,‡ $98-0.27t$.		Mathias, 1890, § $91.87-0.384t-0.000340t^2$.		Mathias, 1890.‖ $91.87-0.384t$	
C.	Fahr.	Calories per Kilo.	B. H. U. per Pound.	Calories per Kilo.	B. H. U. per Pound.	Calories per Kilo.	B. H. U. per Pound.	Calories per Kilo.	B. H. U. per Pound.	Calories per Kilo.	B. H. U. per Pound.	Calories per Kilo.	B. H. U. per Pound.
−10.08	13.86	94.56	170.21	93.92	169.06	93.7	168.66	100.72	181.30	95.71	172.28	95.74	172.33
0	32.	91.70	165.06	91.54	164.77	91.2	164.16	98.00	176.42	91.87	165.37	91.87	165.37
5.74	42.33	89.38	160.88	90.21	162.38	89.2	160.56	96.45	173.61	89.66	161.39	89.67	161.41
9.44	48.99	88.12	158.62	89.35	160.83	88.8	159.84	95.45	171.81	88.22	158.79	88.25	158.85
10.50	50.90	87.32	157.18	89.11	160.40	88.5	159.30	95.17	171.31	87.80	158.04	87.84	158.11
12.23	54.01	87.30	157.14	88.71	159.68	87.8	158.04	94.70	170.46	87.12	156.82	87.17	156.91
19.95	67.91	84.48	152.06	86.97	156.65	84.7	152.46	92.62	166.70	84.07	151.33	84.21	151.58

The experimental figure at − 10.08° is that determined by Favre and Silbermann, and that at 0° by Chappuis.

The remainder are those determined by Mathias.

The temperatures indicated by t are those indicated by a Centigrade thermometric scale, and are not absolute temperatures.

---

* *Annales des Mines*. July, 1878.

† *Journal de Physique*. 1887, pp. 423 and 424. In this formula,

$T = t + 273$.

$E =$ the mechanical equivalent of heat $= 425$.

$u' =$ the specific volume of vapor at t degrees.

$u =$ specific volume of liquid at t degrees.

$p =$ pressure at temperature t.

The results given above, as determined by this formulæ, are found by interpolating in a table the numerical values given by Cailletet and Mathias.

‡ *Thermodynamics of the Steam Engine*. By Prof. C. H. Peabody.

§ *Annales de Chimie et de Physique*. 6th Series, Vol. XXI., p. 117.

‖ *Annales de Chimie et de Physique*. 6th Series, Vol. XXI., p. 118.

APPENDIX.

METHOD OF CALCULATING THE LATENT HEAT OF EVAPORATION OF AMMONIA FROM THE EXPERIMENTS MADE BY REGNAULT.

REGNAULT gives for the value of the latent heat,*

$$\frac{Q'}{P'} = \frac{M\Delta\theta - q + P'C\frac{\theta - \theta'}{2}}{P'}; \quad . \quad . \quad . \quad . \quad . \quad (1)$$

in which

Q' = heat imparted to evaporate the weight P' of liquid contained in the vessel ABC before opening the cock R' (Fig. 104), or the latent heat of evaporation multiplied by P';

M = the weight of water in first calorimeter plus the calorific equivalent of the metal of which the calorimeter is composed;

$\Delta\theta = \theta - \theta'$ = difference of temperature of the water in the first calorimeter at the beginning and end of the experiment;

q = heat absorbed by the weight of gas $P - P'$, contained in the vessel ABC before the cock R' is opened, on expanding it to the pressure of the atmosphere; and

C = specific heat of the ammonia liquid.

$P'C\frac{\theta - \theta'}{2}$ is the heat that the weight P' of liquid gives upon being cooled from the initial temperature θ to the average temperature $\frac{\theta + \theta'}{2}$.

To find q, Regnault assumes that the effect of the expansion of the weight of gas $P - P'$ will be the same in the actual experiment as if it were contained in a separate vessel and allowed to escape at a varying pressure until the pressure in the vessel becomes that of the atmosphere; the temperature of the vessel being maintained constant and the proper amount of heat

* *Annales de Chimie et de Physique*, Series IV., Vol. XXIV., p. 420.

added to the gas after it passes through the throttling orifice to bring it to the temperature of the gas in the vessel.

In which case the amount of heat absorbed in order to preserve the temperature θ would be *

$$q = \frac{10333}{1.2932 \times .5894} \cdot \frac{P - P'}{e} (1 + \alpha\theta), \quad . \quad . \quad . \quad (2)$$

α being the coefficient of dilatation of the gas.

Regnault states, after giving equation (1), that as the specific heat C is not known, he did not calculate the values of the latent heats of evaporation, but he afterward employs the value .8 in his calculations.

There is an element, however, for which Regnault does not entirely allow in equation (1), and that is the cooling effect produced by the total quantity of gas which remains in the vessel ABC after the liquid is evaporated.

This gas expands from the pressure at which the liquid is boiled down to the pressure of the atmosphere, and thus absorbs an additional quantity of heat that should be subtracted from the total quantity in order to obtain Q'. A portion of this heat is allowed for in the quantity q, for we may consider the weight of gas $P - P'$ to be held intact in the calorimeter until the time that all the liquid is evaporated, and then allow it to

* The value of e is determined by the use of an apparatus equivalent to the one shown in Fig. 107. The gas is compressed in the cylinder A, and, after being brought to a known temperature, is allowed to escape through a small orifice at m, and thence into the coil BCD, where it is at the pressure of the atmosphere and has a slow velocity. The value of e is determined by the use of the equation given on page 482 of *Annales de Chimie et de Physique*, Series IV., Vol. XXIV., which is

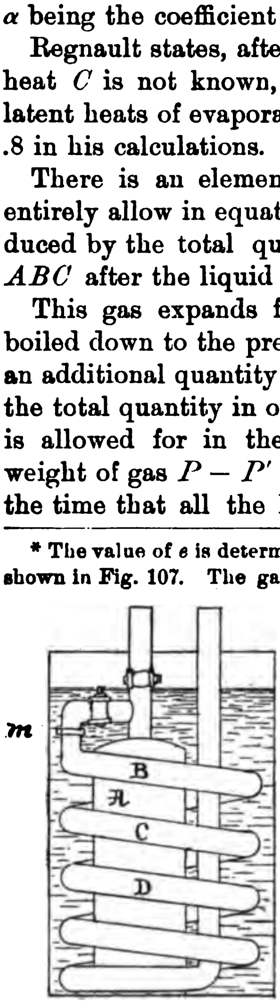


FIG. 107.

$$e = \frac{10333}{1.2932 \times .5894} \frac{\Pi - \pi}{Q} \left(1 + \alpha \frac{\theta + \theta'}{2}\right);$$

in which

Π = initial weight of gas in A.
π = final " " "
Q = heat imparted to the calorimeter after the proper connections have been made for the superheating of the gas that remains in the calorimeter, etc.

α = coefficient of dilatation of the gas.
1.2932 = weight in grammes of one litre of air at 0° C. and at atmospheric pressure
0.5894 = density of the ammonia vapor.

expand along with the other portion of gas that fills the vessel *ABC*.

The portion of the gas which is in the vessels *DEF* and *FLJ* at the time that all the liquid is evaporated from *ABC* also produces a cooling effect, but this is very nearly balanced by the heat which is produced by compressing the gas in them at the beginning of the experiment, and may be, therefore, omitted.

In order to determine the latent heat of evaporation from the results obtained from the first calorimeter, we will make use of equation (1), but will substitute for q another quantity, which we will call q_1. The formula employed in the present article for calculating the latent heat is, therefore,

$$\lambda' = \frac{Q'}{P'} = \frac{M\Delta\theta - q_1 + P'C\,\frac{\theta - \theta'}{2}}{P'}; \quad . \quad . \quad . \quad . \quad (3)$$

in which q_1 = the heat necessary to impart to the weight W of the gas that is in the vessel ABC (Fig. 104) at the time that the liquid has just become completely evaporated, in order to preserve it at a constant temperature during its expansion down to the pressure of the atmosphere. The other quantities are the same as those in equation (1). The weight of gas W which is in the vessel ABC at the instant that all the liquid is evaporated is substituted in place of $P - P'$ in equation (2), in order to determine q_1. This gives

$$q_1 = \frac{10333}{1.2932 \times .5894}\,\frac{W}{e}\,(1 + \alpha'\theta'). \quad . \quad . \quad . \quad (4)$$

To find the weight of gas W, we assume it to be at the final temperature θ' of the calorimeter, and make use of the formula given on page 418 of the work already mentioned, which is,

$$W = 0.18774\,\frac{1}{1 + \alpha' t}\,\frac{H}{760}; \quad . \quad . \quad . \quad . \quad . \quad (5)$$

in which α' = coefficient of dilatation of the ammonia gas;
in which H = pressure in m.m. of mercury.

In our calculations for determining q_1, H is taken equal to F_{θ}', and t equal to θ'. There are discrepancies in the numerical values given for $P - P'$ in Regnault's work. We have recalculated $P - P'$ and all the quantities which depend on its value. Regnault takes the specific gravity of ammonia at .76; we have used the figure determined by D'Andreeff instead of this figure, which, for the average temperature of the experiment, is .62.*

To determine $P - P'$ we make use of the following formula:

$$P - P' = \left(1 - \frac{P}{.62 \times 246.3}\right) \times 0.18774 \frac{1}{1 + \alpha' t} \frac{H}{760}; \quad . \quad . \quad (6)$$

in which the notation is the same as in equation (5).

In equation (6) $.62 \times 246.3$ is equal to the weight of liquid ammonia that will completely fill the vessel ABC. As the gas $P - P'$ is at the initial temperature of the calorimeter, H is taken equal to F_0 and t to θ. Regnault has assumed the specific heat of the liquid C to equal .8. We have employed this value in Tables IV. and V., together with the theoretical density of the gas. In Table VI. the specific heat of unity, with the value given by theoretical investigations, is employed, together with the density of the gas determined by experiment. It is shown that the effect of altering the value of these quantities is a small one. The values of $M\Delta\theta$ are taken directly from Regnault's work.

The results of the calculations are given in Table IV.

TABLE IV.

REGNAULT'S EXPERIMENTS ON THE LATENT HEAT OF AMMONIA—RESULTS OBTAINED FROM THE FIRST CALORIMETER.

Number of experiment.	θ Degree C.	θ' Degree C.	$\frac{\theta + \theta'}{2}$ Degree C.	$P - P'$ Grammes.	P' Grammes.	W Grammes.	F_0 m.m.	$F_{\theta'}$ m.m.	$\frac{F_0 + F_{\theta'}}{2}$ m.m.	$M\Delta\theta$	q_1	S'	Q'	λ'
1	12.31	9.49	10.90	.94	30.74	1.07	4966.4	4486	4726.9	9002.3	47.3	34.6	8989.6	292.4
2	16.37	14.68	15.53	1.17	17.68	1.26	5687.6	5364	5525.7	5184.6	56.7	11.9	5139.8	290.7
3	16.85	15.16	16.00	1.19	17.31	1.28	5780.2	5452	5616.3	5183.8	57.7	11.7	5137.8	296.8
4	14.53	11.35	12.94	.98	33.32	1.13	5340.	4770	5055.	9649.9	50.3	42.3	9641.9	289.4
5	14.07	9.72	11.90	.87	45.23	1.08	5255.	4525	4890.	13077.3	47.8	78.7	13108.2	289.8
6	16.36	5.09	10.73	.29	119.21	.90	5666.	3718	4692.	34235.	39.2	537.8	34733.1	291.4
7	15.07	7.00	11.04	.58	83.12	.98	5424.	4060	4742.	24151.	42.9	268.3	24376.4	293.3
8	16.36	3.94	10.15	.16	134.13	.86	5666.	3518	4592.	38545.	37.3	666.4	39174.1	292.1
9	15.10	3.93	9.52	.30	117.00	.87	5440.	3544	4492.	34248.	37.2	522.8	34783.1	296.9
10	14.81	7.17	10.99	.62	79.18	.99	5388.	4098	4743.	23044.	43.4	242.0	23242.6	293.6
11	17.34	7.87	12.60	.50	97.05	.99	5850.	4138	4994.	28208.4	43.5	367.6	28592.5	294.0
12	10.23	5.37	7.80	.73	51.67	.93	4610.	3850	4230.	15186.3	40.0	100.4	15246.7	295.1

The quantity λ' would be equal to the latent heat of evaporation, provided there was no superheating of the partly expanded gas in the first calorimeter. The average value obtained for λ' is 293.0 calories per kilogramme. This is less than the average value of λ'' in Table V., which includes the heat required to reheat the gas after it is expanded to the pressure of the atmosphere. In some of the experiments, however, λ' is as great as λ'', thus showing that superheating existed in the first calorimeter, so that the results obtained by it are unreliable.

* *Annales de Chimie et de Physique*, 3d Series, Vol. LVI., p. 327.

Regnault, however, remarks that it is best not to separate the results given by the first calorimeter and the calorimeter of expansion, and proceeds to calculate the total heat of evaporation by employing the results given by both the calorimeters. This is by far the best method, for the reason that, in order to regulate the ebullition during the experiments, the cock R', Fig. 104, was not entirely opened, and, as has been already remarked, there may have been a superheating effect in the vessels DEF and FLJ. That this superheating effect existed is shown by examining the quantities designated by λ' and λ'' in Tables IV. and V. If there was no superheating effect in the first calorimeter, λ' would be equal to the latent heat, which would be a certain amount less than the quantity λ'', which includes the heat added to the expanded gas. λ' is, however, in some cases, as great as λ'', thus showing that superheating existed in the first calorimeter. For this reason, the results obtained by employing both calorimeters are made use of in this article. The writer does not, however, consider that all the steps gone through by Regnault in deducing the values of a quantity designated by L, and which is stated to nearly equal the total heat of evaporation, are as straightforward as they might be, and for this reason has employed methods differing from Regnault in obtaining some of the quantities involved. The following formula has been used in the present article in order to determine the latent heat of evaporation;

$$\lambda''' = \frac{Q''}{P'} = \frac{1}{P'}[M\Delta\theta + M'\Delta\tau - S' + S'' - q' - q_1]\,; \quad . \quad (5)$$

in which the notation is similar and many of the quantities the same as those given by Regnault.

The additional quantities to those already defined are:

$M'\Delta\tau$ = heat imparted to the gas by the calorimeter of expansion.

S' = heat required to raise the temperature of the gas that passes into the second calorimeter from the average temperature of the first calorimeter, $\frac{\theta + \theta'}{2}$, to the average temperature of the second calorimeter, $\frac{\tau + \tau'}{2}$.

q' = heat absorbed by the weight $P - W$ of the gas after it has been cooled by expanding through the throttling orifice, in order to raise the temperature of the expanded gas to that of the first calorimeter, or $\frac{\theta + \theta'}{2}$.

$S'' = P'C\frac{\theta - \theta'}{2}$, a quantity already described.

Equation (5) is derived as follows:

The total heat imparted to the gas by the two calorimeters is $M\Delta\theta + M'\Delta\tau$; this, together with the small amount of heat S'' given up by the cooling of the liquid ammonia in the first calorimeter, is the total heating effect.

The heat $M\Delta\theta + M'\Delta\tau + S''$ is absorbed by the gas in the following portions:

1st. The heat Q''' required to supply the latent heat of evaporation of $P - P'$ of ammonia liquid which is contained at the beginning of the experiment in the first calorimeter.

2d. The heat q' which is required to heat the weight $P - W$ of gas to the mean temperature of the first calorimeter after it is cooled by passing through the throttling orifice.

3d. The heat q_1 which it is necessary to impart to the weight W of gas that remains in the vessel ABC in the first calorimeter (see Fig. 104), after the liquid has evaporated and the pressure in the vessel begins to fall. This heat preserves the gas at a constant temperature in the vessel, and reheats it, after it is cooled by passing through the throttling orifice, to the mean temperature of the first calorimeter.

4th. The heat S' which is required to raise the temperature of the weight of gas $P - W'$, which passes into the second calorimeter, from the mean temperature of the first calorimeter to that of the second calorimeter.

This gives the equation:

$$M\Delta\theta + M'\Delta\tau + S'' = Q''' + q' + S' + q_1,$$

from which we may readily determine the value of the latent heat $\frac{Q'''}{P'}$ as given in equation (5).

The equations for determining S'' and q' have already been given; those for S' and q' are as follows:

$$S' = (P - W')\, c\left(\frac{\tau + \tau}{2} - \frac{\theta - \theta'}{2}\right), \text{ and } q' = c(P - W)\, y_1\frac{f - H}{1000};$$

TABLE VI.

RESULTS OBTAINED FROM THE TWO CALORIMETERS IF THE SPECIFIC HEAT OF THE LIQUID AMMONIA IS ASSUMED TO BE UNITY INSTEAD OF .8, AND THE DENSITY OF THE GAS .596 INSTEAD OF .5894.

No. of experiment.	$P-P$	P'	S'	S''	q'	q_1	Q''	λ'''
1	0.95	30.73	79.2	43.2	154.4	47.8	8820.4	287.0
2	1.18	17.67	20.7	14.9	106.6	57.4	5038.9	285.2
3	1.20	17.30	17.4	14.6	106.2	58.4	5024.9	290.5
4	0.99	33.31	27 5	52.9	181.2	50.8	9453.6	283.8
5	0.88	45.22	23.4	98.4	236.3	48.3	12922.2	285.8
6	0.30	119.20	270.0	671.6	592.7	39.6	34338.1	288.1
7	0.59	83.11	109.8	335.4	418.7	43.4	24312.2	292.5
8	0.17	184.13	338.6	833.0	649.0	37.7	39219.9	292.4
9	0.30	117.00	260.0	653.5	552.3	38.1	34519.8	295.0
10	0.62	79.18	47.9	302.5	399.5	43.9	23223.8	293.3
11	0.51	97.04	216.7	459.0	519.6	44.0	28296.0	291.6
12	0.74	51.66	138.7	125.5	227.0	41.0	15073.5	291.8

Average value of probable latent heat of evaporation.............. 289.8

CCCCXXXII.*

PERFORMANCE OF SEVENTY-FIVE TON REFRIGERATING MACHINE OF THE AMMONIA COMPRESSION TYPE.

BY J. E. DENTON, HOBOKEN, N. J.
(Member of the Society.)

INTRODUCTION.

THE following investigation was made with a refrigerating machine of 75† tons maximum rated capacity, located at the Knickerbocker Brewery, in Seventeenth Street, New York City, and constructed by the Consolidated Ice-machine Company, of Chicago, Ill. The work was undertaken in the interest of the Quincy Cold Storage Company, of Boston. Messrs. J. C. Melvin and G. H. Stoddard, officers of this company, having in charge the construction of an extensive refrigerating system for the market districts of Boston, found, after an extensive study of the subject of mechanical refrigeration under the ammonia-compression system, in both this country and Europe, that no experimental data were apparently available regarding the performance of such machines when worked to produce temperatures in brine or other material in the neighborhood of zero Fahr., the temperature which they desired to have available in their plant. They accordingly arranged with makers of ammonia-compression machines, desiring to bid for their work, to submit their systems to a rigorous test of performance, and the writer was consulted regarding the making of such tests. After due discussion of the matter, it was concluded that there was so little probability of any sensible difference of performance between the several first-class makes of compression machines, that a careful test of one make would answer all purposes in view. Accordingly, the machine at the Knickerbocker

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

† While the machine performed 75 tons of work during the tests, it is proper to state that it is rated, and was sold by the makers, as a 50-ton machine.

Brewery was selected, first because it seemed to combine maximum simplicity with first-class construction and finish; second, because it was near the writer's place of business, and therefore better available for attention; and third, because it was the only machine in the near neighborhood of New York whose makers were among those consenting to submit to a test at the hands of an expert not in their own employ. As the writer wished to make the occasion one for obtaining certain data, desirable as furnishing certain constants, needed to confirm the thermodynamic formulæ applying to the theory of these machines, but which was not essential for the purposes of the Boston Company, the latter's arrangement with him was that through them would be provided the means, for whatever facilities and expenses should be connected with the carrying out of as complete a programme as he might regard desirable, and that the results should be available for publication as a purely scientific investigation, on the understanding that there should be no charge for personal services. This arrangement was cordially carried out. The Ice-machine Company was directed to prepare the machine for test as thoroughly as they chose. Early in the present year they cleaned the surface of the condenser pipe, adjusted the piston packings, ground the compressor valves to new bearings and added 100 lbs. of ammonia to the charge in the machine. The latter had been in service for the purposes of the brewery about two years without repairs. The charge of ammonia is usually 400 lbs.; it had wasted about 100 lbs. during a year's daily service. Indicator connections were attached, and mercury wells let into the pipes, and water and brine meters connected, about the 1st of March. As, at this season of the year, but a small fraction of the capacity of the machine could be absorbed by the product of the brewery, the brine circulated was sent through a No. 4 Nason feed-water heater, so that it could be heated by a current of steam or hot water circulated through the spiral coil of pipe with which it is provided. By this means any desired amount of work could be assigned to the machine. About ten days of practice was required to enable this artificial source of heat to the brine to be controlled with sufficient skill to provide conditions of satisfactory steadiness. On March 13th a twenty-four hours' test was made, in which it was attempted to secure the utmost capacity consistent with a temperature of brine about equal to the melting point of ice. This

CCCCXXXII.*

PERFORMANCE OF SEVENTY-FIVE TON REFRIGERATING MACHINE OF THE AMMONIA COMPRESSION TYPE.

BY J. E. DENTON, HOBOKEN, N. J.
(Member of the Society.)

INTRODUCTION.

THE following investigation was made with a refrigerating machine of 75† tons maximum rated capacity, located at the Knickerbocker Brewery, in Seventeenth Street, New York City, and constructed by the Consolidated Ice-machine Company, of Chicago, Ill. The work was undertaken in the interest of the Quincy Cold Storage Company, of Boston. Messrs. J. C. Melvin and G. H. Stoddard, officers of this company, having in charge the construction of an extensive refrigerating system for the market districts of Boston, found, after an extensive study of the subject of mechanical refrigeration under the ammonia-compression system, in both this country and Europe, that no experimental data were apparently available regarding the performance of such machines when worked to produce temperatures in brine or other material in the neighborhood of zero Fahr., the temperature which they desired to have available in their plant. They accordingly arranged with makers of ammonia-compression machines, desiring to bid for their work, to submit their systems to a rigorous test of performance, and the writer was consulted regarding the making of such tests. After due discussion of the matter, it was concluded that there was so little probability of any sensible difference of performance between the several first-class makes of compression machines, that a careful test of one make would answer all purposes in view. Accordingly, the machine at the Knickerbocker

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

† While the machine performed 75 tons of work during the tests, it is proper to state that it is rated, and was sold by the makers, as a 50-ton machine.

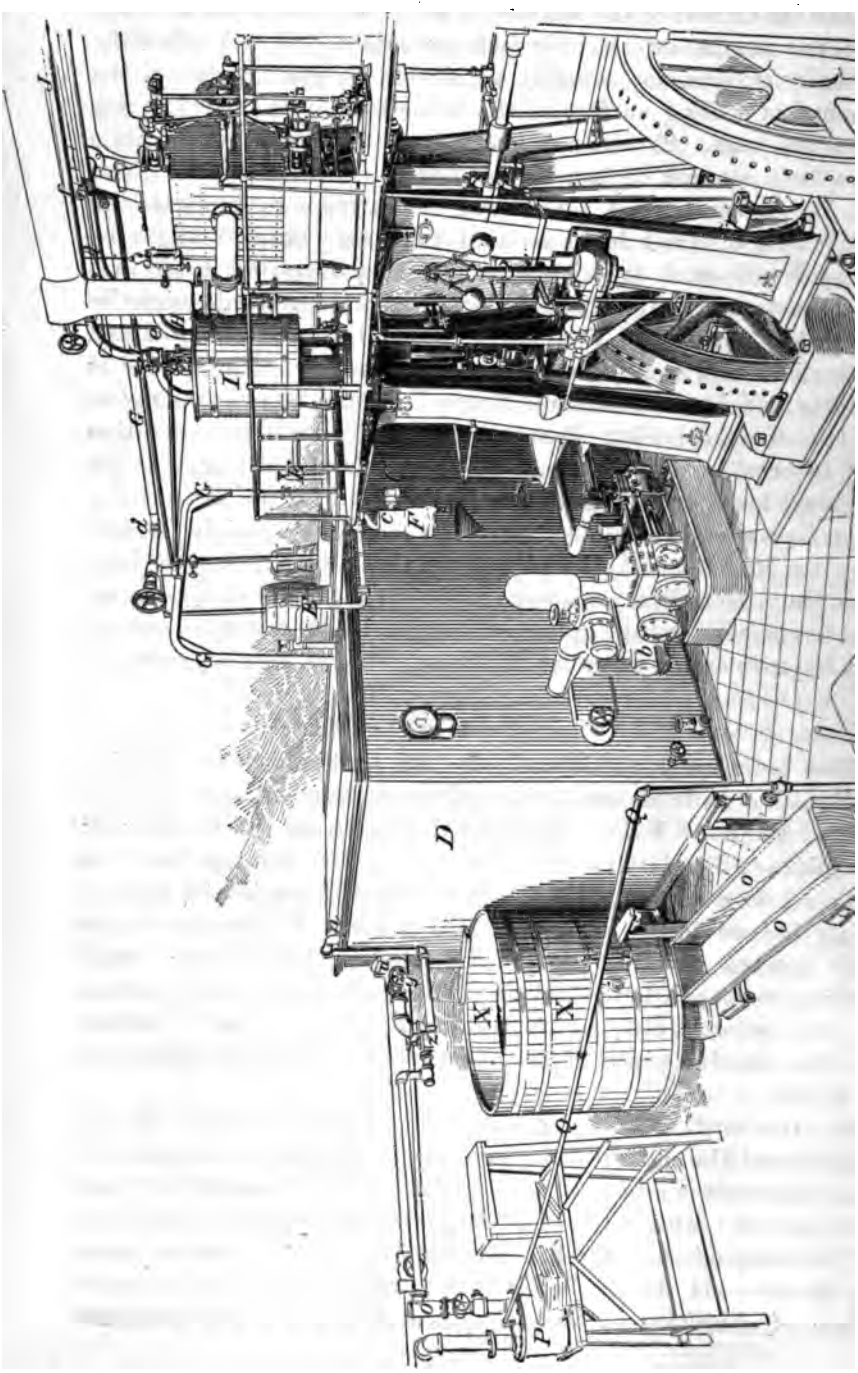
Brewery was selected, first because it seemed to combine maximum simplicity with first-class construction and finish; second, because it was near the writer's place of business, and therefore better available for attention; and third, because it was the only machine in the near neighborhood of New York whose makers were among those consenting to submit to a test at the hands of an expert not in their own employ. As the writer wished to make the occasion one for obtaining certain data, desirable as furnishing certain constants, needed to confirm the thermodynamic formulæ applying to the theory of these machines, but which was not essential for the purposes of the Boston Company, the latter's arrangement with him was that through them would be provided the means, for whatever facilities and expenses should be connected with the carrying out of as complete a programme as he might regard desirable, and that the results should be available for publication as a purely scientific investigation, on the understanding that there should be no charge for personal services. This arrangement was cordially carried out. The Ice-machine Company was directed to prepare the machine for test as thoroughly as they chose. Early in the present year they cleaned the surface of the condenser pipe, adjusted the piston packings, ground the compressor valves to new bearings and added 100 lbs. of ammonia to the charge in the machine. The latter had been in service for the purposes of the brewery about two years without repairs. The charge of ammonia is usually 400 lbs.; it had wasted about 100 lbs. during a year's daily service. Indicator connections were attached, and mercury wells let into the pipes, and water and brine meters connected, about the 1st of March. As, at this season of the year, but a small fraction of the capacity of the machine could be absorbed by the product of the brewery, the brine circulated was sent through a No. 4 Nason feed-water heater, so that it could be heated by a current of steam or hot water circulated through the spiral coil of pipe with which it is provided. By this means any desired amount of work could be assigned to the machine. About ten days of practice was required to enable this artificial source of heat to the brine to be controlled with sufficient skill to provide conditions of satisfactory steadiness. On March 13th a twenty-four hours' test was made, in which it was attempted to secure the utmost capacity consistent with a temperature of brine about equal to the melting point of ice. This was attained with 28 lbs. back pressure. The results are those of Trial No. 1, Table I. The degree of success in maintaining steady conditions is shown by the graphical record of observation, Fig. 112.

It was found that when the machine was doing all of the work consistent with the fixed back pressure of 28 lbs., the temperature of the gas about 10 feet beyond the point where it left the brine tank was about 34° Fahr. or about the temperature of the inlet brine. There was then no escape of gas at the stuffing box of the compressor detectable by smell. But as soon as it was attempted to raise the back pressure above this point, a portion of the ammonia would leave the brine tank in liquid form, and instantly the temperature would fall from 34° to 14°, the boiling point corresponding to the back pressure. When this occurred a strong smell of ammonia emanated from the stuffing boxes of the compressor. This method of finding the maximum work corresponding to a given back pressure was carried out throughout all the tests; and for each back pressure the work given, in tons of capacity, and the brine temperature, are respectively the greatest and least to be had with the given suction pressure. An examination of the line on the graphical charts Figs. 112 to 119, showing the temperatures at the point 10 feet beyond the brine tank in the return ammonia pipe, will show that the temperatures were kept at just about the point where liquid ammonia would occasionally come over, thereby giving warning that the suction pressure must be slightly reduced. During Test No. 1, an expert mechanic representing the machine company was present and virtually in charge of the machine. During all subsequent tests no representative from the company was present, the machine being run entirely under the direction of the writer, through Mr. Kellogg the regular engineer of the brewery. The second test made affords data regarding the capacity and economy of the machine at about zero pounds suction pressure above the atmosphere. It is less reliable than subsequent tests, as it was only intended as an introduction to the latter. The figures of performance are, however, quite sufficiently reliable for general practical purposes. Trial No. 3 was preliminary to trial No. 4, which is the typical test for maximum capacity and economy for zero temperature of brine. Trial No. 5 is preliminary to No. 6, which is typical of the influence, upon economy, of an excessive supply of condensing water. Comparison of Trials 5 and 6 also shows that whether the amount of brine circulated was 30 cubic feet per minute or 15 cubic feet per minute did not affect the capacity or economy sensibly, as should be the case when the amount of cooling surface in the brine tank is as liberal as was here the case. Trial No. 7 was made to afford thermodynamic data for a suction pressure intermediate between that of Trial No 1 and Trial No. 4. Trial No. 8 was made to duplicate the conditions of Trial No. 1 so that the data obtained might be favored with such experience as had been acquired in the previous tests, and so that the apparatus and measurements might be subjected to a test of their ability to duplicate results obtained under identical conditions six weeks previously. The writer is greatly indebted to Messrs. Rensselaer and Schalk, managers of the Knickerbocker Brewery, for their cordial coöperation and interest in the investigation, and the facilities placed at his disposal for specific heat determination, and for the use of lodgings for assistants required to be on duty during night hours; also to Mr. Edward Murphy of the Consolidated Ice Machine Company, for skilful assistance in the arrangement of apparatus, and in carrying out the apparently original and somewhat dangerous experiment of metering liquid ammonia at high pressure.

DESCRIPTION OF PLANT AND APPARATUS.

The cylindrical receiver *A*, Fig. 108, contains about 300 lbs of liquid anhydrous ammonia under a pressure of, say, 150 lbs., and at about 70° Fahr. This liquid flows from the receiver by the pipe *B*, Fig. 110, to the valve *C*, Fig. 108, through which it passes into a system of piping distributed throughout a mass of brine lodged in the brine tank *D*, Fig. 108. At one end of this pipe system are the pipes *EEE*, running out of the small receiver or expansion vessel, *F*. At the other end of the system are the pipes *GGG*, which terminate in the pipes *II*, which are the suction pipes of the compression pumps *II*, Figs. 108 and 109.

By the suction of these pumps a pressure of, say, 7 lbs. is maintained throughout the pipe system, *EE*, etc., so that the liquid ammonia is relieved of the high pressure in passing through the valve *C*. This relief of pressure causes about ten per cent. of the ammonia to volatilize, and thereby cool the entire mass to about − 11° Fahrenheit, which corresponds to the temperature of ebullition of the substance at about 7 lbs. pressure



ture sufficiently low to cause the ammonia gas to liquefy and cool to about 70° by the time it has traversed the entire system of piping. The latter terminates in the receiver, *A*, Fig. 110, in which the liquefied ammonia is deposited ready to recommence the cycle of movements described.

The useful effect of the system is the refrigeration of the brine. The latter is circulated by the pump, *N*, through pipes distributed throughout spaces to be cooled, or has immersed in it vessels of water to be frozen, and thereby is heated through the range of temperature which measures the refrigerating power of the ammonia.

The work spent to accomplish this refrigeration is the steam-power exerted to operate the compression pumps, and to circulate the brine and condensing water.

SPECIAL ARRANGEMENTS FOR TEST OF PERFORMANCE.

Quantity of Brine Circulated.—This was measured by a No. 1 Gem meter, *O*, Fig. 108, through which the brine passed after being circulated in the cellars of the brewery. As a check on this measurement the brine pump was provided with a stroke counter, and its average length of piston stroke noted.

Heater for Warming Brine.—The amount of heat absorbed by the brine by circulation throughout the brewery cellars is not, in winter, sufficient to provide work of refrigeration equal to the full capacity of the plant. Consequently a feed-water heater, *P*, Fig. 108, was arranged to receive the brine on its return from the cellars, and live steam being supplied by the pipe *Q*, the brine could be heated to any desired extent within about 100 tons of ice-melting capacity.

Quantity Condensing Water Circulated through Tank, M.—This was measured with a two-inch Worthington meter, *R*, Fig. 110.

Quantity of Liquid Ammonia Circulated.—This was measured by a three-quarter-inch Worthington meter, *S*, Fig. 110.

Quantity of Water Circulated through Jackets of Compression Pumps.—This was determined by causing the overflow from the jackets to run into the vessel *T*, Fig. 109, and noting the head over an orifice at the bottom, out of which the water was allowed to escape.

Horse-power to Operate Plant.—The steam cylinder and both compressor pumps were indicated.

The indicator motion is shown at *V*, Figs. 108, 109, and 110.

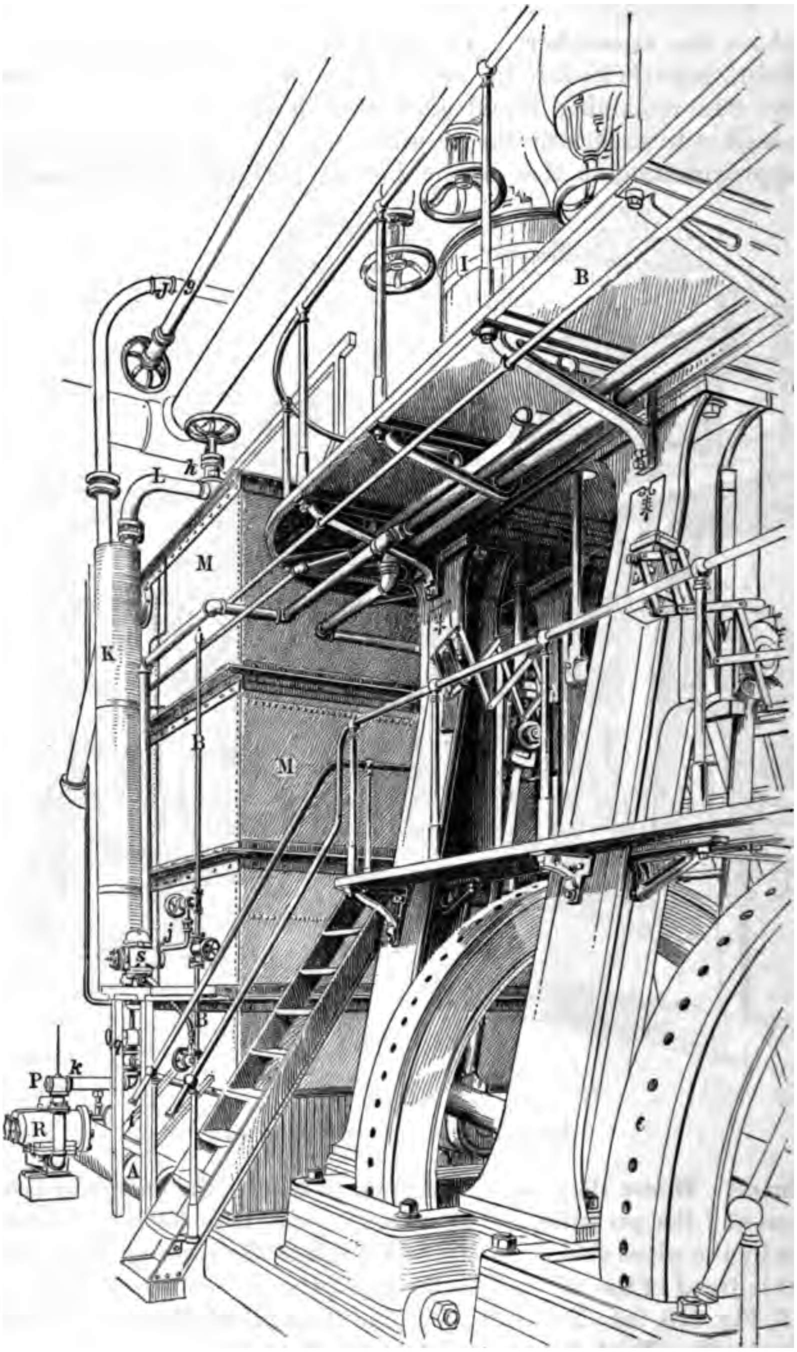


FIG. 110.

Back of Machine—Ammonia Meter at *S*. The pipe *g* should be shown as uniting with *J*.

ture sufficiently low to cause the ammonia gas to liquefy and cool to about 70° by the time it has traversed the entire system of piping. The latter terminates in the receiver, *A*, Fig. 110, in which the liquefied ammonia is deposited ready to recommence the cycle of movements described.

The useful effect of the system is the refrigeration of the brine. The latter is circulated by the pump, *N*, through pipes distributed throughout spaces to be cooled, or has immersed in it vessels of water to be frozen, and thereby is heated through the range of temperature which measures the refrigerating power of the ammonia.

The work spent to accomplish this refrigeration is the steam-power exerted to operate the compression pumps, and to circulate the brine and condensing water.

SPECIAL ARRANGEMENTS FOR TEST OF PERFORMANCE.

Quantity of Brine Circulated.—This was measured by a No. 1 Gem meter, *O*, Fig. 108, through which the brine passed after being circulated in the cellars of the brewery. As a check on this measurement the brine pump was provided with a stroke counter, and its average length of piston stroke noted.

Heater for Warming Brine.—The amount of heat absorbed by the brine by circulation throughout the brewery cellars is not, in winter, sufficient to provide work of refrigeration equal to the full capacity of the plant. Consequently a feed-water heater, *P*, Fig. 108, was arranged to receive the brine on its return from the cellars, and live steam being supplied by the pipe *Q*, the brine could be heated to any desired extent within about 100 tons of ice-melting capacity.

Quantity Condensing Water Circulated through Tank, M.—This was measured with a two-inch Worthington meter, *R*, Fig. 110.

Quantity of Liquid Ammonia Circulated.—This was measured by a three-quarter-inch Worthington meter, *S*, Fig. 110.

Quantity of Water Circulated through Jackets of Compression Pumps.—This was determined by causing the overflow from the jackets to run into the vessel *T*, Fig. 109, and noting the head over an orifice at the bottom, out of which the water was allowed to escape.

Horse-power to Operate Plant.—The steam cylinder and both compressor pumps were indicated.

The indicator motion is shown at *V*, Figs. 108, 109, and 110.

ter-inch size. Its valves were specially made of wrought and a few extra bolts were inserted around the circumferen the top and bottom joints; otherwise the meter did not from the stock types. The tightness of the joints was nurs that when submerged in water and subjected to 250 lbs. o pressure, no air bubbles were discernible. Aside from th newal of the packing about the arbor, actuating the clock- the meter gave no trouble during about a month of use.

To calibrate the ammonia meter, the instrument was adj so that when metering water, under the same pressures as vailed with the ammonia, the constant of the dial was fc The least rate at which the ammonia circulated was such cause the displacing plungers of the meter to knock agains ends of their cylinders at each stroke. It was therefore assı that the meter registered cubic feet truly, and by knowin density of the liquid ammonia the indications of the instru were reducible to pounds.

The calibration of the meter resulted as follows:

No. of Test.	Pressure on Meter. Lbs. per sq. in.	Rate of Flow. Cu. ft. per min	Weight of wa per cu. ft. regist Lbs.
1	160	0.8	59.6
2	160	0.8	58.5
3	160	0.8	58.6
4	150	0.8	59.0
5	150	0.8	58.3
6	160	0.6	58.9
7	155	0.3	59.1
8	160	0.3	58.2
Average			58.8

$$\text{Meter constant } \frac{58.8}{62.4} = 0.92.$$

Taking the density of liquid ammonia as 0.62, the weig ammonia corresponding to a cubic foot registered by the n is 35.6 lbs.

PRINCIPAL DIMENSIONS OF PLANT.

ENGINE.

Diameter ammonia cylinder single acting .... 12 in
Stroke " " " " 30
Diameter steam " double " 18
Stroke " " " " 36
Diameter piston rod, steam cylinder, double acting .... $2\frac{1}{8}$

during a test was obtained. Then the elbow N was suddenly swung over so as to deliver into the hogshead, and exactly 80 or 100 cubic feet, as indicated by the meter, allowed to empty into the tank when the elbow was swung off over the brine tank. This amount of brine was then weighed in the tank O in portions of about 800 lbs.

The meter was of the Gem type, 3-inch size, and its calibration resulted as follows:

Date.	Rate of Flow per Minute.	Equivalent of each Cubic Foot shown by Meter.	Date.	Rate of Flow per Minute.	Equivalent of each Cubic Foot shown by Meter.
	Cubic Feet.	Lbs.		Cubic Feet.	Lbs.
March 11.	30	74.6	April 8.	15	72.4
March 14.	30	75.8	April 27.	16	75.6
April 8.	30	75.6	April 27.	16	75.8
April 27.	32	76.0	April 27.	12	72.6
April 27.	31	75.8	April 27.	12	72.0
April 27.	31	76.3	April 27.	13	75.2
			April 27.	10	72.6
Av. for 30-foot rate of flow..		75.68	April 27.	10	75.4
			April 27.	10	72.9
			Av. for less than 30-foot rate..		73.8

For the tests which circulated brine at about 30 cubic feet per minute, each cubic foot registered by the meter was considered to weigh 75.68 lbs., and for the remaining tests 73.8 lbs.

To Calibrate the Water Meter R.—A swing outlet with a valve was connected at p, Fig. 106. The valve q leading to the tank M was then closed and p opened until the working rate of the water was attained, the water flowing on to the engine-house floor. The outlet was then swung so as to empty into the tank o, and 10 cubic feet, as indicated by the meter, were weighed in the tank.

Two meters were used; a 2-inch for trial No. 1, and a 3-inch for higher rates of flow. The calibration resulted as follows:

1st test.	2-inch meter	65.9	lbs. per cubic foot registered.
2d "	2-inch meter	66.5	" " "
3d "	2-inch meter	66.8	" " "
1st "	3-inch meter	63.4	" " "
2d "	3-inch meter	63.2	" " "

The average of each set was used in reducing results.

The Ammonia Meter was of the Worthington type, three-quar-

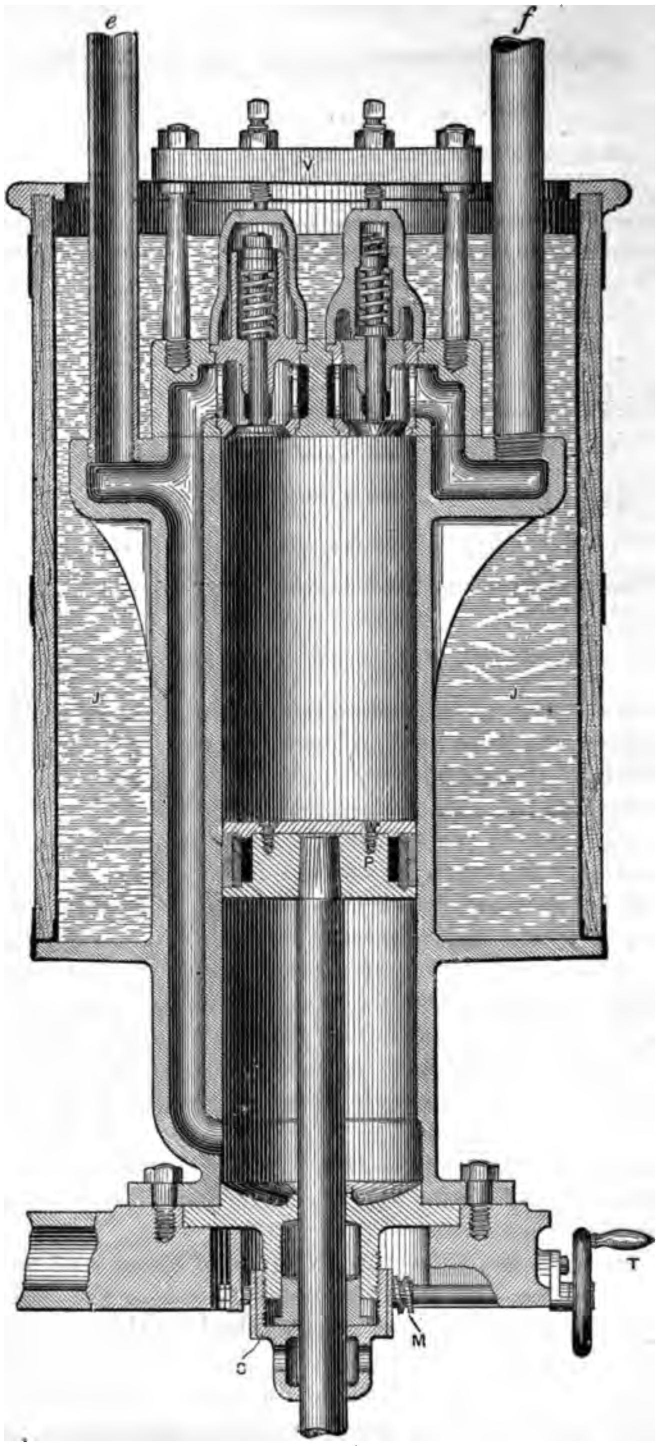


Fig. 111.

Section of Ammonia Cylinders.

On April 30th a second clearance measurement was made, and the pistons found to be 0.033 inch from the cylinder heads.

The indicator was attached to a one-fourth inch pipe, eight inches long, set in the cylinder head.

The waste space was made up as follows :

Four slats in screw heads in follower	0.117	cu. in.
Two lifting holes in follower	0.417	"
Length of $\frac{1}{32}$ inch between piston and head	3.534	"
Counterbore about single inlet valve	2.250	"
Counterbore about two outlet valves	2.788	"
Indicator connection	0.800	"
Half displacement of indicator piston	0.500	"
Total waste space each cylinder	10.406	"
Displacement of compressor piston	3393.0	"

Waste space in per cent. of displacement, $\frac{3}{10}$%.

The outlet valves were 2$\frac{3}{16}$ inches, and the inlet valve was 3$\frac{1}{4}$ inches diameter, at the base.

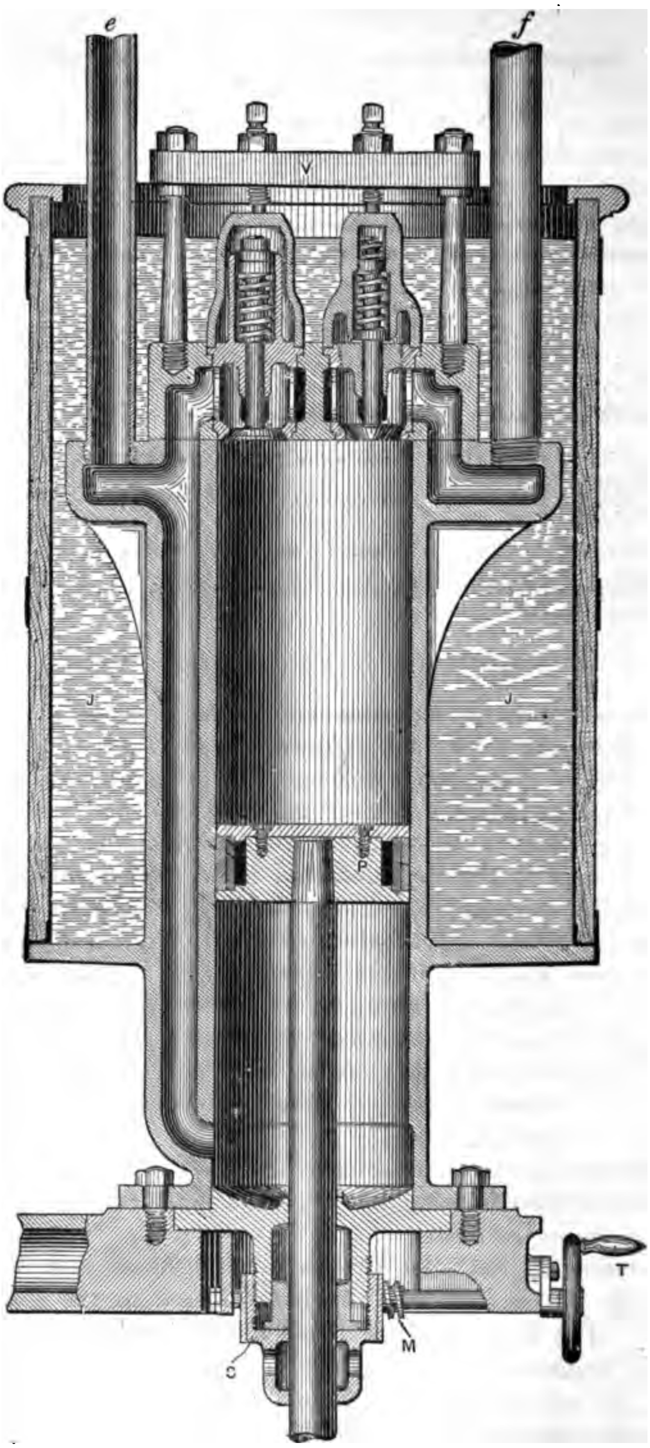


Fig. 111.

Section of Ammonia Cylinders.

On April 30th a sécond clearance measurement was made, and the pistons found to be 0.033 inch from the cylinder heads.

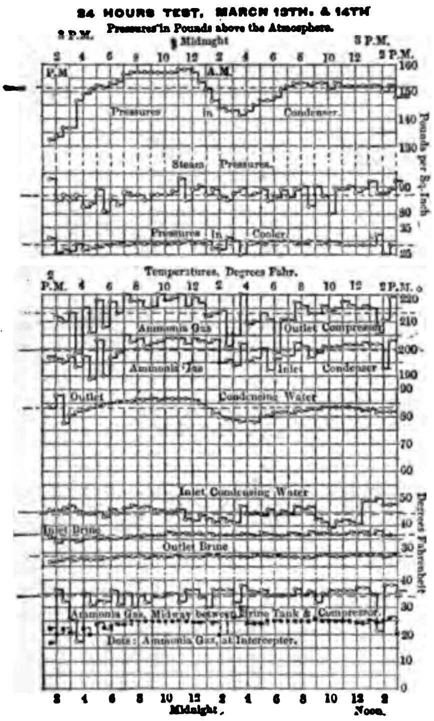
The indicator was attached to a one-fourth inch pipe, eight inches long, set in the cylinder head.

The waste space was made up as follows :

Four slats in screw heads in follower	0.117	cu. in.
Two lifting holes in follower	0.417	"
Length of $\frac{1}{32}$ inch between piston and head	3.534	"
Counterbore about single inlet valve	2.250	"
Counterbore about two outlet valves	2.788	"
Indicator connection	0.800	"
Half displacement of indicator piston	0.500	"
Total waste space each cylinder	10.406	"
Displacement of compressor piston	3393.0	"

Waste space in per cent. of displacement, $\frac{3}{10}$%.

The outlet valves were $2\frac{3}{8}$ inches, and the inlet valve was $3\frac{1}{2}$ inches diameter, at the base.



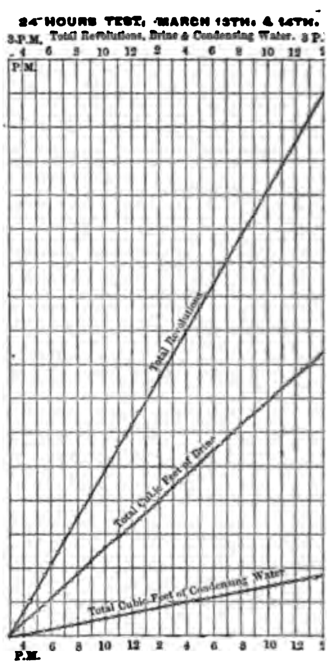


FIG. 112.—Trial No. 1.

23	*Pounds of water circulated per minute through Condenser*	449.0	268.0	948.0	816.0	669.0	709.0	867.0	601.0	Jacket Water.
24	*Pounds of water circulated per minute through Jackets*	25.0	*26.5	*17.0	44.0	35.0	37.0	40.0	14.0	
25	Average temperature of water entering Jackets	44.65°	55.4°	*56.0°	56.7°	51.0°	46.0°	46.9°	54.3°	
26	Average temperature of water leaving Jackets	69.0°	79.4°	80.0°	72.9°	70.0°	60.3°	63.3°	63.45°	
27	Range of temperature in Jackets	24.0°	*24.0°	*24.0°	16.2°	19.0°	14.3°	16.4°	9.1°	
28	Cubic feet of Liquid Ammonia circulated per hour	......	18.6	25.00	24.96	26.23	25.16	28.26	48.01	Quantity of Ammonia.
29	Weight of Ammonia circulated per minute, lbs.	*28.17	10.94	15.24	14.68	15.40	14.81	16.67	28.32	
30	Probable temp. of Liquid Ammonia, entrance to Brine tank	*71.3°	*58.0°	*69.°	*68.°	*60.5°	*52.5°	*63.7°	76.7°	
31	Lbs. of Amm. per min. due to displacement of Comp. Pistons.	......	14.82	......	......	......	19.02	22.10	34.51	Displacement of Compressors.
32	Difference unaccounted for by Compressor displacement	......	3.48	......	......	......	4.18	5.43	5.19	
33	Degrees Fahr. of superheating by walls of Cylinder	......	130.°	......	......	......	108.°	160.°	66.°	
34	Temp. of Ammonia corresponding to average back pressure	+14.0	−22.0	−13.0	−8.0	−7.0	−11.0	−5.0	14.0	
35	Average temperature of Gas after leaving Brine tanks	34.2	15.5	11.8	14.7	10.1	4.7	3.0	29.2	Gas Temperatures.
36	Temperature of Gas entering Compressor	*39.°	25.°	25.°	25.°	20.°	15.°	10.13°	34.°	
37	Average temperature of Gas leaving Compressor	213.0°	259.0°	205.0°	263.0°	248.0°	222.0°	239.0	221.0°	
38	Theoretical temperature $t = t_0 \left(\frac{p_1}{p_0}\right)^{0.24}$	*220.°	370.°	331.°	304.°	275.°	258.°	260.°	237.°	
39	Average temperature of Gas entering Condenser	200.0°	209.0°	221.0°	218.0°	205.0°	192.0°	209.0°	168.0°	
40	Temperature due to Condensing pressure	84.5°	78.5°	86.0°	84.0°	74.5°	66.0°	82.5°	88.0°	
41	Temperature of Engine room	68.°	63.°	64.°	64.°	63.°	63.°	67.°	68.°	
42	Specific Gravity of Brine	1.163	1.174	1.174	1.174	1.174	1.174	1.174	1.174	Specific Heat of Brine.
43	Specific Heat of Brine	0.82	0.78	0.78	0.78	0.78	0.78	0.78	0.783	
44	Barometer, inches	30.07	......	......	29.59	30.10	30.28	29.87	30.01	
45	Strokes of Brine pump per minute	......	27.23	19.9	21.66	22.65	11.06	11.04	23.55	Brine Pump.
46	Heat given to Ammonia by Brine, B. T. U. per minute	11776	5308	7460	7186	7612	6925	8824	14647	Account of Heat Received and Rejected.
47	Heat given to Ammonia by Compressor, B. T. U. per minute	2786	1942	2325	2320	2205	2074	2518	3020	
47a	" " " Atmosphere, B. T. U. " "	140	110	152	147	155	148	167	141	
48	Total heat received by Ammonia, B. T. U. " "	17702	7360	9937	9653	9012	9147	11409	17708	
49	Heat taken from Ammonia by Condenser, B. T.U. " "	17242	6703	7874	9056	12330	9997	9910	17359	
50	Heat taken from Ammonia by Jackets, B. T. U. " "	608	616	425	712	665	529	656	406	
50a	" " " " Atmosphere, B.T.U. " "	182	275	330	338	328	222	250	252	
51	Total Heat rejected by Ammonia, B. T. U. " "	18032	7594	8629	10106	13323	10748	10816	18017	
52	Difference of Heat received and rejected, B. T. U. " "	330	234	1308	453	3411	1601	407	309	
53	Difference in per cent. of Heat received	2%	3%	13%	5%	34%	17%	3.5%	2%	
54	Per cent. of work of Compression removed by Jackets	22%	31%	18%	31%	30%	25%	26%	13%	
55	Boiler pressure above atmosphere, lbs.	87.0	83.0	84.0	83.0	85.	86.	84.	93.	
56	Average revolutions per minute	58.09	58.7	58.38	57.7	57.83	59.2	57.88	58.89	Revolutions per Minute.
57	Mean effective Pressure Steam Cylinder, lbs. per square inch.	32.5	22.12	27.4	27.17	25.48	22.866	27.83	32.97	
58	Mean effective Pressure Ammonia Cylinder, lbs. per sq. inch.	65.9	45.75	54.77	53.3	52.45	47.51	59.86	70.54	
59	Average Horse-power Steam Cylinder	85.00	59.28	73.0	71.7	67.13	61.79	73.6	88.63	Horse-powers.
60	Average Horse-power Ammonia Cylinder	65.7	46.02	54.8	54.7	51.97	48.19	59.37	71.20	
61	Average Horse-power in Friction	19.3	13.26	18.2	17.0	15.16	13.6	14.23	17.43	
62	Friction in per centum of steam horse-power	23.0	22.0	25.0	24.0	23.0	22.0	20.0	19.67	Friction.
63	Total cooling water gallons per minute per ton per 24 hours	0.75	1.45	0.81	1.185	2.189	2.527	0.797	0.990	
64	Latent Heat per lb. of ammonia for condensing pressure, B.T.U	537.0	522.0	......	569.0	......	529.0	596.0	574.0	Latent Heat.
65	Latent Heat per lb. of ammonia for back pressure, B. T. U	577.0	544.0	......	535.0	......	592.0	508.0	538.0	
66	Jacket water gallons per minute per ton in 24 hours	0.04	*0.12	0.02	0.145	0.109	0.127	0.107	0.022	Water Consumption.
67	Condensing water gallons per minute per ton	0.71	1.33	0.79	1.04	2.08	2.40	0.69	0.968	
68	Tons Ice melting capacity per 24 hours	71.8	*25.9	37.79	36.43	38.59	35.11	44.64	74.56	Commercial Efficiency.
69	Lbs. of Ice refrig. effect per lb. of Coal at 3 lbs. per horse-power	24.1	*12.6	14.13	14.1	15.96	16.03	17.27	23.37	
70	Cost of Coal per 1 ton of Ice refrigerating effect at $4.00 per ton.	$0.166	$0.317	$0.283	$0.283	......	$0.245	$0.231	$0.170	
71	Cost of Water per 1 ton of Ice refrig. eff. at $1 per 1,000 cu. ft.	$0.128	$0.212	$0.136	$0.200	......	$0.360	$0.136	$0.169	
72	Total cost of one ton of Ice, refrigerating effect	$0.294	$0.529	$0.419	$0.483	......	$0.605	$0.467	$0.339	

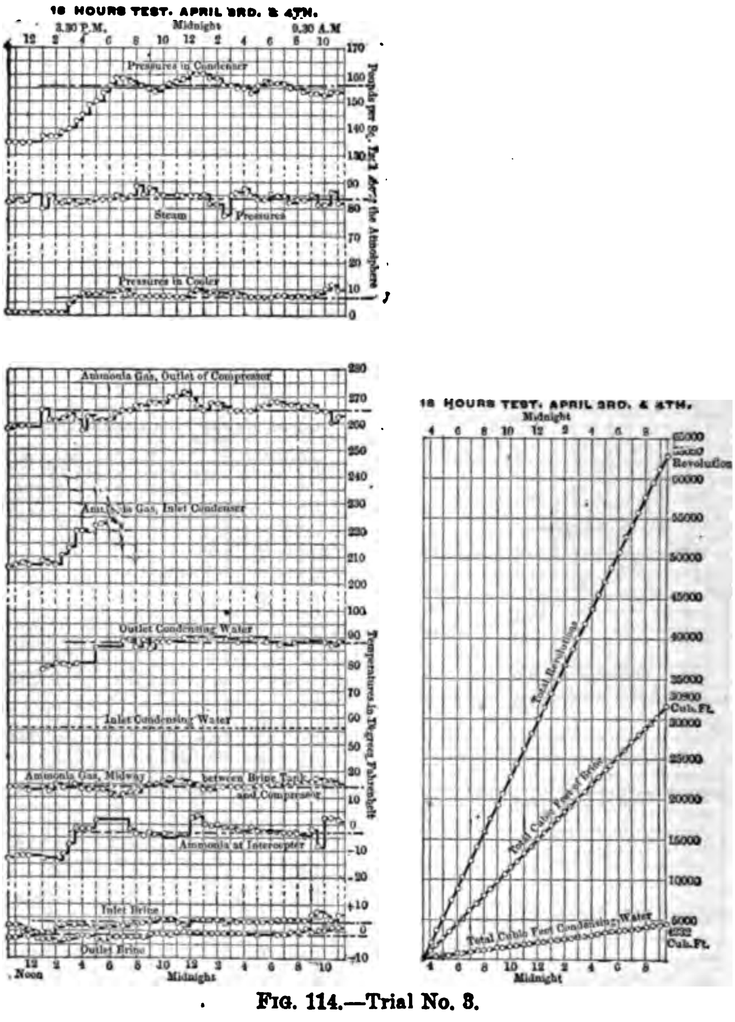
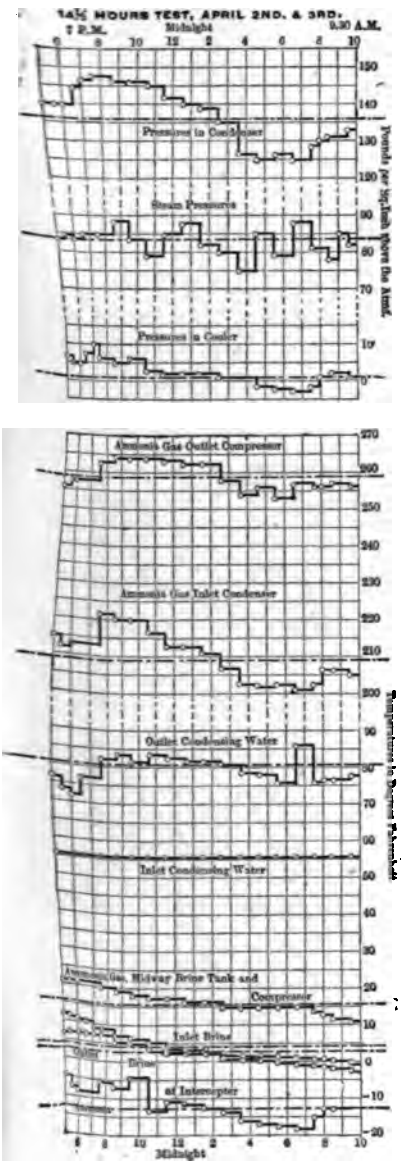


FIG. 114.—Trial No. 3.



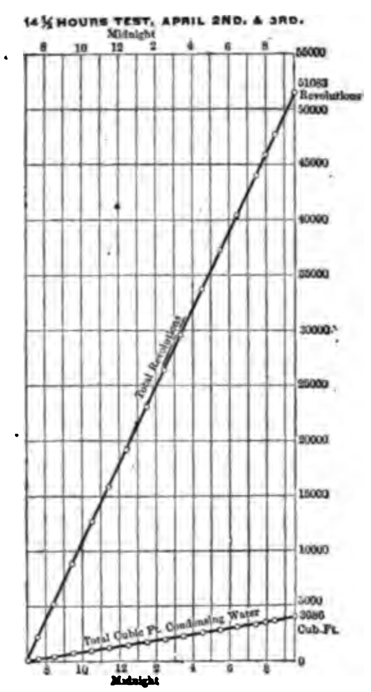


FIG. 118.—Trial No. 2.

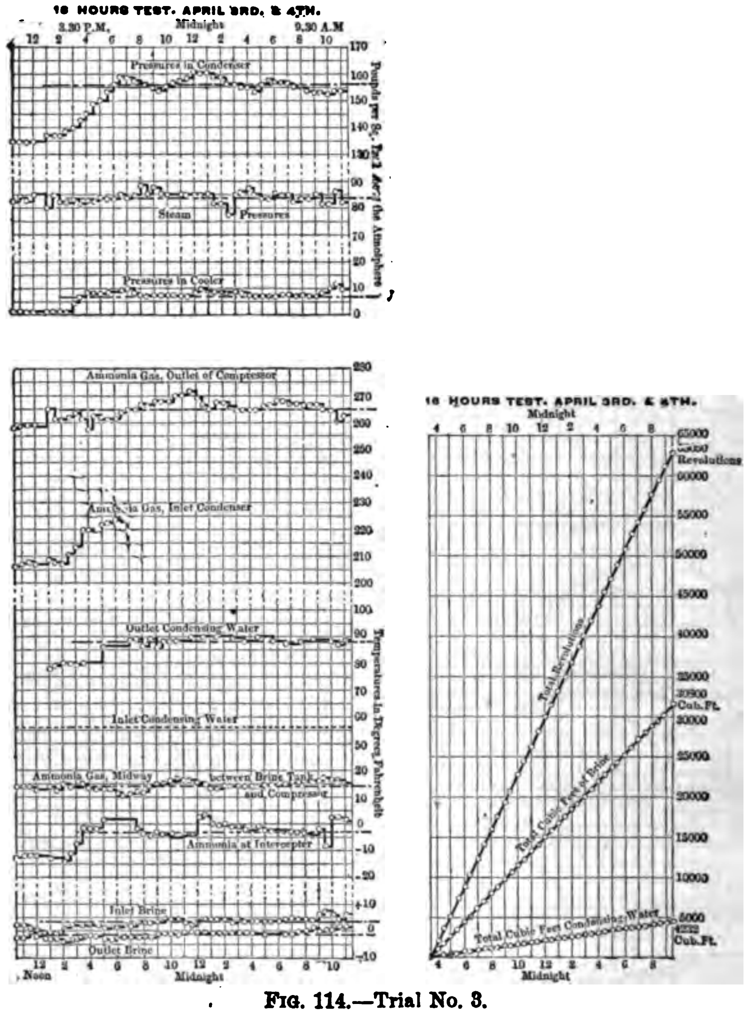


FIG. 114.—Trial No. 3.

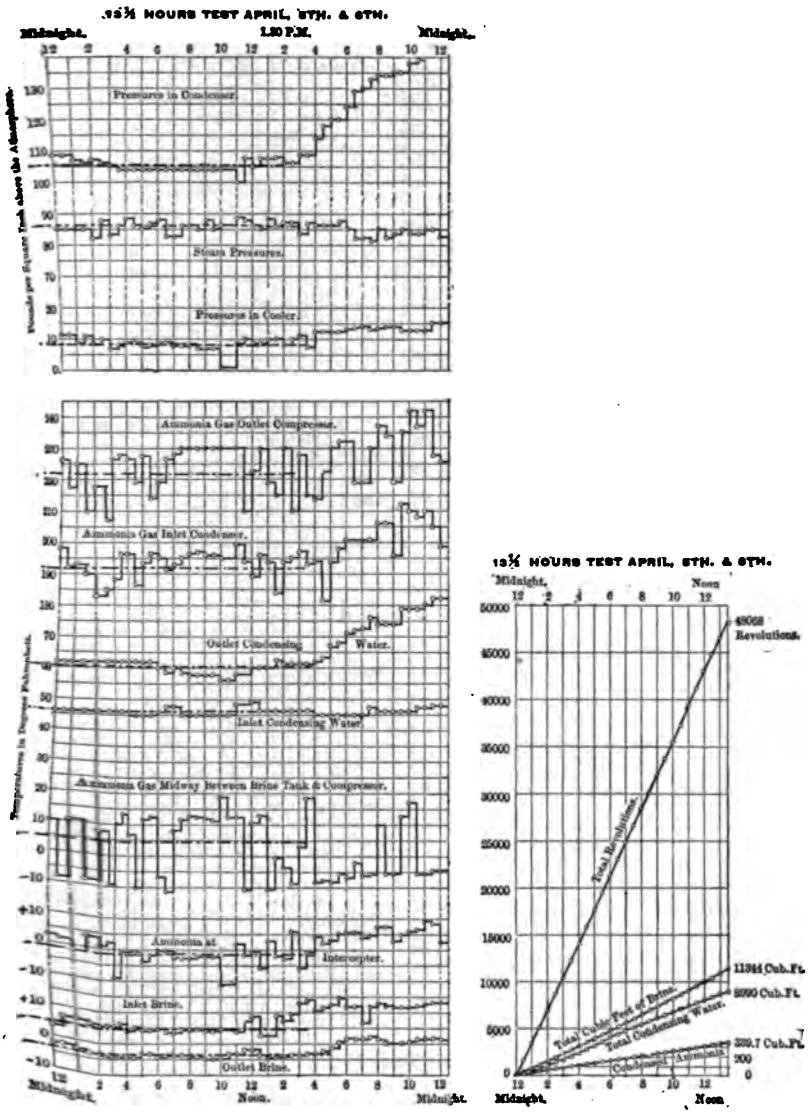


FIG. 117.—Trial No. 6.

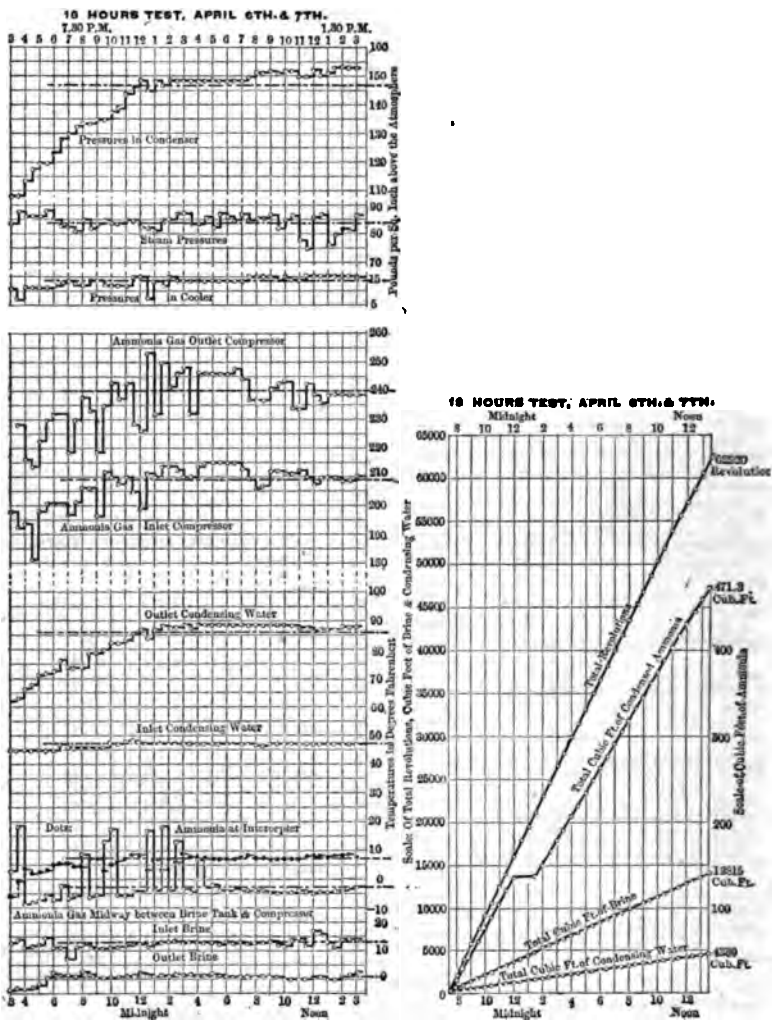


FIG. 118.—Trial No. 7.

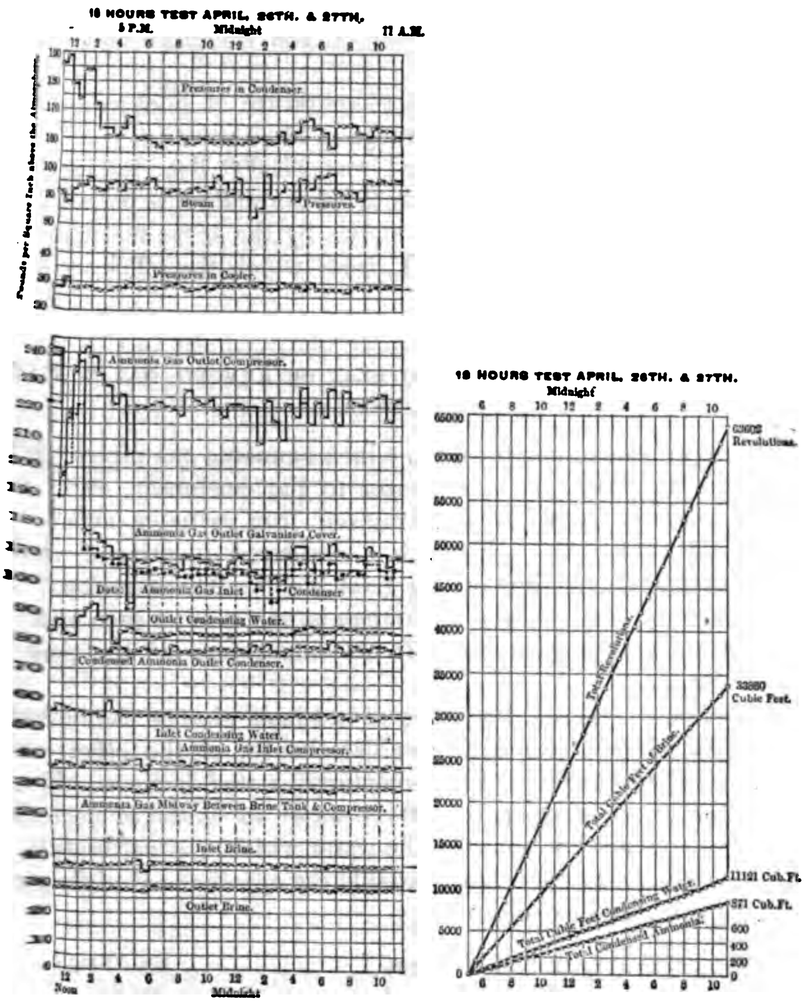


FIG. 119.—Trial No. 8.

ESTIMATE OF REFRIGERATIVE EFFECT, TRIAL NO. 2.

During this trial the brine was being reduced to zero for the succeeding trials. The work of the machine was therefore not obtainable from the brine meter alone, as the reduction of the temperature of the brine was a large fraction of this work. The refrigerating effect given in Table I. has therefore been computed as follows:

During 14½ hours the mean temperature of inlet and outlet brine fell from 9.25 to − 2°, a range of 11.25.

The weight of brine involved was 225,000 lbs., making the work of lowering the temperature 2,249 thermal units per minute. This, added to the 3,059 units shown by the brine meter, gives the figures in line 47, Table I., from which the efficiency and capacity are computed.

SPECIAL EXPERIMENT ON SPECIFIC HEAT OF AMMONIA GAS AT HIGH PRESSURE.

Thirteen feet of length of the 4-inch ammonia gas pipe g', Fig. 110, leading from the compressor to the condenser, were enclosed, water tight, by a galvanized iron cylinder about 12 inches diameter, through which a measured quantity of water was circulated steadily during trial No. 8. Thermometers inserted in mercury wells in the ammonia gas pipe at each end of the water jacket gave the fall of temperature of the gas to half degrees.

The water entered the jacket at a 1-inch pipe at the bottom of the condenser end, and issued at the top of the other end, through a similar pipe. Thermometers inserted in these orifices gave the rise of temperature of the water to tenths of degrees. A meter was used to measure the water which was calibrated in place. From the data thus available the specific heat of the gas can be computed as follows:

Mean temperature Gas inlet to jacket	221.0°	Fahr.
" Gas outlet "	168.0°	"
Mean range of Gas	53.0°	"
Mean temperature Water inlet to jacket	54.3°	"
" Water outlet "	63.5°	"
Range	9.2°	"
Water circulated per minute	91.23	lbs.
Ammonia Gas circulated per minute	28.82	lbs.
Mean temperature of atmosphere	68.0°	

The radiation effects being neglectable we may write:

$$91.23 \times 9.2 = \text{Specific heat of gas} \times 28.32 \times 53.$$
$$\text{Specific heat of gas} = 0.557.$$

The controlling error of the determination is, of course, that involved in the two meters used for the water and ammonia respectively. The error of the constant for the ammonia meter discussed elsewhere is possibly 5%, and may be either plus or minus.

The meter for the water measurement in the specific heat determination was calibrated but once in place; but from subsequent trials, under conditions similar to those obtaining during the specific heat test, it is certain that the constant used for the latter can err only in the direction tending to give to a specific heat value.

Assuming the whole 5% error of the water meter to apply towards reducing the above figure, we should have the specific heat = 0.532, which is considered the most probable value.

WEIGHT OF AMMONIA CIRCULATED, AND TEMPERATURE OF LIQUID AMMONIA WHEN METERED.

In the test of April 26th and 27th, the temperature of liquid ammonia was observed at the point of exit from the condenser and at the entrance to the ammonia meter.

The temperature at each point was 76.7°, no sensible change occurring during the passage through the reservoir A, Fig. 110. During the preceding tests no observation of these temperatures was made. I have therefore resorted to the following method to calculate approximately the temperature of the liquid at the meter. Let θ_2 be the arithmetical mean of the inlet and outlet condensing water temperatures.

Let θ_1 be the inlet condensing water temperature.

Assume that $\frac{\theta_2 - \theta_1}{2}$ is the mean temperature of the condensing water to which the ammonia gives up its heat during the cooling of the liquid from the temperature due to the condenser pressure to the temperature at which it issues from the condenser into the reservoir A. Assume, also, that the latter temperature is the same as that at which the liquid passes the ammonia meter. Let T_1^1 be the temperature due the condensing pressure of the liquid ammonia.

Let C be a constant equal to the temperature lost by the ammonia per degree difference of mean temperatures of ammonia and water and per unit weight of ammonia circulated per unit of time. Let W be the weight of ammonia circulated per unit of time. Let ΔX be the difference between the temperature due pressure of the ammonia and the temperature at which it passes the meter.

Then we may write $\left[T_1^1 - \left(\frac{\theta_2 - \theta_1}{2}\right)\right]\frac{C}{W} = \Delta X.$

From the data of the test of April 26th, we have $T_1^1 = 88°$ $\frac{\theta_2 - \theta_1}{2} = 61°$, $W = 27.5$ lbs. per minute at assumed density of 0.62 $\Delta X = 11.3°$. Using these values to determine C, we have $\Delta X = 11.5 \frac{\theta_2 - \theta_1}{W}$.

By this formula is computed the last column of the following table:

TABLE XX.

SHOWING TEMPERATURES OF AMMONIA AND CONDENSING WATER FOR THE PRINCIPAL TESTS.

No. of Trial.	P_1 Condensing Pressure above atmosphere, lbs., per sq. inch.	T_1^1 Temp. Fahr. due Pressure P_1	W Weight of Ammonia per minute lbs.	θ Temp. of Cond'g. water at outlet.	θ_1 Temp. of Cond'g. water at inlet.	θ_2	$\frac{\theta_2-\theta_1}{2}$	Temp. of liquid Ammonia at Exit from Condenser or at meter.
8.....	161	88°	28.32	82.9°	54.0	68	61.0°	76.7° Observed.
4.....	152	85°	14.68	85.4°	56.7	71	64.0°	68.0° Calculated.
6.....	105	66°	14.84	60.2°	46.0	53	49.0°	52.5° "
	142	81°	16.67	85.5°	46.9	66	57.0°	63.7° "

By the following table of densities of liquid ammonia it is evident that if the average density of the liquid ammonia for all the tests be taken at 0.62 or 35.6 lbs. per cubic foot, this figure will be sufficiently accurate for any practical deduction.

The radiation effects being neglectable we may write:

$$91.23 \times 9.2 = \text{Specific heat of gas} \times 28.32 \times 53.$$
$$\text{Specific heat of gas} = 0.557.$$

The controlling error of the determination is, of course, that involved in the two meters used for the water and ammonia respectively. The error of the constant for the ammonia meter discussed elsewhere is possibly 5%, and may be either plus or minus.

The meter for the water measurement in the specific heat determination was calibrated but once in place; but from subsequent trials, under conditions similar to those obtaining during the specific heat test, it is certain that the constant used for the latter can err only in the direction tending to give to a specific heat value.

Assuming the whole 5% error of the water meter to apply towards reducing the above figure, we should have the specific heat = 0.532, which is considered the most probable value.

WEIGHT OF AMMONIA CIRCULATED, AND TEMPERATURE OF LIQUID AMMONIA WHEN METERED.

In the test of April 26th and 27th, the temperature of liquid ammonia was observed at the point of exit from the condenser and at the entrance to the ammonia meter.

The temperature at each point was 76.7°, no sensible change occurring during the passage through the reservoir *A*, Fig. 110. During the preceding tests no observation of these temperatures was made. I have therefore resorted to the following method to calculate approximately the temperature of the liquid at the meter. Let θ_2 be the arithmetical mean of the inlet and outlet condensing water temperatures.

Let θ_1 be the inlet condensing water temperature.

Assume that $\frac{\theta_2 - \theta_1}{2}$ is the mean temperature of the condensing water to which the ammonia gives up its heat during the cooling of the liquid from the temperature due to the condenser pressure to the temperature at which it issues from the condenser into the reservoir *A*. Assume, also, that the latter temperature is the same as that at which the liquid passes the ammonia meter. Let T_1^1 be the temperature due the condensing pressure of the liquid ammonia.

temperature of the contents of the brine tank under these conditions was determined by means of a special thermometer 11 feet long, made by Green. Its stem correction for full immersion in the brine tank was 0.07°. The following table shows the results of exploring the brine tank with this instrument:

MEAN TEMPERATURE OF BRINE TANK.

Location of Thermometer.	Beginning of 24 Hours Interval.		End of 24 Hours Interval.			Beginning.	End.
	Bottom.	Top.	Bottom.	Top.			
Middle	29.6	34.5	28.9	33.5	Inlet	36.25	33.75
Northwest Corner	30.85	34.25	29.35	33.7	Outlet	28.30	28.25
Northeast Corner	27.60	35.00	28.20	33.2			
Southeast Corner	27.85	34.25	27.50	33.2		64.55	64.00
Southwest Corner	27.85	34.25	27.1	33.0			
South end, Middle	27.60	33.25	28.0	33.0	Inlet	37.0	35.9
West end, Middle	30.35	33.75	29.4	33.6	Outlet	28.5	28.0
Average	200.90 28.70	239.25 34.18	198.45 26.92	233.2 33.3		65.5	63.9
Sum top and bottom	62.88		60.22				
Average top and bottom	31.44		30.11				

The weight of brine involved was 225,000 lbs.

If the temperature of this mass altered 1.23° in 24 hours and no account is taken of this fact in reducing results, the heat units which would thus escape being accounted for would amount to 160 per minute. This makes a possible error in results of from 2 to 5% according to the capacity and length of the trials.

4. The temperature ranges contain the error of the thermometer readings, which may be as great as ½° in any single reading; but the error of the average of the many readings is but a fraction of this amount, and cannot affect the results to more than 1% in any case.

5. The error of the specific heat value has been discussed under a special head and shown to be about 1.5%.

6. The mean effective pressure as determined from the indicator cards is about 3%. For practical purposes the results of trials, for which the heat received is within about 5% of the heat rejected, are sufficiently accurate; and on this basis trials 1, 4, and 8 are excellent, and constitute a measure of the probable combined errors due to the above five sources. Trials 1 and 8 being largely duplicates of each other have thereby additional

Temperature Fahrenheit.	Density. Distilled Water being Unity.	Authority.
5°	0.731	Faraday.
14°	0.6492	D'Andreff.
23°	0.6429	D'Andreff.
32°	0.6364	D'Andreff.
41°	0.6298	D'Andreff.
50°	0.6230	D'Andreff.
59°	0.6160	D'Andreff.
68°	0.6089	D'Andreff.
86°	0.6018	D'Andreff.

LIMIT OF ACCURACY OF RESULTS.

1. The amount of heat radiated from the brine tank to the atmosphere was not determined. Rough observations were made, however, while the brine was $-4°$ Fahr., and the machine stopped during an interval of about three hours, from which it was concluded that the loss of heat did not exceed a small fraction of 1% of the heat involved in the lowest capacity trial.

2. The constant for each of the meters must be considered liable to an error of 5%. The brine meter determinations are regarded as the best, first, because the index figures moved continuously, and thereby permitted the number of cubic feet used for calibrating to be more accurately determined; second, because the brine pump was under the regulation of a counter, and available to adjustment by the observers. The water supply depended on pumps at considerable distance, and was complicated with several circumstances more or less beyond steady regulation. The large discrepancies shown in the disagreement of the heat received and rejected in trials 2, 3, 5, and 6, lines 47–54, Table I., are due to these irregularities in the water supply.

3. The difference of temperature of the contents of the brine tank before and after the experiment may amount to a considerable fraction of the heat involved if the experiment is short or the variations of temperature are not under thorough control. The degree of steadiness of the temperature is shown by the charts, Figs. 112 to 119.

While not as perfect as is possible when the work of refrigeration is wholly due to cooling warehouse space, the degree of steadiness for the conditions of load by means of a heater are quite consistent with all necessary accuracy. The variation of

LATENT HEAT OF AMMONIA AT SUCTION PRESSURES.

No. of Trial.	Suction Pressure. Lbs. per sq. in. above atmosphere.	W Lbs.	Deg. Fahr. t^1	Deg. Fahr. t_2	British Thermal Units. q^1_1	q_2	Latent Heat. Experimental.	Latent Heat. Theoretical.	Heat abstracted from Brine. Line 46. Table I.
1	28.0	28.17	+24	+14.0	71.5	+14.09	577.0	547.0	14776
2	1.0	10.94	+19	-22.0	58.15	-21.67	544.0	568.5	5308
4	8.2	14.68	+4.0	-8.0	68.22	-7.36	569.0	559.5	7186
6	7.6	14.84	-3.0	-11.0	52.68	-10.93	529.0	562.0	6925
7	13.0	16.67	-1.0	-5.0	63.93	-4.63	596.0	559.0	8824
8	27.5	28.32	+22.0	+14.0	77.03	+14.09	574.0	547.0	14647

LATENT HEAT OF AMMONIA AT CONDENSING PRESSURES.

No. of Trial.	Condensing Pressure. Lbs. per sq. in. above atmosphere.	W Lbs.	Deg. Fahr. t_3 Line 39. Table I.	Deg. Fahr. t_1 Line 40. Table I.	British Thermal Units. q_1	q^1_1	Latent Heat. Experimental.	Latent Heat. Theoretical.	Heat given to Condensing Water. Line 49. Table I.
1	151	28.17	200	84.5	84.80	71.5	537	502	17243
2	135	20.94	209	78.5	78.71	58.15	522	506.0	6703
4	152	14.68	218	84.5	84.80	68.22	535.0	502.0	9056
6	105	14.84	192	66.0	66.40	52.68	592.0	514.5	9997
7	147	14.84	209	82.5	82.80	63.93	508	503.5	9910
8	161	28.82	208	88.0	88.24	77.03	541	499.5	17859

The theoretical figures are from Wood's Thermodynamics.

CONCLUSIONS REGARDING LATENT HEAT.

The accuracy of the theoretical values for latent heat is almost a certainty within a few heat units. The constants upon which the theoretical formulæ depend are due to sufficiently high physical authority to be assumed very nearly accurate, and the only purely thermodynamic expression involved is of axiomatic foundation and very direct in its application to the phenomena of evaporation. Furthermore, a few values of the latent heat of ammonia are extant as made by Regnault, and by the paper of Prof. Jacobus on the subject it is shown that the theoretical calculations practically agree with these. Such refined physical methods of determination naturally appeal less strongly to the ideas of the practical engineer, than do measurements on a large scale made with the actual machinery making use of the laws of latent heat in securing its useful effect.

claims to accuracy, but even these two trials are liable to an error of upwards of 5%.

This fact is shown in the discussion under "Latent Heat."

LATENT HEAT OF ANHYDROUS AMMONIA.

The 1st, 2d, 4th, 6th, 7th and 8th trials are best available to determine the latent heat at the suction and condensing pressures.

For the suction pressures we have:

$$\left.\begin{array}{l}\text{Heat extracted from brine}\\ \text{Thermal units per minute}\end{array}\right\} = W\,[0.508\,(t_2 - t_2) + r_2 - (q^1_1 - q_2)]\ (A)$$

In which

W = the ammonia circulated per minute.

0.508 = specific heat of gas.

t^1_2 = temperature at which gas leaves brine.

t_2 = boiling point of ammonia due suction pressure.

r_2 = latent heat at suction pressure.

q^1_1 = thermal units in liquid ammonia at entrance to brine tank.

q_2 = thermal units in liquid ammonia at t_2. As t'_2 was not observed, its value is assumed equal to the mean of t_2, and the temperature of the gas about ten feet away from the point of exit from the brine tank (at the point Z, Fig. 108). See line 35, Table I.

For the condensing pressure we have:

$$\left.\begin{array}{l}\text{Heat given to condensing water}\\ \text{Thermal units per minute}\end{array}\right\} = W\,[r_1 + 0.532\,(t_3 - t_1) + c\,(q_1 - q^1_1)]$$

In which

r_1 = latent heat at condensing pressure.

0.532 = specific heat of gas before liquifaction in the condenser.

t_3 = temperature of gas at entrance to condenser.

t_1 = boiling point of ammonia at condensing pressure.

c = specific heat of liquid ammonia—taken as 1.0.

q^1_1 = thermal units as in formula (A).

W = weight of ammonia as in formula (A).

q_1 = thermal units in liquid ammonia at t.

The following results are obtained:

Columns 9 and 10 show how exactly the refrigerating capac ties are proportional to the weight of ammonia circulated.

The figures in column 7, also, are relatively in very satisfactoı agreement with theory. Thus, in trials 4 and 6, we have pra tically the same back pressure, and hence the same weight ammonia in circulation. But because of the lower condensin pressure in No. 6, the horse-power required for compression i theoretically only five-sixths as much as in No. 4. Also th proportion of the latent heat available in No. 6 is 6% more tha in No. 4. Hence the economy of No. 6 should be $\frac{6}{5} \times 1.06$ time that of No. 4, or 17.7 lbs., as against 17.0 by experiment.

Again, comparing No. 4 and No. 8, the horse-power increase 25%, the amount of ammonia circulated increases 92%, and 2 more of the latent heat is available. Hence the increase economy is $\frac{1.92 \times 1.02}{1.25} = 1.55$ times that of No. 4, which give 21.7 against 22.27 actual. Comparing No. 2 and No. 8, the latter horse-power is 0.54% greater, and its latent heat more avai able to the extent of 4%; but 2.6 times as much ammonia is ci culated. Hence the pounds of ice per pound of coal of No. should be that of No. 8 multiplied by $\frac{1.54}{2.6 \times 1.02} = 0.58$. Th gives 12.9 lbs. for No. 2, as against 13.2 by test. The small di crepancies are, of course, due to difference of friction, radiatio etc., which the theoretical horse-powers do not consider.

DISCUSSION OF EFFECT OF CONDENSING WATER.

The most reliable data regarding the quantity and influence the condensing water are shown in the following tables, in whic the latent heat and amounts of condensing water are take according to the conclusions under "Latent Heat."

Column 14 shows the total heat which must be absorbed fc each pound of ammonia; and in columns 11, 12 and 13 is show the manner in which this heat is distributed.

About 12½% is devoted to reducing the superheated gas to th state of a saturated vapor; at the condensing pressure about 85 is the latent heat of evaporation, and about 2½% is consumed i cooling the liquid ammonia after it has been condensed. Th actual quantities of condensing water must vary in proportio to the values in column 14, but the latter are so nearly pr portional to the values in column 19, that the condensin

Nevertheless, the best possible measurements with practical machinery must not be regarded as better than a means of crudely corroborating such a constant as the latent heat when the value of the latter is as definitely settled by the nicer methods of physics as is now the case. The only proper use of the experimental latent heat values is to consider them in error to the full extent of their difference from the theoretical values, and, by comparing with the discrepancies of heat balance, arrive at corrections which may be applied to the commercial efficiency values resulting from the tests, to reduce the error of the latter.

Thus, in the case of trial No. 4, we have the latent heat at the condensing pressure only, in disagreement with the correct figure. We, therefore, regard as correct the figures for capacity and pounds of ice per pound of coal, which depend mainly on the brine end or latent heat at suction pressure; while the water consumption at the condenser is to be regarded as about 6% too high.

In the case of trial No. 6, the brine end should be increased about 6%, and the water end decreased about 13%.

In trials No. 1 and No. 8 both the brine and water end should be decreased about 6%.

In trial No. 7 the brine end should decrease about 6% and the water end about 1%.

In trial No. 2 the brine end should be increased 5% and the water end decreased about 3%.

These corrections are fairly consistent with a nearly exact heat balance, a fact which corroborates their probability.

The most probable figures of performance are, therefore, as follows:

No. of Trial.	Ammonia Pressures. Lbs. above Atmosphere.		Brine Temps., Degrees Fahr.		Capacity tons refrigerating effect per 24 hours.	Efficiency lbs. of ice per lb. of coal at 3 lbs. coal per hour per H.P.	Water Consumption. Gallons of water per min. per ton of capacity.	Ratio of actual weights of Ammonia. Trial No. 8 being unity.	Ratio of capacities. Trial No. 8 being unity.
	Condensing	Suction.	Inlet.	Outlet.					
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
1	151	28	36.76	23.86	70.3	22.60	0.80	1.0	1.0
8	161	27.5	36.36	28.45	70.1	22.27	1.09	1.0	1.0
7	147	13.0	14.29	2.29	42.0	16.27	0.83	1.70	1.66
4	152	8.2	6.27	2.08	36.43	14.10	1.1	1.93	1.92
6	105	7.6	6.40	−2.22	37.20	17.00	2.00	1.91	1.88
2	135	15.7	4.62	3.22	27.2	13.20	1.25	2.59	2.57

CORRECTED CONDENSING WATER DATA.

	Pressure. Lbs. per sq. in. above Atmosphere.		Temperatures. Degrees Fahrenheit. Boiling Points.		Temperatures. Degrees Fahrenheit. Condensing Water.			Temp. Degrees Fahr. Ammonia.		British Thermal Units Abstracted from 1 lb. Ammonia.									Condensing Water.				
No. of Trial.	Condensing.	Suction.	Condensing pressure.	Suction pressure.	Inlet.	Outlet.	Range.	Entering condenser.	Exit from condenser and entrance to brine tank.	In reducing gas from temp. Col. 9 to Col. 4.	In absorbing latent heat.	In reducing liquid from temp. Col. 4 to Col. 10.	Total heat given up in condenser. Sum of Cols. 11, 12, and 13.	Latent heat at suction pressure.	Necessary to cool ammonia during expansion.	Refrigerative effect. Diff. Cols. 17 and 16.	Col. 18 in per cent. of Col. 14.	Lbs. ammonia circulated per minute.	Per min. lbs. Jacket not included.	Per lb. ammonia.	Ratio of ranges. Col. 8. Trial No. 8 being unity.	Ratio of gals. Col. 23. Trial No. 8 being unity.	Gals. per min. per ton of refrigerative effect per 24 hours.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	16	17	18	15	19	20	21	23	24	22
2	135	28.0	78.5	−22.0	55.40	80.40	25.0	209	78.3	69.43	506.0	20.56	597.0	568	79.82	488.2	82.0%	10.94	260.0	24.0	0.86	0.88	1.24
8	161	27.5	88.0	14.0	54.0	82 86	28.86	203	76.7	60.60	499.0	11.70	571.0	547	62.94	484.06	85.0	28.32	265.0	20.0	1.0	1.0	1.09
1	151	28.0	84.5	14.0	44.65	83.66	39.01	200	71.3	61.50	502.0	13.3	576.0	547	57.41	489.59	84.0	28.17	417.0	14.8	1.35	1.36	0.80
4	152	8.2	84.5	− 8.0	56.65	85.40	28.75	218	68.0	65.96	502.0	16.28	584.0	559	75.58	483.42	83.0	14.68	296.0	20.2	1.0	1.0	1.10
6	105	7.6	66.0	−11.0	46.0	60.22	14.22	192	52.5	67.03	514.0	13.72	595.0	502	63.61	498.39	85.0	14.84	615.0	41.0	0.49	0.54	2.00

COST OF WATER, LINE 68, TABLE I.

Let G = consumption of water in gallons per minute per ton of capacity in 24 hours.

Let P = price of water per 1,000 cubic feet in cents.

Then $\frac{P}{8.5 \times 1{,}000}$ = cost in cents of one gallon of water.

$1440\,G$ = gallons consumed per one ton of capacity in 24 hours.

Cost of water per ton of refrigerating effect $= \frac{1440\,GP}{8500} = 0.169$ cents.

Example: Let $G = 1.0$ and
Let P = one dollar, or 100 cents, an ordinary city price; then cost of water for one ton of effect = 0.169 dollar

COST OF COAL, LINE 67, TABLE I.

Let C = price of a ton of coal in cents.

Let N = pounds of ice refrigerating effect per pound of coal.

Then $\frac{2000}{N}$ = pounds of coal to produce one ton of refrigerating effect.

$\frac{C}{2000}$ = cost of one pound of coal.

$\frac{C}{2000} \times \frac{2000}{N} = \frac{C}{N}$ = cost of coal for one ton of refrigerating effect.

Example: Let $C = 400$ cents, an ordinary price.
Let $N = 24.1$, as in trial No. 1 or 8.

Then cost of coal for one ton of refrigerating effect, = \$0.166.

APPROXIMATE RULE FOR ESTIMATE OF CONDENSING WATER.

From the foregoing discussion it appears that the condensing water may be estimated approximately as follows:

Let r_2 = latent heat at suction pressure, British thermal units.
Let r_1 = " condensing " "
Let t_4 = temperature of condensing water at inlet.
Let t_1' = " ammonia at exit from condenser.
Let t_2 = boiling point of ammonia at suction pressure.
Let t_1 = " " condensing pressure.

G = gallons of water per minute per ton of capacity per 24 hours.

Then $$t_1' = t_1 - 0.025r_1. \quad . \quad . \quad . \quad . \quad . \quad . \quad (A)$$

$r_2 - (t_1' - t_2)$ = thermal units per pound of ammonia producing refrigeration.

$1.125r_1 + t_1 - t_1'$ = thermal units per pound of ammonia absorbed by cooling water.

$\dfrac{1.125r_1 + t_1 - t_1'}{t_2 - t_4}$ = pounds condensing water per pound of ammonia.

Hence, $G = 26.3 \times \dfrac{1.125r_1 + t_1 - t_1'}{(t_1 - t_4)\,[r_2 - (t_1' - t_2)]}$.

Or, substituting t_1' from (A),

$$G = 26.3 \times \frac{1.1r_1}{(t_1 - t_4)\,[r_2 - (t_1 - t_2) - 0.025r_1]}.$$

EXAMPLE.—Let condensing pressure = 150 lbs. above atmosphere.

Let suction pressure = 28 lbs. above atmosphere.

Let $t_4 = 56°$ Fahr.

Then $r_2 = 547$, $r_1 = 500$, $t_1 = 88°$, $t_2 = 14°$,

$$G = 26.3 \times \frac{500}{32 \times (547 - 74 - 12.5)} = \frac{11465}{14866} = 1.98.$$

GENERAL PRINCIPLES CONTROLLING CAPACITY AND COST OF COAL AND WATER.

CASE I.—Let the lowest temperature of brine required be about 28.0° Fahr., which suffices for cooling the storage cellars of breweries, and represents about the highest temperature for which machines are employed. Then a suction pressure may be used, for which the boiling point of ammonia is about 14° below the lowest brine limit. Such a pressure is 28 lbs. above the atmosphere. Let the condensing pressure be assumed at 150 lbs. above the atmosphere.

Then if we assume that the supply of cooling water shall be such as to permit a range of 30° in the condenser, the amount of cooling water will be 1 gallon per ton of capacity, which at the price of $1 per 1,000 cubic feet will make the cost of water 17 cents per ton of capacity. The mechanical work to operate

COST OF WATER, LINE 68, TABLE I.

Let G = consumption of water in gallons per minute per ton of capacity in 24 hours.

Let P = price of water per 1,000 cubic feet in cents.

Then $\frac{P}{8.5 \times 1,000}$ = cost in cents of one gallon of water.

$1440G$ = gallons consumed per one ton of capacity in 24 hours.

Cost of water per ton of refrigerating effect $= \frac{1440GP}{8500} = 0.169$ cents.

Example: Let $G = 1.0$ and

Let P = one dollar, or 100 cents, an ordinary city price; then cost of water for one ton of effect = 0.169 dollar

COST OF COAL, LINE 67, TABLE I.

Let C = price of a ton of coal in cents.

Let N = pounds of ice refrigerating effect per pound of coal.

Then $\frac{2000}{N}$ = pounds of coal to produce one ton of refrigerating effect.

$\frac{C}{2000}$ = cost of one pound of coal.

$\frac{C}{2000} \times \frac{2000}{N} = \frac{C}{N}$ = cost of coal for one ton of refrigerating effect.

Example: Let $C = 400$ cents, an ordinary price.

Let $N = 24.1$, as in trial No. 1 or 8.

Then cost of coal for one ton of refrigerating effect, = $0.166.

APPROXIMATE RULE FOR ESTIMATE OF CONDENSING WATER.

From the foregoing discussion it appears that the condensing water may be estimated approximately as follows:

Let r_2 = latent heat at suction pressure, British thermal units.
Let r_1 = " condensing " "
Let t_4 = temperature of condensing water at inlet.
Let t_1' = " ammonia at exit from condenser.
Let t_2 = boiling point of ammonia at suction pressure.
Let t_1 = " " condensing pressure.

G = gallons of water per minute per ton of capacity per 24 hours.

Then $$t_1' = t_1 - 0.025r_1. \quad . \quad . \quad . \quad . \quad . \quad . \quad (A)$$

$r_2 - (t_1' - t_2)$ = thermal units per pound of ammonia producing refrigeration.

$1.125r_1 + t_1 - t_1'$ = thermal units per pound of ammonia absorbed by cooling water.

$\dfrac{1.125r_1 + t_1 - t_1'}{t_2 - t_4}$ = pounds condensing water per pound of ammonia.

Hence, $$G = 26.3 \times \frac{1.125r_1 + t_1 - t_1'}{(t_1 - t_4)\,[r_2 - (t_1' - t_2)]}.$$

Or, substituting t_1' from (A),

$$G = 26.3 \times \frac{1.1r_1}{(t_1 - t_4)\,[r_2 - (t_1 - t_2) - 0.025r_1]}.$$

EXAMPLE.—Let condensing pressure = 150 lbs. above atmosphere.

Let suction pressure = 28 lbs. above atmosphere.

Let $t_4 = 56°$ Fahr.

Then $r_2 = 547$, $\quad r_1 = 500$, $\quad t_1 = 88°$, $\quad t_2 = 14°$,

$$G = 26.3 \times \frac{500}{32 \times (547 - 74 - 12.5)} = \frac{11465}{14866} = 1.98.$$

GENERAL PRINCIPLES CONTROLLING CAPACITY AND COST OF COAL AND WATER.

CASE I.—Let the lowest temperature of brine required be about 28.0° Fahr., which suffices for cooling the storage cellars of breweries, and represents about the highest temperature for which machines are employed. Then a suction pressure may be used, for which the boiling point of ammonia is about 14° below the lowest brine limit. Such a pressure is 28 lbs. above the atmosphere. Let the condensing pressure be assumed at 150 lbs. above the atmosphere.

Then if we assume that the supply of cooling water shall be such as to permit a range of 30° in the condenser, the amount of cooling water will be 1 gallon per ton of capacity, which at the price of $1 per 1,000 cubic feet will make the cost of water 17 cents per ton of capacity. The mechanical work to operate economy of coal and water combined, at 28 lbs. suction pressure, is not practically affected by an increase of condensing pressure from 150 to 215 lbs. above the atmosphere.

But a decrease of condensing pressure below 150 lbs. requires so great an increase of cost of water that the economy rapidly decreases. The cause of this wide difference of results caused by about an equal number of pounds variation of condensing pressure either side of 150 lbs. is, that the 8° increase of boiling point due to the 40 lbs. increase of pressure increases the range of the condensing water from 33° to 38°, or only about 25%, while the decrease of pressure, lowering the boiling point 22°, alters the range in the condenser 66%, and proportionally increases the water supply.

Also the changes in the latent heat at condensing pressure and variation in curvature of the compression curve favor a compensation between the increased cost of power and decreased cost of water in the case of the increase of pressure, whereas they are opposed to such compensation in the case of the reduction of condensation pressure.

CASE V.—Suppose that a temperature of about zero Fahr. is required in the brine or other material to be refrigerated.

This is frequently the case in public cold-storage warehouses. Then we must have a suction pressure affording a boiling point of about − 12° Fahr.

Such pressure is about 7.0 lbs. above the atmosphere.

Let the condensing pressure be 150 lbs. and the cooling water one gallon per ton, with an initial temperature of 56° Fahr., as in Case I.

The weight of ammonia circulated will now be about as much less than in Case I., as the absolute suction pressure 7.0 + 14.7 = 22 lbs. is less than the absolute suction pressure 28.0 + 14.7 = 43 lbs. in Case I., or in round numbers *one-half* as much ammonia will be circulated, and therefore the refrigerating capacity is only half that of Case I.*, assuming the same revolutions per minute. But the power to operate the machine, which is proportional to the area *KNEP*, Fig. 120, is only 20% less than that for Case I., while the proportion of latent heat available for refrigerating external bodies is only 0.95 as much as for Case I. Conse-

* The weight of ammonia is not quite reduced one-half by halving the suction pressure, but as the proportion of latent heat available for refrigeration is only 0.95 of that for Case I., the capacity is very nearly half.

quently the economy or the cooling effect per unit of power or per lb. of coal is equal to $\frac{0.5 \times 0.95}{0.8} \times 24.1 = 14.2$ lbs. of ice per lb. of coal. The cost of water and coal per ton of effect at above prices would be 45 cents. Increasing or decreasing the condensing pressure, by varying the allowance of cooling water, will produce practically the same variations of cost as in Cases II. and III., slightly modified by the difference of curvature of the curves *KP* and *AF*, and the relative latent heats.

It therefore follows that for all brine temperatures or all suction pressures, a price of $1 per 1,000 cubic feet for water and $4 per ton for coal, maximum economy of coal and water combined, is attained at any condensing pressure between 150 and upwards of 215 lbs.

Maximum economy of coal only is obtained by using the highest suction pressure consistent with the desired refrigeration, and the lowest condensing pressure available by the use of as large a supply of cooling water as can be procured. But these conditions at the above prices for water and coal make the cost per ton of capacity for coal and water together about 1.75 times the same cost at 150 to 215 lbs. condensing pressure. If the cost of water is 50 cents per 1,000 cubic feet with coal at $4 per ton, then the combined cost of coal and water per ton of capacity for 105 lbs. condensing pressure and 28 lbs. suction pressure is the same as for 150 to 215 lbs. and 28 lbs. suction pressure, viz. about 34 cents. The coal economy is then at the maximum consistent with practical refrigeration, viz., 34.5 lbs. of ice effect per pound of coal for non-condensing steam-engines using 3 lbs. of (or about 26 lbs. of steam) coal per hour per horse-power, unless water is sufficiently cheap to permit the use of condensing pressures lower than 105 lbs. above the atmosphere.

The use of lower suction pressures imposes a reduction of coal economy, so that at $1 per 1,000 cubic feet for water and $4 per ton for coal, the least cost for both these items is about 45 cents per ton of capacity, and obtains when the condensing pressure is 150 to 215 lbs. A reduction of condensing pressure to 105 lbs. by excessive use of water would increase this cost about 50%, so that to make the cost the same at 105 lbs. as at 150 to 215 lbs., the price of water would have to be about 50 cents per 1,000 cubic feet, with coal at $4 per ton. The pounds of ice effect per pound of coal would then be about 20 for non-condensing steam-engines using 3 lbs. of coal per hour per horse-power.

These conclusions practically apply also to refrigerating machines of the ammonia absorption type.

If a non-compound condensing engine consuming about 20 lbs. of steam per hour per horse-power can be used, which is only possible where water costs practically nothing, the coal economy at 105 lbs. condensing and 28 lbs. suction pressure may be about 43 lbs. of ice effect per pound of coal, and 26 lbs. at 7 lbs. suction pressure.

If a compound condensing engine of the best type, using about 16 lbs. of steam per hour per horse-power, can be used, the coal economy at 105 lbs. condensing and 28 lbs. suction pressure may be about 54 lbs. of ice effect per pound of coal, and about 33 lbs. at 7 lbs. suction pressure.

STANDARDS OF PRESSURE AND AVERAGE INDICATOR CARDS OF TRIALS—THEORETICAL TEMPERATURE DUE COMPRESSION.

The pressure gauges were carefully calibrated by reference to a Shaw mercury pressure gauge, and also with a special gauge made to order by the Ashcroft Manufacturing Co. The ammonia indicator was of the Tabor pattern, fresh from the makers, and provided with a certified spring, whose scale was 100 lbs. per square inch. The steam indicator was correct in all its features, and the indicator motion was a trully parallel one.

The proportions of the steam cards are indicated by Fig. 130, which is a facsimile of the average card of Trial No. 8. None of the other trials are represented, as the steam distribution was not an element of investigation.

Average ammonia cards from each of the trials are given in Figs. 122 to 129, except for Trial No. 8, which is selected to show a special effect.

The striking feature of these cards is the remarkably perfect behavior of the gas confined in the clearance spaces. It is usually believed that this gas can only attain the suction pressure by increasing its volume according to some regular law, as the piston recedes from the cylinder head. In this case the inlet gas valve cannot open until the increase of volume of the clearance gas has occurred, and the capacity or useful displacement of the pump is proportionately imperfect. It appears, however, that the gas confined in the clearance collapses by cooling, so that the reduction of its pressure to the suction limit takes place before the piston has commenced to recede, or during the travel of the latter over only a fraction of the distance which would be necessary to expand the gas adiabatically to suction pressure. The appearance of a perfectly square corner or heel on the cards, such as in Fig. 122 or 127, appears to be obtained more perfectly under the higher back pressures or lower outlet temperatures, as in Trials 1, 8, and 6.

All of the 48 cards taken during Trial No. 1 were like Fig. 122,* and six weeks later in Trial No. 8, the same conditions yielded but one rounded heel like Fig. 129, all of the remaining 40 cards then taken possessing the square heel like Fig. 125.

The fair conclusion is, that the displacement of the compressor pistons was mechanically perfect, the maximum clearance loss shown by the cards being less than 1%. Oil was fed into the compressors by gravity, from a reservoir hung above the cylinders, and having the suction pressure present above the oil, so that the head due the volume of the latter was available to cause the oil to flow into the suction pipe. This oil was collected by precipitation in the reservoir K, Fig. 110, and thence occasionally pumped by hand back into the oil reservoir.

The amount of oil fed was about one-fiftieth pint per minute to each cylinder, or one two-hundredth cubic inch per revolution. The volume of the clearance spaces was about 10 cubic inches. The oil could not, therefore, sensibly affect the displacement or the appearance of the card.

The curve of compression lies slightly below a hyperbola, whose law is $PV^{1.3} =$ constant, which coincides with the adiabatic † for superheated ammonia gas at 7.6 lbs. back pressure, and nearly for the other back pressures used.

Points marked a are on such a curve at 130 lbs. absolute pressure.

The excess of pressure during expulsion of gas from cylinder, above that in the reservoir, averages 5% of the mean effective pressure, which is equivalent to an increase of 12° Fahr. in the

* It is natural to connect so square a heel and so vertical a clearance line with the idea that the indicator was knocking when the cards were taken; such was not the case, however, as the cards were taken under special precautions to avoid this action.

† Equation of adiabatic for ammonia $P_1 V = BT_0 \left(\frac{P_1}{P_0}\right)^{0.24} - CP_1^{0.33}$.

temperature due compression of the gas. The difference between the actual temperature of the gas at exit from the compressor, and the temperature calculated from the mechanical work of compression, agrees fairly well with the heat given to the jacket water, and also with the formula

$$T = T_0 \left(\frac{P_1}{P_0}\right)^0 .$$

The following table exhibits these facts in detail. The temperatures of the gas at entrance to the compressor, except in No. 8, when it was observed, are obtained by adding 10° to the temperature observed half way between the compressor and brine tank, line 36, Table I., in the case of Nos. 2 to 7, and 4° in the case of No. 1. These amounts are based on a comparison of the changes of temperature between the brine tank and the half-way point, and while somewhat arbitrary, are very nearly correct.

TABLE SHOWING THEORETICAL TEMPERATURES DUE COMPRESSION, AND THEORETICAL EFFECT OF JACKET WATER.

Trial Number.	1	2	3	4	5	6	7	8
1. Horse-power developed in ammonia cyls.	65.7	46.0	54.8	54.7	52.0	48.2	59.4	71.2
2. Equivalent of horse-power British thermal units per minute	2786	1942	2325	2320	2205	2074	2518	3020
3. Per cent. of latter above perfect adiabatic card	6	5½	4	4½	6	5	5	4
4. Equivalent of latter in thermal units	166	104	93	103	132	103	126	120
5. Difference of lines 2 and 4, perfect adiabatic heat	2620	1838	2232	2217	2073	1971	2392	2900
6. Weight of ammonia per minute, lbs.	28.17	10.94	15.24	14.68	15.46	14.84	16.67	28.32
7. Increase of temperature of ammonia due heat in line 5	186°	321°	281°	300°	260°	260°	275°	200°
8. Temperature of gas at commencement of comp. = T_0	39°	25°	25°	25°	20°	15°	13°	34°
9. Increase of temperature of ammonia due heat in line 4	12°	19°	12°	14°	17°	14°	10°	8°
10. Sum of 7, 8, and 9, theoretical temperature due compression	237°	365°	318°	339°	297°	289°	298°	242°
11. Temperature of gas outlet of comp.	213°	259°	265°	263°	248°	222°	239°	221°
12. Temp. absorbed by jacket. Diff. 10 and 11	24°	106°	53°	76°	49°	67°	59°	21°
13. Theoretical thermal units heat absorbed by jacket	358	616	425	568	401	525	517	308
14. Heat absorbed by jacket, as measured	608	.....	.....	712	665	529	656	406
15. Condensing pressure = p_1 absolute	166	150	169	167	139	120	162	176
16. Suction pressure = p_0 absolute	43	15.7	22	24	24	22	28	42
17. Theoretical temperature equivalent to line 10, $T = T_0 \left(\frac{P_1}{P_0}\right)^{0.24}$	229	370	331	304	275	253	260	237

SPECIAL INDICATOR CARDS AT VARIABLE SPEED OIL FEED AND CLEARANCE VOLUMES.

In order to determine the influence of these elements upon the form of the indicator card from the ammonia cylinders, diagrams Figs. 131 to 140 were taken. Several cards under each condition

24

are given in order to show to what extent the compression line varies from a given law. The cylinder head was removed, and all oil wiped away.

Cards were then taken at 48 and 30 revolutions per minute with the piston traveling within one-thirty-second of an inch of the cylinder head. Figs. 131 and 140.

The cubic inches of clearance space then aggregated three-tenths per cent. of the piston displacement. The piston rod was then screwed into the cross head, so that the piston traveled only within five-thirty-seconds of the head.

The clearance space then aggregated about three times as much or nine-tenths per cent. of the piston displacement.

Cards were again taken at 30 and 48 revolutions without oil. Figs. 132 and 133. Contrary to expectation, this large increase of clearance did not produce a proportional increase in the curvature of the line *cf* of the card, which should theoretically be an adiabatic curve such as the dotted line *yy* in the enlarged diagrams. Figs. 141 to 145. The fact that the actual line of the cards is more nearly vertical is attributable to the collapse of the highly heated ammonia gas, due to the chilling action of the piston. At the end of the upward stroke the gas in the clearance spaces is confined as a thin layer between the piston and cylinder head, and the former has been in contact with the refrigerated gas during the entire upward stroke. Figs. 134 to 137 show the effect of introducing oil in sufficient quantity to at least fill a considerable fraction of the clearance space at each stroke.

The resulting cards are not practically affected by the oil. With such small amount of clearance as can be safely maintained with a single-acting compressor there seems to be no good reason for using oil, except to the small extent which good lubrication may require.

Figs. 138 to 140 show the effect of removing the water from the jackets. No sensible change in the law of compression can be found. The jackets may remove upwards of half the heat represented by tho power spent in compression, and yet not sensibly reduce the amount of the latter.

This fact, long since established in connection with the compression of air, is easily explainable on the ground that the cooling influence of the jacket is mainly confined to the period of the piston's motion occurring after the compression is com-temperature due compression of the gas. The difference between the actual temperature of the gas at exit from the compressor, and the temperature calculated from the mechanical work of compression, agrees fairly well with the heat given to the jacket water, and also with the formula

$$T = T_0 \left(\frac{P_1}{P_0}\right)^0 .$$

The following table exhibits these facts in detail. The temperatures of the gas at entrance to the compressor, except in No. 8, when it was observed, are obtained by adding 10° to the temperature observed half way between the compressor and brine tank, line 36, Table I., in the case of Nos. 2 to 7, and 4° in the case of No. 1. These amounts are based on a comparison of the changes of temperature between the brine tank and the half-way point, and while somewhat arbitrary, are very nearly correct.

TABLE SHOWING THEORETICAL TEMPERATURES DUE COMPRESSION, AND THEORETICAL EFFECT OF JACKET WATER.

	Trial Number.	1	2	3	4	5	6	7	8
1	Horse-power developed in ammonia cyls.	65.7	46.0	54.8	54.7	52.0	48.2	59.4	71.2
2	Equivalent of horse-power British thermal units per minute	2786	1942	2325	2320	2205	2074	2518	3020
3	Per cent. of latter above perfect adiabatic card	6	5¼	4	4½	6	5	5	4
4	Equivalent of latter in thermal units	166	104	93	103	132	103	126	120
5	Difference of lines 2 and 4, perfect adiabatic heat	2620	1838	2232	2217	2073	1971	2392	2900
6	Weight of ammonia per minute, lbs.	28.17	10.94	15.24	14.68	15.46	14.84	16.67	28.32
7	Increase of temperature of ammonia due heat in line 5	186°	321°	281°	300°	260°	260°	275°	200°
8	Temperature of gas at commencement of comp. = T_0	39°	25°	25°	25°	20°	15°	13°	34°
9	Increase of temperature of ammonia due heat in line 4	12°	19°	12°	14°	17°	14°	10°	8°
10	Sum of 7, 8, and 9, theoretical temperature due compression	237°	365°	318°	339°	297°	289°	298°	242°
11	Temperature of gas outlet of comp.	213°	259°	265°	263°	248°	222°	239°	221°
12	Temp. absorbed by jacket. Diff. 10 and 11	24°	106°	53°	76°	49°	67°	59°	21°
13	Theoretical thermal units heat absorbed by jacket	358	616	425	568	401	525	517	308
14	Heat absorbed by jacket, as measured	608	.....	.....	712	665	529	656	406
15	Condensing pressure = p_1 absolute	166	150	169	167	139	120	162	176
16	Suction pressure = p_0 absolute	43	15.7	22	24	24	22	28	42
17	Theoretical temperature equivalent to line 10, $T = T_0 \left(\frac{P_1}{P_0}\right)^{0.24}$	229	370	331	304	275	253	260	237

SPECIAL INDICATOR CARDS AT VARIABLE SPEED OIL FEED AND CLEARANCE VOLUMES.

In order to determine the influence of these elements upon the form of the indicator card from the ammonia cylinders, diagrams Figs. 131 to 140 were taken. Several cards under each condition that the displacement of the piston is imperfect to the extent it would be if the contents of the clearance space expanded regularly as per the line *yyy*, Fig. 145.

The result would be that two-thirds of 1% of the volume swept through by the piston would not be expelled from the cylinder. As an outside limit we will call the efficiency of the compressor displacement 99% of the volume swept through by the piston.

Applying this basis to compute the volume of gas circulated, and hence the weight of ammonia which should pass the meter, we find that only about three-fourths as much ammonia is thus accounted for as shown by the meter.

Lines 32 and 33, Table I., show this fact. There was practically no leakage past the pistons, as the packing of the latter was in the most perfect condition, and no evidence of leakage could be detected. I therefore conclude that the gas entering the cylinders was heated by the sides of the latter, and thereby so rarefied as to cause a cylinder full of gas to weigh upwards of 25% less than it would if the gas remained at the temperature of entrance while the cylinder filled. The temperature to which it would be necessary to heat the gas is given in line 34 of Table I. The action would be similar to the phenomenon of cylinder condensation in the steam engines, but reversed in its order of acting. A loss of economy is caused, and the pumping or refrigerating capacity of a given size of cylinder is made less. The following is an example of the calculation of the temperature in line 34 in the case of trial No. 8.

We have :

Suction pressure absolute............................ = 43 lbs. per sq. in. abs.
Absolute temperature at entrance to compressor...... = 475° Fahr.
Revolutions per minute............................ 59.9.
99% displacement of pistons per revolution...... ..... 3.89 cu. ft.
Volume of 1 lb. of ammonia gas at 42 lbs. pressure and
at 475°.................................... $= V = \dfrac{BT - CP^{.33}}{P}$;

or,* $$\frac{\frac{2}{3}T - 4.9\,(42)^{0.33}}{42} = 6.8 \text{ cubic feet.}$$

Pounds of ammonia pumped per minute $= \dfrac{3.89 \times 59.9}{6.8} = 34.5.$

" " shown by meter.................. = 28.34.
Per cent. of displacement unaccounted for............ 21.4%.

* This is Zeuner's formula with English constants in round numbers.

Required the temperature to which the gas must be heated to swell its volume.......................... 21.4%.

Let T_1 be this temperature, then we must have

$$6.8 \times 1.214 = \frac{\frac{3}{4}\, T_1 - 4.9 \times (42)^{0.22}}{42}\,; \text{ whence } T_1 = 5.41.$$

Hence the gas would have be heated 66° above its temperature of entrance to the cylinder.

The weight of gas pumped per revolution is.......... 0.46 lb.

To heat this 68° requires $0.508 \times 0.46 \times 68$........... = 17 thermal units.

The surface of cast iron with which the gas is in contact amounts to 18 square feet. If this surface should give heat sufficient to cool itself 25° Fahr. to a depth of only one one-hundredth of an inch, the heat thus supplied would amount to

$$18 \times \frac{1}{100} \times \frac{1}{12} \times 25 \times \left(\text{Sp. heat of iron} = \frac{1}{10}\right)$$
$$\times\ (\text{wt. of cu. ft. of iron} = 480 \text{ lb.}) = 18 \text{ ther. units.}$$

Hence the heat stored in the interior surfaces of the cylinders during compression may easily produce the heating and rarefaction of the gas during suction necessary to account for the 21.4% of ammonia unaccounted for by the meter. The other trials give similar results.

RADIATION OF AMMONIA GAS PIPES, LINES 48*a* AND 50*a*, TABLE I.

In trials 1 and 8 the ammonia gas gained about 10° during its passage from the top of the brine tank to the compressor. For the other trials about twice this difference of temperature was gained. The pipe was 4 inches diameter, 20 feet in length, and heavily covered with snow. The gain of heat per minute was therefore as follows:

HEAT GAINED BY AMMONIA FROM ATMOSPHERE.

No. of Trial.	Weight of ammonia per min. lbs.	British Thermal Units.	
		Taken from atmosphere.	Per cent. of total heat involved.
1	28.17	140	0.8%
2	10.94	110	1.5%
3	15.24	152	1.6%
4	14.68	147	1.5%
5	15.46	155	1.6%
6	14.84	148	1.5%
7	16.67	167	1.4%
8	28.32	141	0.8%

that the displacement of the piston is imperfect to the extent it would be if the contents of the clearance space expanded regularly as per the line *yyy*, Fig. 145.

The result would be that two-thirds of 1% of the volume swept through by the piston would not be expelled from the cylinder. As an outside limit we will call the efficiency of the compressor displacement 99% of the volume swept through by the piston.

Applying this basis to compute the volume of gas circulated, and hence the weight of ammonia which should pass the meter, we find that only about three-fourths as much ammonia is thus accounted for as shown by the meter.

Lines 32 and 33, Table I., show this fact. There was practically no leakage past the pistons, as the packing of the latter was in the most perfect condition, and no evidence of leakage could be detected. I therefore conclude that the gas entering the cylinders was heated by the sides of the latter, and thereby so rarefied as to cause a cylinder full of gas to weigh upwards of 25% less than it would if the gas remained at the temperature of entrance while the cylinder filled. The temperature to which it would be necessary to heat the gas is given in line 34 of Table I. The action would be similar to the phenomenon of cylinder condensation in the steam engines, but reversed in its order of acting. A loss of economy is caused, and the pumping or refrigerating capacity of a given size of cylinder is made less. The following is an example of the calculation of the temperature in line 34 in the case of trial No. 8.

We have:

Suction pressure absolute.......................... = 43 lbs. per sq. in. abs.
Absolute temperature at entrance to compressor...... = 475° Fahr.
Revolutions per minute.......................... 59.9.
99% displacement of pistons per revolution.......... 3.89 cu. ft.
Volume of 1 lb. of ammonia gas at 42 lbs. pressure and
at 475°.................................... $= V = \frac{BT - CP^{.32}}{P}$;

or,* $$\frac{\frac{2}{3}T - 4.9\,(42)^{0.32}}{42} = 6.8 \text{ cubic feet.}$$

Pounds of ammonia pumped per minute $= \frac{3.89 \times 59.9}{6.8} = 34.5.$

" " shown by meter.................. = 28.34.
Per cent. of displacement unaccounted for............ 21.4%.

---

* This is Zeuner's formula with English constants in round numbers.

1. As all pressures are upwards of 40 lbs. per square inch, it is assumed that some coefficient of friction between 5 and 10% will apply to all external bearings.

2. The friction of the stuffing boxes or piston rods is assumed at 6 lbs. per square inch of diameter of the rod. Measurement of the friction of packing made by the writer in the case of a small steam-engine affords this data, which, though meagre, is all that is available, and, as will be seen, no essential error follows from its approximate character

3. The friction of the slide valves is computed on the assumption of a mean unbalanced pressure equal to half of the boiler pressure and a coefficient of friction of 10%, which is deduced from Mr. Gidding's experiments with his valve dynamometer.—*Transactions A. S. M. E.*, Vol. VII.

4. The thrust upon the eccentric is taken at 25 lbs. more than the force to overcome friction of the slide valves. This allows for its weight and some friction in valve mechanism.

5. The piston ring friction is computed on the assumption of 10 lbs. pressure under the rings and 2% coefficient of friction.

This pressure is known as nearly that due the piston springs, and the rings were so snugly fitted that no sensible steam pressure could be beneath them. The coefficient of friction accords with that determined for good lubrication by the writer's experiments on the friction of packing rings in steam-cylinders.—See *Transactions A. S. M. E.*, Vol. X.

The sum of the friction of the pistons, stuffing boxes, and valves only amounts to about 6% of the entire friction of the machine. Consequently the approximate character of the constants upon which these items are based cannot introduce any important discrepancy into the calculations of the friction of the other parts of the machine.

We may, therefore, subtract the friction of the above parts from the total friction, as determined by the difference of indicator cards, and determine what value for the coefficient of friction will, according to Morin's laws, make the total calculated friction equal that given by the experiment. As a sample case, to fix ideas, suppose the mean effective pressure to have been 71 lbs. per square inch for the ammonia-cylinders, and 33 lbs. for the steam cylinders, also that the revolutions were 59 per minute. The indicated steam-power would then be about 89 H.P., and by the simple calculation of multiplying together the normal

For the heat lost in passing from the compressor to the condenser, through a system of piping exposing about 50 square feet of black painted iron, we have the following:

HEAT LOST BY GAS BETWEEN COMPRESSOR AND CONDENSER.

No. of Trial.	Weight of ammonia, lbs.	Fall of temperature, Fahr. deg.	British Ther. Units.		Temperature of atmosphere, Fahr. degrees.	Mean temperature of surface.	Difference of temperature producing radiation.	Thermal units lost per hour per sq. ft. of surface, per degree diff. of temp.
			Lost per min.	Per cent. of total heat received.				
1	28.17	13.0	182.0	1.0%	68	206	138	1.6
2	10.94	50.0	275.0	4.5%	63	234	171	1.9
3	15.24	44.0	330.0	3.5%	64	243	179	2.2
4	14.68	45.0	338.0	3.5%	64	240	176	2.3
5	15.46	43.0	328.0	3.5%	69	226	163	2.4
6	14.84	30.0	222.0	2.5%	63	207	144	1.8
7	16.67	30.0	250.0	2.5%	67	224	157	1.9
8	28.32	18.0	252.0	1.5%	68	212	144	1.7

The rate of loss per square foot evidently increases very rapidly with temperature, as is well known to be the case with all superheated vapors.

LOSS OF POWER IN FRICTION OF MECHANISM.

The following computations show that the amount of friction shown for the several trials, line 62, Table I., is quite consistent with the average loss to be expected with any similar mechanism, under the best practical conditions of lubrication.

CONSTANTS FOR CALCULATION OF FRICTION.

Diam. steam-cyl.	18 in.	Stroke	36 in.
Diam. ammonia-cyl.	12 in.	Stroke	30 in.
Diam. piston rod, ammonia-cyls.	$2\frac{3}{4}$ in.	Area Piston Packing Rings.	
Diam. piston rod, steam cyls.	3 in.	Ammonia-cyl	100 sq. in.
Diam. eccentric	10 in.	Steam-cyl	144 sq. in.

Weight of piston, piston rod, cross-head and connecting rod, each ammonia-cylinder	825 lbs.
Weight of piston, piston rod, cross-head and connecting rod, each steam-cylinder	1,370 lbs.
Weight of main shaft and two fly-wheels	18,500 lbs.
Diam. ammonia crank-pins	4 in.
Diam. steam crank-pins	6 in.
Diam. main shaft bearings	8 in.
Average boiler pressure	90 lbs.
Average cut-off	$\frac{1}{4}$ lb.
Average area of slide valves	16 sq. in.
Average travel	2 in.

1. As all pressures are upwards of 40 lbs. per square inch, it is assumed that some coefficient of friction between 5 and 10% will apply to all external bearings.

2. The friction of the stuffing boxes or piston rods is assumed at 6 lbs. per square inch of diameter of the rod. Measurement of the friction of packing made by the writer in the case of a small steam-engine affords this data, which, though meagre, is all that is available, and, as will be seen, no essential error follows from its approximate character

3. The friction of the slide valves is computed on the assumption of a mean unbalanced pressure equal to half of the boiler pressure and a coefficient of friction of 10%, which is deduced from Mr. Gidding's experiments with his valve dynamometer.—*Transactions A. S. M. E.*, Vol. VII.

4. The thrust upon the eccentric is taken at 25 lbs. more than the force to overcome friction of the slide valves. This allows for its weight and some friction in valve mechanism.

5. The piston ring friction is computed on the assumption of 10 lbs. pressure under the rings and 2% coefficient of friction. This pressure is known as nearly that due the piston springs, and the rings were so snugly fitted that no sensible steam pressure could be beneath them. The coefficient of friction accords with that determined for good lubrication by the writer's experiments on the friction of packing rings in steam-cylinders.—See *Transactions A. S. M. E.*, Vol. X.

The sum of the friction of the pistons, stuffing boxes, and valves only amounts to about 6% of the entire friction of the machine. Consequently the approximate character of the constants upon which these items are based cannot introduce any important discrepancy into the calculations of the friction of the other parts of the machine.

We may, therefore, subtract the friction of the above parts from the total friction, as determined by the difference of indicator cards, and determine what value for the coefficient of friction will, according to Morin's laws, make the total calculated friction equal that given by the experiment. As a sample case, to fix ideas, suppose the mean effective pressure to have been 71 lbs. per square inch for the ammonia-cylinders, and 33 lbs. for the steam cylinders, also that the revolutions were 59 per minute. The indicated steam-power would then be about 89 H.P., and by the simple calculation of multiplying together the normal ence of the wheel, there would be a saving of friction amounting to 2.35 H.P. The total saving due double action would, therefore, be from 1.25 H.P. to 3.45 H.P., or from 1.4% to 4% of the entire power to operate the machine. Assuming that the fly-wheel can always be designed so as to weigh the same for either arrangement, the saving in friction by double action in the case of ammonia compressors is not so important as the advantages claimed for single action; namely, facility for examination of internal parts, maintenance of a minimum clearance, without danger from wear of crank-pin boxes, and a minimum pressure of gas at the piston rod stuffing boxes.

SPECIFIC HEAT OF BRINE.

For the determination of specific heat the management of the brewery cordially devoted to my use one of the storage vaults of their establishment.

A room was thereby available which was capable of being maintained, by the refrigerating system, at about 36° Fahr. for any length of time.

I. SPECIFIC HEAT AT ABOUT 32° FAHR.

The apparatus employed is shown in Fig. 121.

A cylindrical reservoir *L*, about 16 inches diameter and 20 inches high, is mounted upon the wooden case *D*. This vessel is covered with one-inch thick hair felt. An inner cylinder *K*, about 6 inches diameter, fits concentrically into *L*, and is open at both ends. Hot water placed in the interior of *L* can be maintained at a practically constant temperature for about an hour in a room at low temperature.

Distilled water heated to about the temperature of the interior of *L* is placed in a copper flask *F* provided with a dasher *G*. This flask being suspended in *K*, gradually assumes the temperature of the latter to any desired degree of approximation, as determined by the thermometers *A*, *B*, and *C*, inside and outside the flask respectively.

The brine is placed in the open copper vessel *E*, which rests upon insulated supports in the bottom of *D*, and fully exposed to the atmosphere of the experimenting room. The padded slide *H* seals the lower end of *K* until the temperature of the flask and the rate of radiation of the brine vessel have been determined. Then the slide is drawn out long enough to allow the

TABLE.

INTERNAL FRICTION SINGLE ACTING AMMONIA COMPRESSION ICE MACHINE.

No. of Experiment.	1	2	3	4	5	6	7
Date of Test	Mar. 13	Apr. 2	Apr. 4	Apr. 5	Apr. 6	Apr. 7	Apr. 26
Duration—hours	24	14¼	42	11¼	13¼	18	18
Revolutions per Minute	58.1	58.7	58.0	57.8	57.9	59.0	58.9
Mean Effective Pressure:							
Ammonia Pumps, pounds per square inch	32.5	22.12	27.2	25.4	22.9	27.8	33.0
Mean Effective Pressure:							
Steam-cylinder, pounds per square inch	65.9	45.75	55.0	52.4	47.5	59.9	71.0
Horse-power, Steam-cylinder	85.0	59.28	72.5	67.1	59.6		88.6
Horse-power, both Ammonia Pumps	65.7	49.02	54.8	51.9	46.5	Add 1¼ per ct.	71.2
Horse-power expended in Friction Experiment	19.3	13.26	17.7	15.2	13.1	14.4	17.45
Per cent. of Indicated Steam-power:	23.0	22.0	24.5	23.0	22.0	20.0	23.4 per cent.
Coefficient of Friction for External Bearings which makes Computed Friction agree with Experimental Amount	8.8	7.38	9.15	7.8	7.35	7.7	7.6

It is certainly reasonable, therefore, to believe that Morin's laws, with a coefficient of friction of about 8%, are applicable to determine questions of friction regarding this machine. Consequently, we may conclude that the power consumed in the friction is distributed as follows:

	Per cent. of whole.
Crank-pins and thrust on Main Shaft	54.0
Eccentric	00.2
Cross-head Slides	5.0
Wrist Pins	3.0
Weight of Main Shaft	31.0
Piston Packing	4.0
Piston Rods	2.5
Valves	0.30
	100.00

We may also venture to estimate the saving in friction due to using one double-acting compressor cylinder instead of two single-acting cylinders.

This would be:

1. The friction due to the two suction strokes occurring in one revolution, amounting to 0.94 H.P.

2. The friction due half the weight of the main shaft, whose entire weight is 2,500 lbs. This would amount to 0.35 H.P.

3. If the fly-wheels were combined into one of equal diameter no modification of friction would occur, but if by virtue of doubling the diameter, a single wheel was made to have 50% less weight, as might be the case, without altering the dynamic influ-

ence of the wheel, there would be a saving of friction amounting to 2.35 H.P. The total saving due double action would, therefore, be from 1.25 H.P. to 3.45 H.P., or from 1.4% to 4% of the entire power to operate the machine. Assuming that the fly-wheel can always be designed so as to weigh the same for either arrangement, the saving in friction by double action in the case of ammonia compressors is not so important as the advantages claimed for single action; namely, facility for examination of internal parts, maintenance of a minimum clearance, without danger from wear of crank-pin boxes, and a minimum pressure of gas at the piston rod stuffing boxes.

SPECIFIC HEAT OF BRINE.

For the determination of specific heat the management of the brewery cordially devoted to my use one of the storage vaults of their establishment.

A room was thereby available which was capable of being maintained, by the refrigerating system, at about 36° Fahr. for any length of time.

I. SPECIFIC HEAT AT ABOUT 32° FAHR.

The apparatus employed is shown in Fig. 121.

A cylindrical reservoir L, about 16 inches diameter and 20 inches high, is mounted upon the wooden case D. This vessel is covered with one-inch thick hair felt. An inner cylinder K, about 6 inches diameter, fits concentrically into L, and is open at both ends. Hot water placed in the interior of L can be maintained at a practically constant temperature for about an hour in a room at low temperature.

Distilled water heated to about the temperature of the interior of L is placed in a copper flask F provided with a dasher G. This flask being suspended in K, gradually assumes the temperature of the latter to any desired degree of approximation, as determined by the thermometers A, B, and C, inside and outside the flask respectively.

The brine is placed in the open copper vessel E, which rests upon insulated supports in the bottom of D, and fully exposed to the atmosphere of the experimenting room. The padded slide H seals the lower end of K until the temperature of the flask and the rate of radiation of the brine vessel have been determined. Then the slide is drawn out long enough to allow the of temperature of the brine, and the remaining third remains constantly at the temperature of the atmosphere. This assumption is approximate and quite arbitrary; but if in error 50% does not effect the second decimal place of the specific heat value, which is the limit by accuracy sought in the investigation. The total heat given to the brine vessel is therefore $\frac{2}{3} \times 59.3 \times 11.53 = 461.2$.

And to the brine itself $11.53 \times 6000 = 69180$.

The flask loses some heat during its transfer from the interior of L to the brine. The time occupied in making this transfer is ten seconds. The flask filled with hot water and exposed to the atmosphere lost heat at the rate of one-fifteenth degree per ten seconds. If it be assumed that the flask is exposed to a temperature equal to that of the atmosphere for half the interval of transfer, the loss of heat could not exceed one-thirtieth of a degree. The range of temperature of the flask was in all experiments upward of 60°, and hence the loss due to transfer concerns less than one-tenth of one per cent., and is neglectable under our assumed limit of error. The water equivalent of the flask was 30 grammes. We assume that two-thirds of this shared the range of temperature of the water and one-third the temperature of the fluid outside the flask.

The range of water temperatures was 69.6°. Since the outside of the flask was at 113.5°, the water was at 110.5°. One-third of the flask must be considered as falling through the matter range plus 3°. Again, since the brine at the end of the experiment was five-tenths degree lower than the water in the flask, one-third the latter must be considered as falling through the water range plus five-tenth degrees. The total heat given up by the metal of the flask is therefore equivalent to

$$30\left[69.6 + \frac{40.9 - 40.4}{3} + \frac{113.5 - 110.5}{3}\right]$$

or, 2923.1 heat units. The heat lost by the water contained in the flask is $69.6^* \times 800 = 55680$ heat units.

Let X represent the desired specific heat value; then we have $11.53 \times 6000 \times X + \frac{2}{3} \times 59.3 \times 11.53 \doteq 49.6 \times 800 + 30$, whence X equals 0.828.

* The variation of the specific heat of water over this range of temperature would alter the range less than one-hundredth of a degree, and is therefore neglectable.

Any error involved in the assumption regarding the distribution of the heat in the flask is unimportant, as the total heat units involved in the corrections for differences of temperature inside and outside the flask did not aggregate 100 heat units in any experiment, and this affects the specific heat value only in the third decimal place. The thermometers used were made by Green. They were graduated to fifths of degrees, and readings were probably correct to twentieths of degrees.

This fact would make the error due this source about five units in the third decimal place. The aggregate instrumental error from all sources is thought to be one unit in the second place of decimals. The actual error is best determined by the variation of repeated determinations of the specific heat of the same brine or of distilled water. The latter gave between 114° and 45°, a mean specific heat value of 0.984.

Six determinations of the brine used in the test of March 13, having a specific gravity of 1.163, and made with Liverpool salt, gave the following results:

1st test.	Specific heat	0.834
2d "	"	0.815
3d "	"	0.826
4th "	"	0.821
5th "	"	0.800
6th "	"	0.814
	Average,	0.818

Nine determinations of the brine used in the tests made between April 2 and April 27, made by the addition of American salt to the previous brine, and having a specific gravity of 1.174, gave results as follows:

1st test.	Specific gravity	0.763
2d "	"	0.803
3d "	"	0.786
4th "	"	0.798
5th "	"	0.803
6th "	"	0.768
7th "	"	0.786
8th "	"	0.774
9th "	"	0.796
	Average,	0.786

It therefore appears that the probable accidental error was about equal to the instrumental error, making the probable

error of the average figures between one and two places in the hundredth figure.

II. SPECIFIC HEAT OF BRINE NEAR ZERO FAHRENHEIT.

The determination of the absolute specific heat at this temperature was not attempted, as the difficulties are too great in proportion to the practical value of such a determination. For the purposes of the measurement of the performance of refrigerating machines it is sufficient to know that the specific heat at zero does not differ from that at or near 32° Fahr. by more than a fraction of the amount of error involved in the determination of the other elements upon which the measurement of performance depends. This was accomplished as follows:

Samples of brine were placed in tin vessels deposited in the experimenting vault, so that the brine assumed the temperature of the surrounding atmosphere.

Each vessel contained 400 grammes of brine, which occupied about half its contents. The copper brine vessel *E* was fitted with a thick lid, and placed in a galvanized iron pail about seven inches greater diameter than the vessel. The latter was then surrounded at the bottom and sides with hair felt upwards of an inch in thickness. Thus insulated, brine at minus two degrees Fahr. could be taken from the main brine tank of the refrigerating machine, be carried several hundred feet, through rooms at various temperatures, to the experimenting vault, and stand therein for upwards of a quarter of an hour before its temperature would reach zero Fahr. The rate of absorption of heat from the atmosphere could therefore be satisfactorily determined.

This having been done, one of the 400 gramme portions was poured into the cold brine, thereby causing the latter to rise in temperature through about two and one-half degrees in the neighborhood of zero, while the 400 gramme portion fell from about 36 to 3° Fahr.

From the temperature of the mixture can be determined how much the specific heat near zero differs from the mean specific heat between 36 and 3 degrees.

The average results of six experiments indicated that the specific heat of brine at zero Fahr. differed not more than about

Any error involved in the assumption regarding the distribution of the heat in the flask is unimportant, as the total heat units involved in the corrections for differences of temperature inside and outside the flask did not aggregate 100 heat units in any experiment, and this affects the specific heat value only in the third decimal place. The thermometers used were made by Green. They were graduated to fifths of degrees, and readings were probably correct to twentieths of degrees.

This fact would make the error due this source about five units in the third decimal place. The aggregate instrumental error from all sources is thought to be one unit in the second place of decimals. The actual error is best determined by the variation of repeated determinations of the specific heat of the same brine or of distilled water. The latter gave between 114° and 45°, a mean specific heat value of 0.984.

Six determinations of the brine used in the test of March 13, having a specific gravity of 1.163, and made with Liverpool salt, gave the following results:

1st test.	Specific heat	0.834
2d "	"	0.815
3d "	"	0.826
4th "	"	0.821
5th "	"	0.800
6th "	"	0.814
	Average,	0.818

Nine determinations of the brine used in the tests made between April 2 and April 27, made by the addition of American salt to the previous brine, and having a specific gravity of 1.174, gave results as follows:

1st test.	Specific gravity	0.763
2d "	"	0.803
3d "	"	0.786
4th "	"	0.798
5th "	"	0.802
6th "	"	0.768
7th "	"	0.786
8th "	"	0.774
9th "	"	0.796
	Average,	0.786

It therefore appears that the probable accidental error was about equal to the instrumental error, making the probable

error of the average figures between one and two places in the hundredth figure.

II. SPECIFIC HEAT OF BRINE NEAR ZERO FAHRENHEIT.

The determination of the absolute specific heat at this temperature was not attempted, as the difficulties are too great in proportion to the practical value of such a determination. For the purposes of the measurement of the performance of refrigerating machines it is sufficient to know that the specific heat at zero does not differ from that at or near 32° Fahr. by more than a fraction of the amount of error involved in the determination of the other elements upon which the measurement of performance depends. This was accomplished as follows:

Samples of brine were placed in tin vessels deposited in the experimenting vault, so that the brine assumed the temperature of the surrounding atmosphere.

Each vessel contained 400 grammes of brine, which occupied about half its contents. The copper brine vessel *E* was fitted with a thick lid, and placed in a galvanized iron pail about seven inches greater diameter than the vessel. The latter was then surrounded at the bottom and sides with hair felt upwards of an inch in thickness. Thus insulated, brine at minus two degrees Fahr. could be taken from the main brine tank of the refrigerating machine, be carried several hundred feet, through rooms at various temperatures, to the experimenting vault, and stand therein for upwards of a quarter of an hour before its temperature would reach zero Fahr. The rate of absorption of heat from the atmosphere could therefore be satisfactorily determined.

This having been done, one of the 400 gramme portions was poured into the cold brine, thereby causing the latter to rise in temperature through about two and one-half degrees in the neighborhood of zero, while the 400 gramme portion fell from about 36 to 3° Fahr.

From the temperature of the mixture can be determined how much the specific heat near zero differs from the mean specific heat between 36 and 3 degrees.

The average results of six experiments indicated that the specific heat of brine at zero Fahr. differed not more than about

Any error involved in the assumption regarding the distribution of the heat in the flask is unimportant, as the total heat units involved in the corrections for differences of temperature inside and outside the flask did not aggregate 100 heat units in any experiment, and this affects the specific heat value only in the third decimal place. The thermometers used were made by Green. They were graduated to fifths of degrees, and readings were probably correct to twentieths of degrees.

This fact would make the error due this source about five units in the third decimal place. The aggregate instrumental error from all sources is thought to be one unit in the second place of decimals. The actual error is best determined by the variation of repeated determinations of the specific heat of the same brine or of distilled water. The latter gave between 114° and 45°, a mean specific heat value of 0.984.

Six determinations of the brine used in the test of March 13, having a specific gravity of 1.163, and made with Liverpool salt, gave the following results:

1st test.	Specific heat	0.834
2d "	"	0.815
3d "	"	0.826
4th "	"	0.821
5th "	"	0.800
6th "	"	0.814
	Average,	0.818

Nine determinations of the brine used in the tests made between April 2 and April 27, made by the addition of American salt to the previous brine, and having a specific gravity of 1.174, gave results as follows:

1st test.	Specific gravity	0.763
2d "	"	0.803
3d "	"	0.786
4th "	"	0.798
5th "	"	0.803
6th "	"	0.768
7th "	"	0.786
8th "	"	0.774
9th "	"	0.796
	Average,	0.786

It therefore appears that the probable accidental error was about equal to the instrumental error, making the probable

error of the average figures between one and two places in the hundredth figure.

II. SPECIFIC HEAT OF BRINE NEAR ZERO FAHRENHEIT.

The determination of the absolute specific heat at this temperature was not attempted, as the difficulties are too great in proportion to the practical value of such a determination. For the purposes of the measurement of the performance of refrigerating machines it is sufficient to know that the specific heat at zero does not differ from that at or near 32° Fahr. by more than a fraction of the amount of error involved in the determination of the other elements upon which the measurement of performance depends. This was accomplished as follows:

Samples of brine were placed in tin vessels deposited in the experimenting vault, so that the brine assumed the temperature of the surrounding atmosphere.

Each vessel contained 400 grammes of brine, which occupied about half its contents. The copper brine vessel E was fitted with a thick lid, and placed in a galvanized iron pail about seven inches greater diameter than the vessel. The latter was then surrounded at the bottom and sides with hair felt upwards of an inch in thickness. Thus insulated, brine at minus two degrees Fahr. could be taken from the main brine tank of the refrigerating machine, be carried several hundred feet, through rooms at various temperatures, to the experimenting vault, and stand therein for upwards of a quarter of an hour before its temperature would reach zero Fahr. The rate of absorption of heat from the atmosphere could therefore be satisfactorily determined.

This having been done, one of the 400 gramme portions was poured into the cold brine, thereby causing the latter to rise in temperature through about two and one-half degrees in the neighborhood of zero, while the 400 gramme portion fell from about 36 to 3° Fahr.

From the temperature of the mixture can be determined how much the specific heat near zero differs from the mean specific heat between 36 and 3 degrees.

The average results of six experiments indicated that the specific heat of brine at zero Fahr. differed not more than about

one and one-half per cent. from its mean specific heat between 36° and zero.

Table III. gives the detailed data of the experiments. Corrections were carefully determined for the loss of heat to the air by pouring the 400 grammes out of the small vessels; also for the fact that all of the contents of the latter could not be transferred to the vessel containing the cold brine. By means of thermometers inserted at different parts of the hair felt covering of the cold brine vessel, it was determined that the heat absorbed by this felt was not a sensible factor. (See foot-note, Table III.) The thermometers used in these experiments were graduated to twentieths of degrees Fahr., and one degree was represented by one inch of length.

Readings were easily made to hundredths of a degree.

Possibly the instrumental error may be one per cent., but the error of the entire determination is probably upwards of two per cent. As the probable error of the refrigerating effect of the machine as expressed in pounds of ice per pound of coal, is from five to ten per cent, it may be assumed from the above determinations that if the specific heat of the brine is taken at 0.786 for all brine temperatures between April 2 and April 27, no sensible error will be involved. In the calculations of commercial results in Table I. the specific heat is taken roundly at 0.82, the nearest higher two decimal value for the higher gravity brine, and at 0.78 the nearest lower two decimals for the lower gravity samples.

and the above fictitious basis is substituted. The following conclusions are derived from the results of the investigation, and such deductions from the latter as will be found throughout the paper.

No. 1.—The capacity of the machine is proportional, almost entirely, to the weight of ammonia circulated. This weight depends on the suction pressure and the cubic displacement of the compressor pumps. The practical suction pressures range from 7 lbs. above the atmosphere, with which a temperature of zero Fahrenheit can be produced, to 28 lbs. above the atmosphere, with which the temperatures of refrigeration are confined to about 28° Fahr. At the lower pressure only about one-half as much weight of ammonia can be circulated as at the upper pressure, the proportion being about in accordance with the ratio of the absolute pressures, 22 and 42 lbs. respectively. For each cubic foot of piston displacement per minute a capacity of about one-sixth of a ton of "refrigerating effect" per 24 hours can be produced at the lower pressure, and of about one-third of a ton at the upper pressure. No other elements practically affect the capacity of a machine, provided the cooling surface in the brine tank or other space to be cooled is equal to about 36 square feet per ton of capacity at 28 lbs. back-pressure. For example, a difference of 100% in the rate of circulation of brine, while producing a proportional difference in the range of temperature of the latter, made no practical difference in capacity. (See Trials 5–8, Table I.)

No. 2.—The economy in coal consumption depends mainly upon both the suction pressures and condensing pressures. Maximum economy, with a given type of engine, where water must be bought at average city prices, is obtained at 28 lbs. suction pressure and about 150 lbs. condensing pressure. Under these conditions for a non-condensing steam-engine, consuming coal at the rate of 3 lbs. per hour per indicated horse-power of steam-cylinders, 24 lbs. of ice refrigerating effect are obtained per pound of coal consumed. For the same condensing pressure, and with 7 lbs. suction pressure, which affords temperatures of zero Fahr., the possible economy falls to about 14 lbs. of "refrigerating effect" per pound of coal consumed. The condensing pressure is determined by the amount of condensing water supplied to liquefy the ammonia in the condenser. If the latter is about one gallon per minute per ton of refrigerating effect per twenty-four hours,

GENERAL CONCLUSIONS.

By the term "ice-refrigerating effect" is meant the production of an amount of cold equivalent in heat units to the amount of heat which would be necessary to melt a certain weight of ice. In speaking of the "*capacity*" of the machine as being so many tons "ice-refrigerating effect," or "ice effect," or "refrigerating effect," or "ice-melting effect," we mean the production of an amount of cold in twenty-four hours equivalent in thermal units to the amount of heat necessary to melt so many tons of ice at 32° to water at 32°. Each ton so melted requires 284,000 British thermal units. Similarly, in speaking of the "*economy*" of the machine as so many pounds of ice-refrigerating effect per pound of coal, or ice effect per pound of coal, or refrigerating effect per pound of coal, we mean that a pound of coal consumed at the boiler for the purpose of generating steam to work the engine, produces an amount of cold equivalent in thermal units to the amount of heat necessary to melt so many pounds of ice at 32° Fahr. into water at 32° Fahr., each pound so melted requiring 142 British thermal units. The actual ice which a machine will make per pound of coal, or in twenty-four hours, is always less than the above amounts of refrigerating effect; first, because the water frozen is always at a higher temperature than 32° Fahr., and second, because of the total refrigerating effect produced in a bath of brine a considerable percentage is dissipated or wasted in manipulating the cans, of water to be frozen, in and about the freezing tank. Such waste amounts to from 20 to 50% of the refrigerating effect as defined above, varying according to circumstances not available to measurement. Hence, in stating the performance of a refrigerating machine, the actual pounds of ice which the machine might make cannot be used,

---

tion, plus .07 degrees that the temperature is raised by pouring the brine through the air into the larger vessel.

r_2 = temperature of brine in large vessel before that in the small vessel is poured into it [corrected for radiation].

r_3 = final temperature of brine in large vessel [corrected for radiation].

w = weight of brine in small vessel, and W = weight in large.

The hair felt covering about the cold brine vessel was one inch thick, and the area of contact was 230 square inches, weighing one and one-half pounds. The rate at which this felt was being cooled by the brine was about one degree per minute. The pouring and subsequent reading was so quickly done, that only half a minute of time elapsed. The possible heat abstracted from the felt was $1\frac{1}{2} \times 1 \times \frac{1}{2} = \frac{3}{4}$ thermal units if the specific heat was unity. The average heat units involved are 13,800, of which $\frac{3}{4}$ is less than one one-thousandth per cent.

and the above fictitious basis is substituted. The following conclusions are derived from the results of the investigation, and such deductions from the latter as will be found throughout the paper.

No. 1.—The capacity of the machine is proportional, almost entirely, to the weight of ammonia circulated. This weight depends on the suction pressure and the cubic displacement of the compressor pumps. The practical suction pressures range from 7 lbs. above the atmosphere, with which a temperature of zero Fahrenheit can be produced, to 28 lbs. above the atmosphere, with which the temperatures of refrigeration are confined to about 28° Fahr. At the lower pressure only about one-half as much weight of ammonia can be circulated as at the upper pressure, the proportion being about in accordance with the ratio of the absolute pressures, 22 and 42 lbs. respectively. For each cubic foot of piston displacement per minute a capacity of about one-sixth of a ton of "refrigerating effect" per 24 hours can be produced at the lower pressure, and of about one-third of a ton at the upper pressure. No other elements practically affect the capacity of a machine, provided the cooling surface in the brine tank or other space to be cooled is equal to about 36 square feet per ton of capacity at 28 lbs. back-pressure. For example, a difference of 100% in the rate of circulation of brine, while producing a proportional difference in the range of temperature of the latter, made no practical difference in capacity. (See Trials 5–8, Table I.)

No. 2.—The economy in coal consumption depends mainly upon both the suction pressures and condensing pressures. Maximum economy, with a given type of engine, where water must be bought at average city prices, is obtained at 28 lbs. suction pressure and about 150 lbs. condensing pressure. Under these conditions for a non-condensing steam-engine, consuming coal at the rate of 3 lbs. per hour per indicated horse-power of steam-cylinders, 24 lbs. of ice refrigerating effect are obtained per pound of coal consumed. For the same condensing pressure, and with 7 lbs. suction pressure, which affords temperatures of zero Fahr., the possible economy falls to about 14 lbs. of "refrigerating effect" per pound of coal consumed. The condensing pressure is determined by the amount of condensing water supplied to liquefy the ammonia in the condenser. If the latter is about one gallon per minute per ton of refrigerating effect per twenty-four hours, No. 3.—If water cost $1.00 per 1,000 cubic feet, and coal $4.00 per ton, the cost of each of these items per ton of ice produced is equal, being 17 cents, or 34 cents for both together, per ton of refrigerating effect for a suction pressure of 28 lbs., and any condensing pressure between 150 and 215 lbs. above the atmosphere. As the condensing pressure increases, the supply of water per ton is reduced; but it happens that the cost of such reduction is almost exactly compensated by the cost of the increased power to operate the machine. As the condensing pressure is reduced however, the cost of cooling water increases much more rapidly than the cost of power decreases, so that at about 100 lbs. condensing pressure the cost of water per ton of capacity is about 50 cents and the cost of power about 10 cents, making a total cost of about one and three-quarter times that for 150 lbs. condensing pressure. Similar ratios of cost obtain with lower suction pressures. Thus, for 7 lbs. suction pressure and 150 lbs. condensing pressure, the total cost per ton for both coal and water is 45 cents and this remains the same for higher condensing pressures, but increases about 50% when the condensing pressure becomes about 100 lbs. With coal at $4.00 per ton, condensing water must be obtained as cheap as 50 cents per 1,000 cubic feet in order to make the combined cost of coal and water at about 100 lbs. pressure equal to that for 150 lbs. or higher condensing pressure. These conclusions apply also to ammonia absorption machines. They will be found discussed at length under the head of "General Principles controlling the Costs of Coal and Water."

No. 4.—An important deduction from the measurements, by meter, of the quantity of ammonia circulated, compared with the weight of ammonia accounted for by the displacement of the compressors, is that the latter falls about 25% short. From this fact it would appear that a similar phenomenon to that of cylinder condensation in steam-engines takes place in the ammonia compression cylinder, due to the action of the metallic surface of the latter upon the gaseous ammonia, whereby the latter is superheated during its entrance so as to produce a rarefaction which reduces the weight of a cylinder full of the gas upward of 25% below that which corresponds to the density of the gas at the temperature at which it enters the compressor. In steam-engines such action *induces more* steam to enter the cylinder than corresponds to the volume swept through by the

ECONOMY DEPENDING ON COAL ALONE

Steam Engine.			Lbs. of Ice-melting Effect.								British Thermal Units per lb. of Steam.			
			150 lbs. Condensing Pressure.				105 lbs. Condensing Pressure.				150 lbs. Condensing Pressure.		105 lbs. Condensing Pressure.	
			28 lbs. Suction Pressure.		7 lbs. Suction Pressure.		Suction Pressure 28 lbs.		Suction Pressure 7 lbs.		Suction Pressure.		Suction Pressure.	
Type.	Coal per H.P.	Water per H.P.	Per lb. of Coal.	Per lb. of Steam.	Per lb. of Coal.	Per lb. of Steam.	Per lb. of Coal.	Per lb. of Steam.	Per lb. of Coal.	Per lb. of Steam.	28 lbs.	7 lbs.	28 lbs.	7 lbs.
Non-condensing	3	25	24	2.90	14.0	1.69	34.5	4.16	22	2.65	393	240	591	376
Non-compound condensing...	2.4	20	30	3.61	17.5	2.11	43.0	5.18	27.5	3.31	513	300	725*	470
Compound condensing......	1.9	16	37.5	4.51	21.5	2.58	54.0	6.50	34.5	4.16	640	366	923	591

The above figures are equivalent to assuming a boiler efficiency of 8.3 lbs. of water evaporated per pound of coal under working conditions.

* This corresponds most nearly with the conditions of the Linde machine, whose efficiency is given in Prof. Schröter's report as 305 calories of refrigeration per kilo of steam, or 505 thermal units per pound of steam. In this trial a condensing engine was used, consuming 20.2 lbs. of steam per horse-power. The condenser was placed in the wheelrace of a turbine in so superabundant a stream of water that the range in the condenser was too little to measure, thus affording a condensing pressure of 90 lbs. The theoretical performance possible under these conditions is 499 calories per kilo of steam. The actual performance was estimated from weighing the ice made, and is equivalent to only 61% of the theoretical figure.

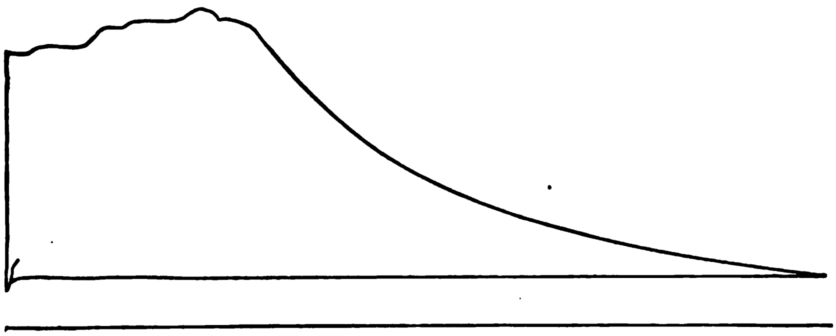


FIG. 122.—Average Ammonia Card. Trial No. 1. Scale, 100.

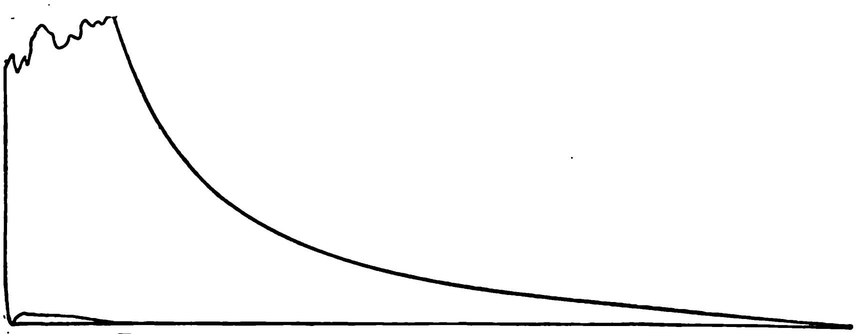


FIG. 123.—Average Ammonia Card. Trial No. 2. Scale, 100.

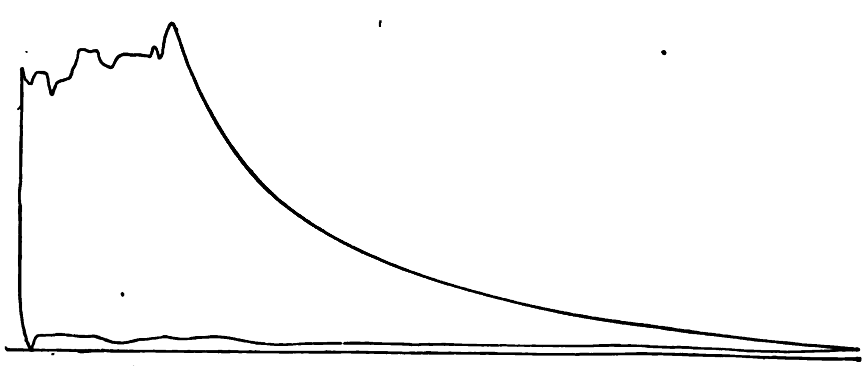


FIG. 124.—Average Ammonia Card. Trial No. 3. Scale, 100.

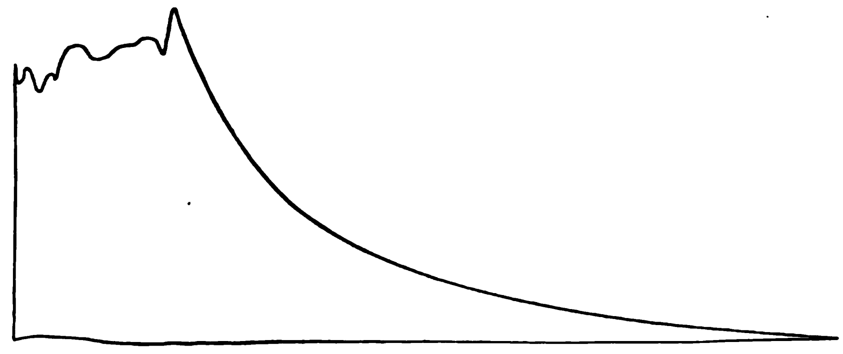


FIG. 125.—Average Ammonia Card. Trial No. 4. Scale, 100.

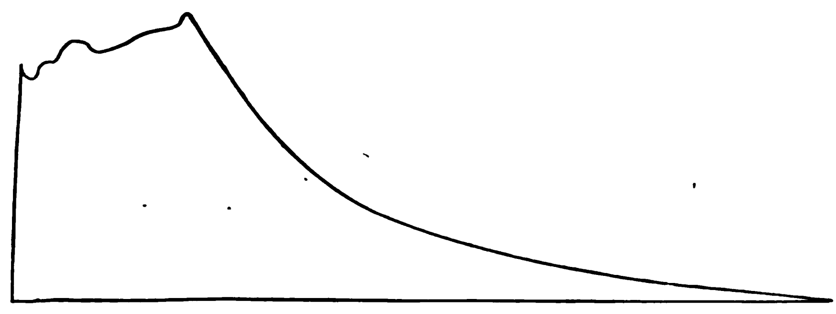


FIG. 126.—Average Ammonia Card. Trial No. 5. Scale, 100.

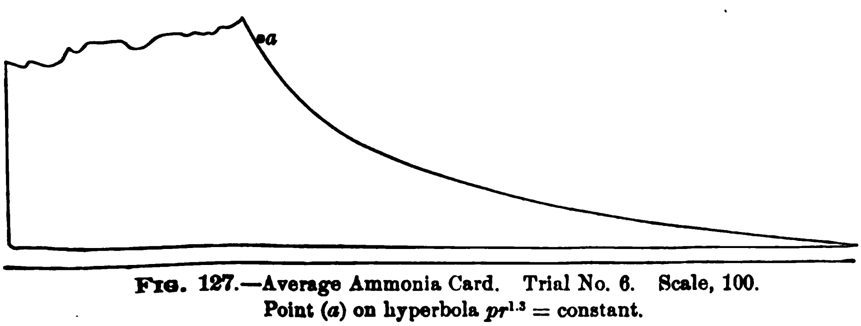


FIG. 127.—Average Ammonia Card. Trial No. 6. Scale, 100.
Point (a) on hyperbola $pv^{1.3}$ = constant.

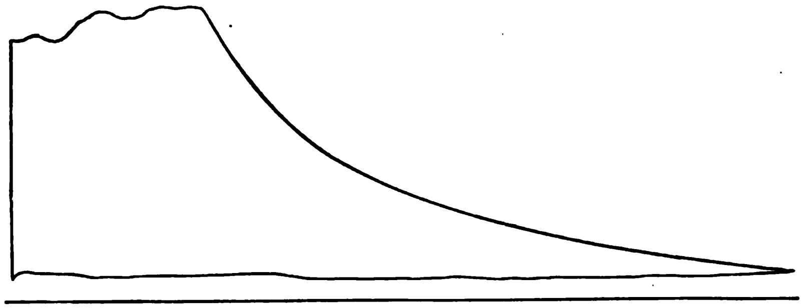


FIG. 128.—Average Ammonia Card. Trial No. 7. Scale, 100.

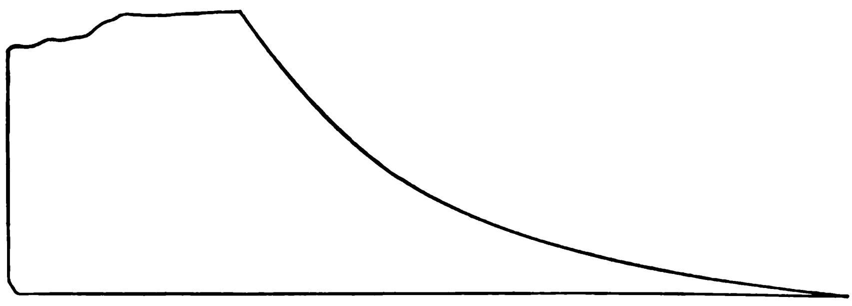


FIG. 129.—Special Ammonia Card. Trial No. 8. Scale, 100.

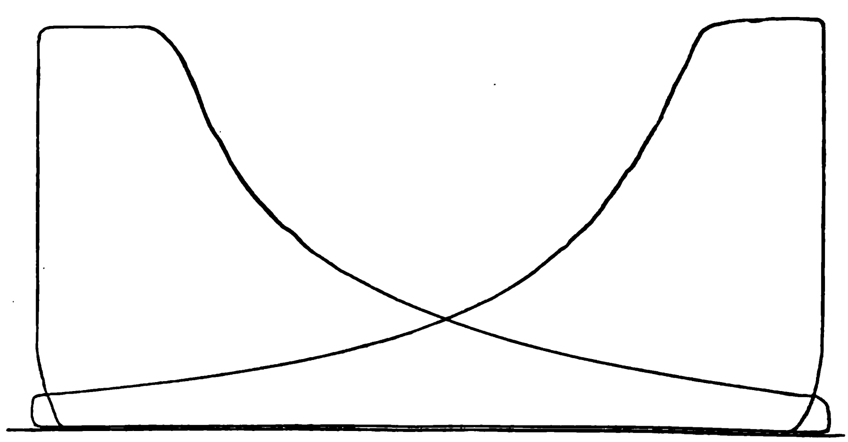


FIG. 130.—Average Steam Card. Trial No. 8. Scale, 40. M. E. P. 32.97.

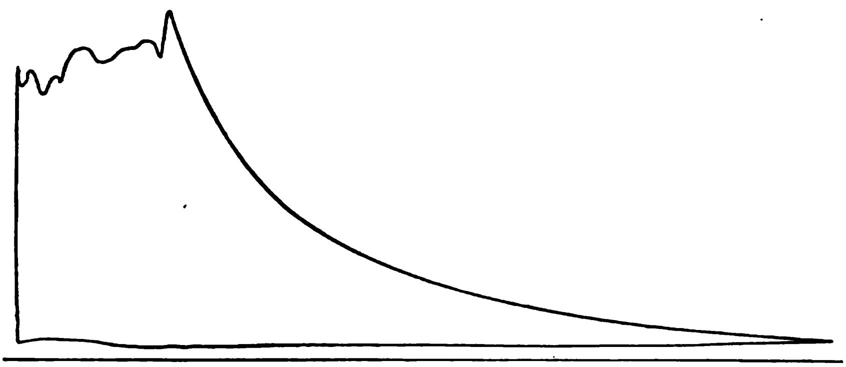


FIG. 125.—Average Ammonia Card. Trial No. 4. Scale, 100.

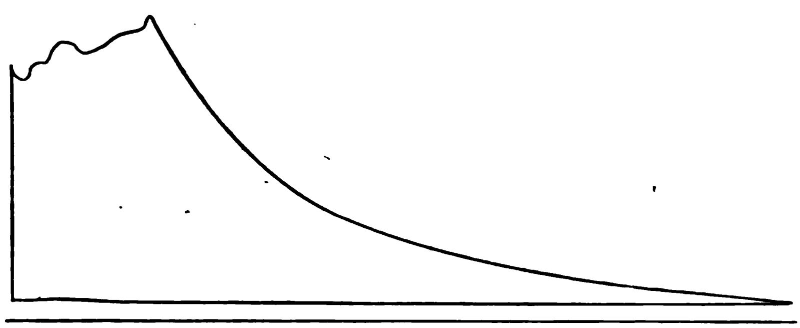


FIG. 126.—Average Ammonia Card. Trial No. 5. Scale, 100.

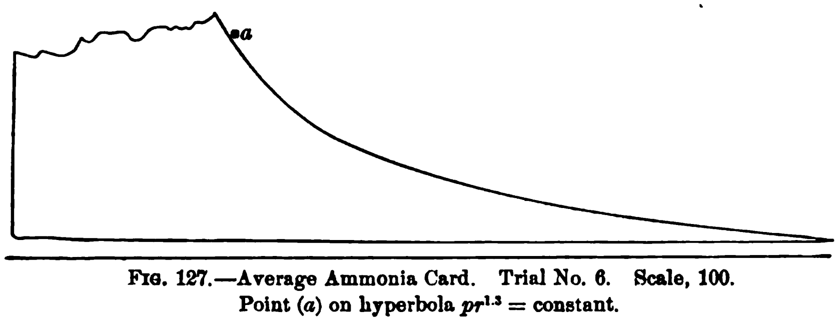


FIG. 127.—Average Ammonia Card. Trial No. 6. Scale, 100.
Point (*a*) on hyperbola $pv^{1.3}$ = constant.

NO. 2.

Clearance: 5/32.
No Oil.
Revolutions per Min.: 80.
Temp. Gas at Inlet: 48° F.
" " " Outlet: 174° "

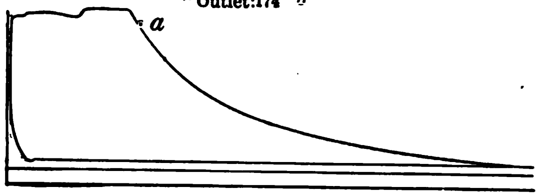


FIG. 132.

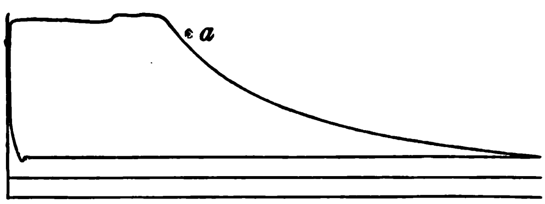


FIG. 132a.

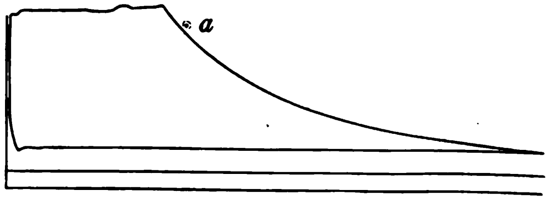
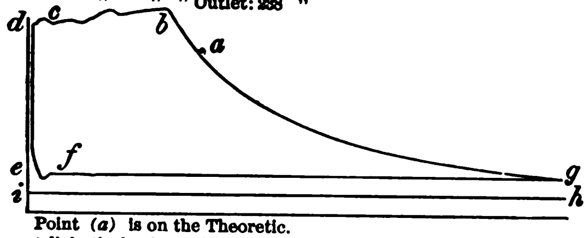


FIG. 132b.

NO. 3.

Clearance: 9/32.
No Oil. Revs. per Min.: 48.
Temp. Gas at Inlet: 84° F.
" " " Outlet: 238° "



Point (a) is on the Theoretic.
Adiabatic for Super-heated Ammonia Gas at 130 lbs. Absolute.
Line *ih* : Atmospheric Pressure.

FIG. 183.

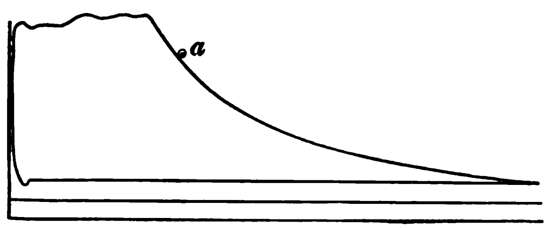


FIG. 183*a*.

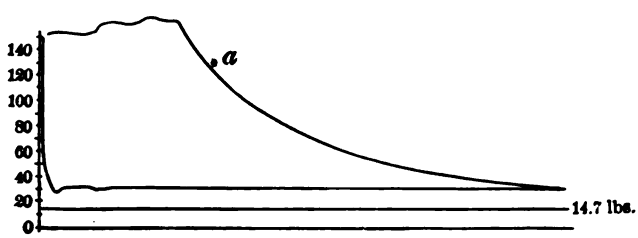


FIG. 183*b*.

NO. 4.

Clearance: $\frac{5}{32}$.
60 Drops of Oil per Min.
48 Revolutions " "
Temp. Gas at Inlet: 84° F.
" " " Outlet: 228° "

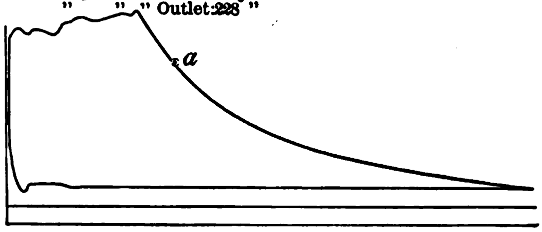
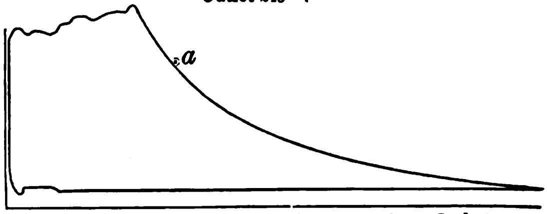


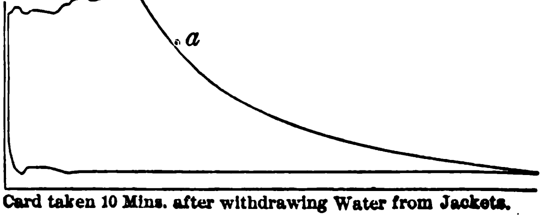
FIG. 134.

NO. 8.

Clearance: $\frac{5}{32}$:
50 Drops of Oil per Min,
Revolutions: 48.
Temp. of Gas at Inlet; 31° F.
" " " " Outlet 246° ".



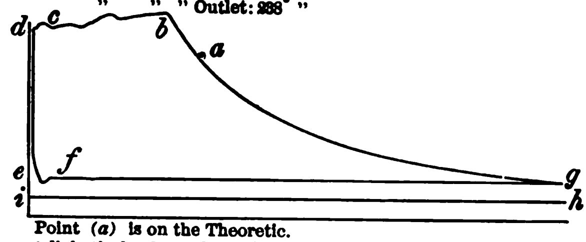
Card taken 5 Mins. after withdrawing Water from Jackets.
FIG. 134*a*.



Card taken 10 Mins. after withdrawing Water from Jackets.
FIG. 134*b*.

NO. 3.

Clearance: 5/32.
No Oil. Revs. per Min.: 48.
Temp. Gas at Inlet: 34° F.
" " " Outlet: 233° "



Point (*a*) is on the Theoretic.
Adiabatic for Super-heated Ammonia Gas at 130 lbs. Absolute.
Line *ih* : Atmospheric Pressure.

FIG. 183.

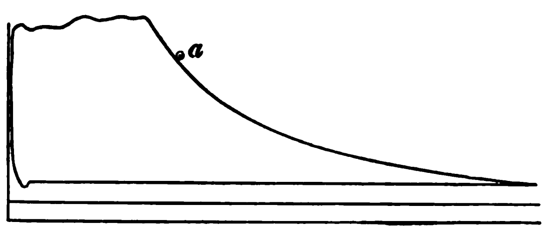


FIG. 183*a*.

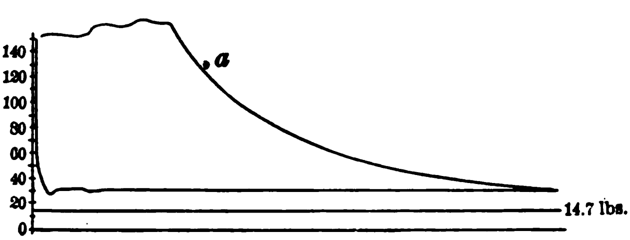


FIG. 183*b*.

NO. 6.

Clearance: 5/32.
Sight Glass Full of Oil.
Revolutions: 48.
Temp. Gas at Inlet: 32° F.
" " " Outlet: 233° "

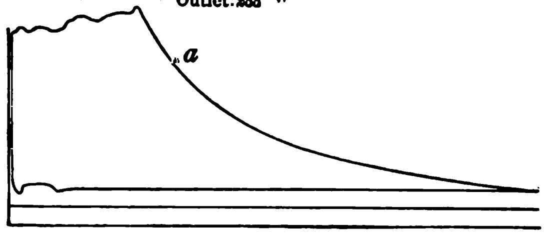


Fig. 136.

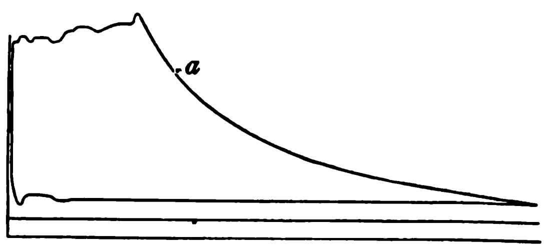


Fig. 136*a*.

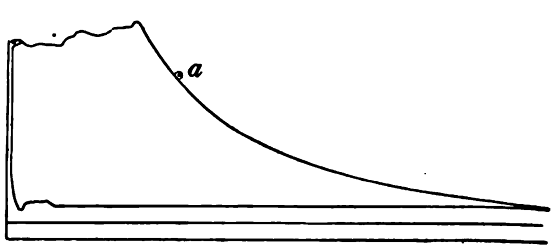


Fig. 136*b*.

NO. 5.

Clearance: $\frac{1}{32}$.
Continuous Stream of Oil.
Revs. per Min.: 48.
Temp. Gas at Inlet: 32° F.
" " " Outlet: 232° "

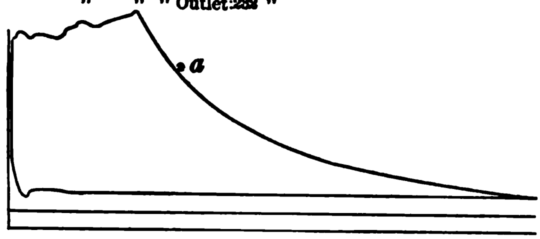


Fig. 135.

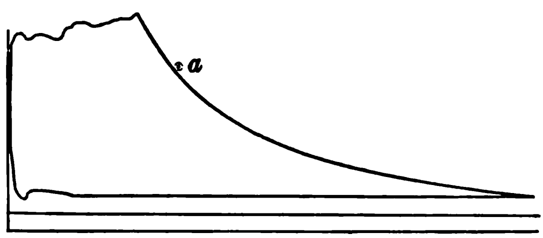


Fig. 135a.

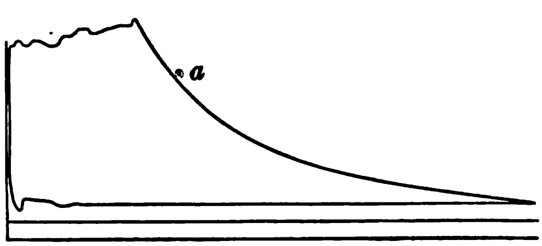
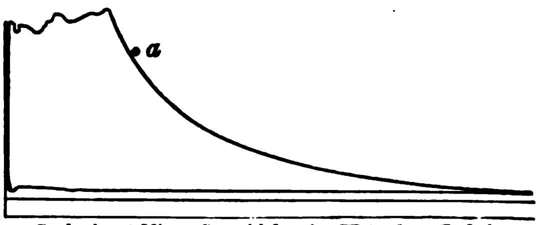


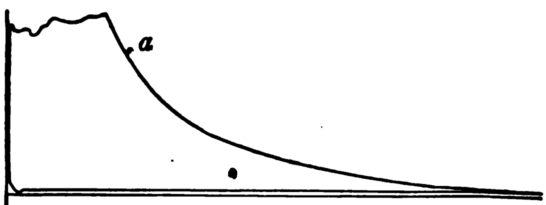
Fig. 135b.

NO. 8.

Clearance: [illegible].
50 Drops of Oil per Minute.
Revolutions: 48.
Temp. Gas at Inlet: 81° F.
" " " Outlet: 246° F.



Card taken 5 Mins. after withdrawing Water from Jacket.
FIG. 138.



Card taken 10 Mins. after withdrawing Water from Jacket.
FIG. 138*a*.

NO. 7.

Clearance: $\frac{5}{32}$.
50 Drops of Oil per Min.
Revolutions: 48.
Temp. of Gas at Inlet: 33° F.
" " " " Outlet 233° ".

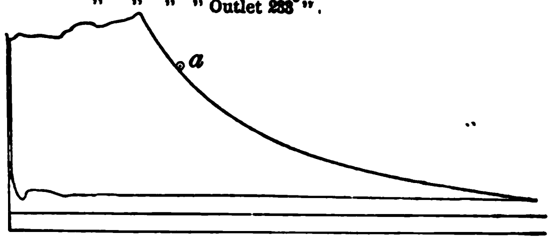


FIG. 187.

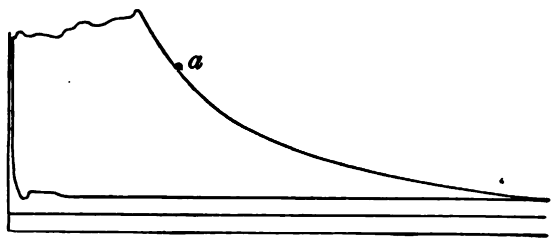


FIG. 187*a*.

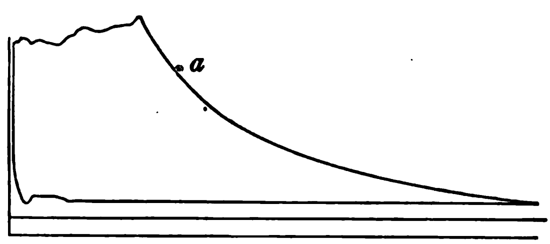


FIG. 187*b*.

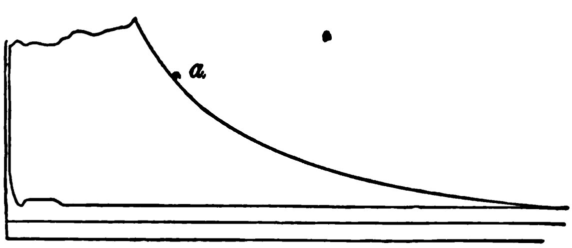


FIG. 187*c*.

NO. 10.

Clearance: $\frac{1}{32}$
No Oil. Revolutions: 80.
Temp. Gas at Inlet: $14\frac{1}{6}°$ F
" " " Outlet: 183° "

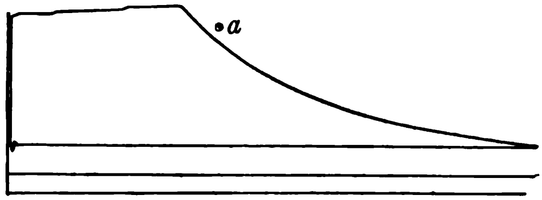


FIG. 140.

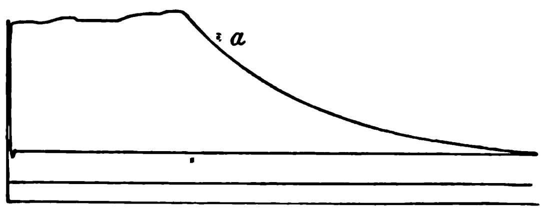


FIG. 140*a*.

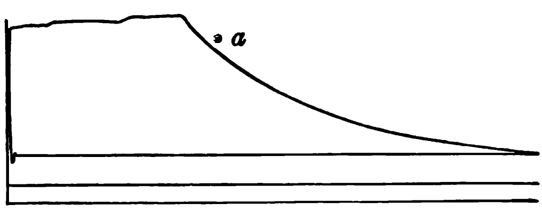


FIG. 140*b*.

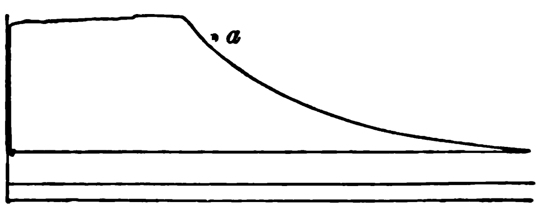
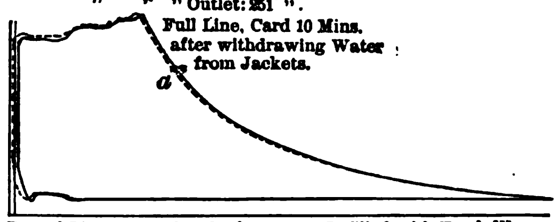


FIG. 140*c*.

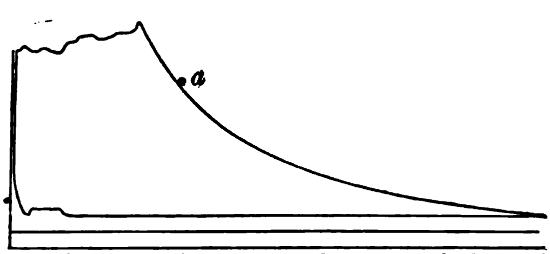
NO. 9.

Clearance: 5/32.
50 Drops of Oil per Min. Revs.: 48.
Temp. Gas at Inlet: 31° F.
" " " Outlet: 251° ".



Dotted Line; Just after Jackets were Refilled with Fresh Water at 60° F.

FIG. 139.



Card taken 7 Mins. after Refilling Jackets with Fresh Water at 60° F.

FIG. 139a.

NO. 10.

Clearance: 1/32
No Oil. Revolutions: 80.
Temp. Gas at Inlet: 14½° F
" " " Outlet: 188° "

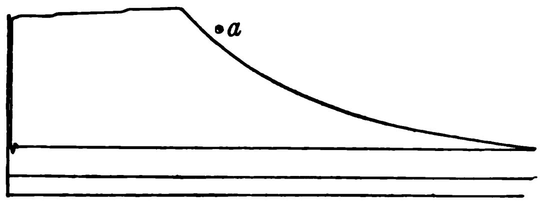


FIG. 140.

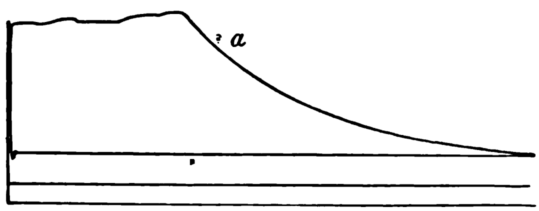


FIG. 140a.

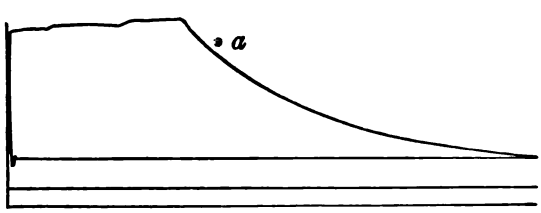


FIG. 140b.

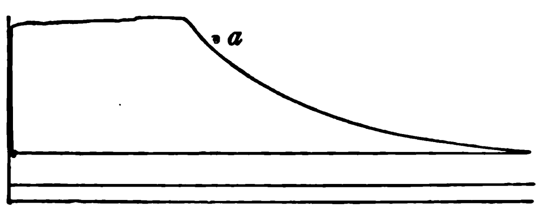


FIG. 140c.

CCCCXXXIII.*

SPECIAL EXPERIMENTS WITH LUBRICANTS.

BY J. E. DENTON, HOBOKEN, N. J.
(Member of the Society.)

THE object of the following paper is to make public the details of construction of two pieces† of apparatus for investigating the action of lubricants, under special conditions which attempt to approximate to those of practical service: *first*, in the lubrication of steam-cylinders; and, *second*, in the lubrication of journals subjected to heavy pressure.

The experiments which will be recited are offered not as completely exhaustive, but as suggesting lines of investigation which are so extensive in their opportunities as to make it desirable to secure as wide a discussion as possible regarding them, and to enlist as many other workers in the same field as possible.

DESCRIPTION OF APPARATUS FOR CYLINDER OILS.

Referring to Fig. 146, a piston X of a 6-inch steam-cylinder, 9-inch stroke, is fitted with a special packing ring C carried upon spring levers, so that the force of friction created between it and the walls of the cylinder shall move a pencil arm E, and, as the piston reciprocates, form an approximately rectangular diagram on a piece of paper A held against the pencil. One-half the width of the rectangular diagram measures the total frictional resistance of the ring on a scale of 230 lbs. to the inch. The packing ring C is made almost perfectly flexible by saw slits on its interior surface, and can be expanded by levers actuated by the screw F, so that any desired intensity of pressure up to 150 lbs. per square inch can be created between the ring and the surface of the cylinder. Two ordinary packing rings S pre-

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

† The apparatus described has been constructed for the Lubricating Committee of the Standard Oil Company. It is erected and in use at the Stevens Institute, pending the selection of a location for a laboratory elsewhere. I am indebted to the courtesy of this committee for the use of the apparatus to secure such data as are presented.

CCCCXXXIII.*

SPECIAL EXPERIMENTS WITH LUBRICANTS.

BY J. E. DENTON, HOBOKEN, N. J.
(Member of the Society.)

THE object of the following paper is to make public the details of construction of two pieces† of apparatus for investigating the action of lubricants, under special conditions which attempt to approximate to those of practical service: *first*, in the lubrication of steam-cylinders; and, *second*, in the lubrication of journals subjected to heavy pressure.

The experiments which will be recited are offered not as completely exhaustive, but as suggesting lines of investigation which are so extensive in their opportunities as to make it desirable to secure as wide a discussion as possible regarding them, and to enlist as many other workers in the same field as possible.

DESCRIPTION OF APPARATUS FOR CYLINDER OILS.

Referring to Fig. 146, a piston X of a 6-inch steam-cylinder, 9-inch stroke, is fitted with a special packing ring C carried upon spring levers, so that the force of friction created between it and the walls of the cylinder shall move a pencil arm E, and, as the piston reciprocates, form an approximately rectangular diagram on a piece of paper A held against the pencil. One-half the width of the rectangular diagram measures the total frictional resistance of the ring on a scale of 230 lbs. to the inch. The packing ring C is made almost perfectly flexible by saw slits on its interior surface, and can be expanded by levers actuated by the screw F, so that any desired intensity of pressure up to 150 lbs. per square inch can be created between the ring and the surface of the cylinder. Two ordinary packing rings S pre-

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

† The apparatus described has been constructed for the Lubricating Committee of the Standard Oil Company. It is erected and in use at the Stevens Institute, pending the selection of a location for a laboratory elsewhere. I am indebted to the courtesy of this committee for the use of the apparatus to secure such data as are presented.

vent the steam pressure from entering the space devoted to the special ring, so that by connecting this space with the atmosphere the only pressure upon the packing ring will be that due to the adjustment of the screw *F*. Provision is made, however, to create a pressure under the test ring by steam if desired. In this case a connection with the boiler is made at the stationary end of the trussed pipes, Fig. 148, one end of which vibrates with the piston and delivers steam into the hollow piston rod *B*, Fig. 146.

The sectional view, Fig. 147, shows the internal construction. The test ring is fastened to one end of the lever *D*, which has its support *O* upon the solid part of the piston. The other end of this lever is fastened to the rod *E*. The latter communicates with a crank *G* actuating the pencil *J*. There are four such levers situated at the quadrants of the circle. The resistance opposed to the friction on the test ring is that due to the joint torsional distortion of the three rectangular strips of steel which form respectively the connection between the lever and the test ring, the rod *E*, and fulcrum *O*. The mass of the moving parts is by the design so distributed that their centre of gravity lies exactly in the fulcrum *O*. Hence no movement of the pencil can result from forces due to accelerations depending upon the speed of running the engine. To calibrate the torsional springs the test ring is contracted by the screw *F*, Fig. 146, so that it is withdrawn from contact with the walls of the cylinder. A plug directly above the top of the rod *E*, Fig. 147, is removed, and weights applied to the top of this rod. The movement of the pencil, corresponding to various weights, thus becomes known by direct experiments, which can be made while the apparatus is in a steam cylinder and at the temperature corresponding to the steam. To calibrate the test ring for radial pressure the screw *F*, Fig. 146, is tightened, so that the resistance of the ring against the walls of the cylinder is just sensible to the effort of the fingers. A wedge is then fitted to the space where the test ring is parted The piston is then removed from the cylinder and supported horizontally. A flexible wire is wound around the test ring, making one complete turn. One end is fastened to an external support and the other end successively weighted with increasing amounts. For each weight applied, the screw *F*, Fig. 146, is turned so as to expand the ring until the wedge in the parting space can be moved with the fingers. Twice the

the groaning will occur during the first few turns, and disappear as the engine reaches speed. Occasionally, with all lubricants, the groaning will be heard for possibly half an hour, while the rate of feeding is below that necessary to afford a friction much less than when no oil is used, and then the noise will disappear permanently, as the oil supply increases beyond this amount. The diagram never fails to indicate the slightest "groaning," even when the latter is inaudible, without special attention on the part of the operator. In several months of use not the faintest roughing or cutting of the surface has occurred from these occasional groaning periods, both the surface of the test ring and that of the cylinder possessing a very high polish, which gradually improves.

The following table shows the results of comparing three petroleum lubricants of good reputation for cylinder lubrication with pure melted tallow. All the petroleum lubricants contain some animal mixture to a very limited extent, not more than about 5%, and this ingredient is different in each.

The general appearance and nature of all three is, however, about the same. They are defined according to modern methods of distinction as oils of about 550° flash point, 26° gravity Baumé, 45° cold test, and 135 units of viscosity at 212° Fahr.

The tallow, when melted, has practically the same gravity and flash point, but its viscosity at 212° is only 55 units. That is, it is about 2½ times more fluid than the petroleum oils. Tallow, during its extensive use, in the past, as an almost universal lubricant for steam cylinders, has acquired the reputation of affording better lubrication than any product of petroleum moderately mixed with some animal ingredient. The acid effects of tallow, causing it rapidly to corrode most kinds of cast iron, render it less desirable on the whole than the petroleum cylinder lubricants. But so far as the lubrication or anti-friction producing properties are concerned, it is held by many that tallow is superior to petroleum lubricants.

The results in the table presented apparently prove quite the opposite. The amount of tallow shown to be required to produce a coefficient of 1.5% under about locomotive conditions of steam pressure is about twice the required quantity of the petroleum oils. The figures for surface uncovered per pint, Col. 6, indicate that the rate of feeding is about that common in engine practice where oil is used with medium economy.

It may be noticed that the experiments commence on September 20th, with tallow, and, after several weeks of experimenting with the other oils, tallow is again put through the apparatus with practically the same results, thus proving that the rubbing surfaces have not altered their condition. Between each test the apparatus was calibrated, and found unchanged in its scales. After each experiment with the tallow, an accumulation of black paste covered all parts of the interior of the engine to quite a tangible thickness; while, with the petroleum oils, only an oily film of the same consistency of the oils themselves was found on these surfaces.

While the subject is too little exhausted to venture upon final conclusions, the results suggest that the greater viscosity of the petroleum oils, under steam heat may be responsible for the economy shown as compared to the tallow, as this element certainly controls the rate of consumption in the case of external bearings, and must have a similar influence between the rubbing surfaces of the interior of steam-engines, provided the more viscous oil can precipitate itself out of the steam as readily as do the more fluid animal lubricants, like tallow or lard, etc.

The experiments afford no evidence of such superior powers of precipitation on the part of tallow, however.

Analysis of deposit made by tallow:

Animal oil	28.2
Carbon	11.1
Oxide of iron	4.2
Metallic iron	53.3
Lime	0.5
Water	0.9
Undetermined	1.8

COMPARISON OF OILS ON CYLINDER OIL TESTER.

1	2*	3	4	5	6	7	8	9	10	11	12	
DATE.	Time of Running. Hours.	Drops per minute.	Hours to consume a pint.	Drops per pint.	Sq. ft. of surface uncovered by piston per pint of oil.*	Width of Card.	Lubricant.	Steam press.	Revs. per minute.	Total friction in lbs. of test ring.	Coef. of friction.	REMARKS.
September 20	3	2				0.5	Tallow.	70	130	57.50	0.085	
September 20	2	5				0.6	"	128	145	69.00	0.102	
September 22	1	8	24	11520	432000	0.7	"	130	140	80.50	0.119	
September 22	2.75	12	16	11520	288000	0.7	"	130	140	80.50	0.119	
September 22	3.75	24	8	11520	144000	0.11	"	130	140	12.65	0.018	Friction fell from $\frac{1}{2}$ to $\frac{1}{8}$ in 1.30 hours.
September 23	5.5	18	10	10800	180000	0.11	"	126	136	12.65	0.018	Black paste on rings and cylinder.
September 23	2.5	2				0.63	No. A.	130	138	72.45	0.107	
September 25	1.5	5	16	4800	288000	0.10	"	130	138	11.50	0.017	Surfaces rather dry.
September 25	3.75	8	10	4860	180000	0.06	"	130	127	6.90	0.010	
September 26	2.75	6	13	4680	234000	0.10	"	130	127	11.50	0.017	
September 27	2.75	6	13	4680	234000	0.08	"	130	127	9.20	0.013	Surplus oil on heads. No deposit.
September 29	4.0	4				0.18	No. B.	130	127	20.70	0.031	Groaning with 2 drops. None, 4 drops.
September 30	3.0	4	24	5760	432000	0.13	"	130	127	14.95	0.022	
September 30	3.0	5	19	5700	342000	0.10	"	130	127	11.50	0.017	Surplus on surfaces.
October 1	3.75	6	16	5760	288000	0.10	"	130	127	11.50	0.017	
October 1	3.0	6	16	5760	288000	0.10	"	130	127	11.50	0.017	Surplus on surfaces.
October 3	2.75	4				0.28	No. A.	135	130	32.20	0.047	Groaning for 40 minutes.
October 3	3.0	6	16	5760	288000	0.10	"	135	130	11.50	0.017	
October 4	2.0	6	14	5040	252000	0.10	"	135	138	11.50	0.017	
October 4	2.0	5	18	5400	324000	0.13	"	135	138	14.95	0.022	No deposit.
October 7	1.5	2				0.30	No. C.	130	130	34.50	0.051	
October 8	3.75	4	24	5760	432000	0.11	"	130	130	12.65	0.018	Surplus of oil.
October 9	3.25	4	24	5760	432000	0.11	"	130	130	12.65	0.018	
October 10	3.75	4	24	5760	432000	0.10	"	130	130	11.50	0.017	Surplus oil on surfaces. No deposit.
October 11	2.50	4				0.10	"	130	130	11.50	0.017	Surplus oil on surfaces. No deposit.
October 14	3.0	5	32	9600	576000	0.6	Tallow.	128	126	69.00	0.102	
October 15	3.50	8	20	9600	360000	0.33	"	135	134	37.95	0.056	
October 16	3.50	12	12	8640	216000	0.10	"	135	134	11.50	0.017	
October 17	2.0	12	12	8640	216000	0.10	"	135	134	11.50	0.017	Black grit $\frac{1}{16}$ thick.

* A locomotive using 1 pint per 50 miles uncovers about 500,000 square feet. Where oil is used liberally, 100,000 square feet corresponds to a pint. The above figures are midway of these.

DESCRIPTION OF APPARATUS FOR HEAVY BEARING PRESSURES.

An ordinary railroad car axle *D*, Fig. 150, has forced upon it a 12-inch wheel *A*, as is the ordinary car wheel. The wheel *A* supports one end of the axle on the friction rollers *B*, 33 inches diameter. The other end of the axle is carried in an ordinary bearing on brick work and can be coupled positively to the main shaft of a 25 H.P. steam-engine *C*. The axle can also be driven by a belt, in the direction opposite to that of the engine, for purposes of calibration, to be explained below. The test journal is at *E*. It is pressed upon by the brass, which may be of special form, as per Fig. 151, so as to fit the block *F*, Fig. 150; or it may be the ordinary master car-builder's brass, fitting the regular form of railway car box, which may be substituted in place of the block *F*. The latter is drawn toward the journal by a jack-screw *K*, connected by the linkage shown in dotted lines to the steelyard *M*, which is anchored to the beam *N*, lodged in the brick-work *O*. The screw *K* has for its nut the cross-head *H*, having knife edges *J*, from which the links *G* extend upward to connect with the yokes in which the block *F* is confined. The steelyard has its counterpoise weights, *S* to S_4, suspended upon two rods S_5, S_6, so that when these are lowered by a screw in the hand-wheel S_9, the weights come upon the steelyard in succession. Thus: when the button S_8 is at 5,000 of its scale, the weight *S* only depends from the steelyard. If then the jack-screw *K* is made to lift upon the steelyard when the index pointer *R* is at its zero, a load of 5,000 lbs. is upon the test journal *E*. The other weights, in a similar manner, provide for 20,000 lbs. of load upon the test journal. A reciprocating motion is given to the brass by a worm gear *Z*, which moves the slide *a* back and forth at one-fifth the rate at which the test journal revolves. The movement of *a* can be varied by an eccentric cam from nothing to one-half an inch in each direction. A fan draws air from the atmosphere outside the building, which is available to create a breeze against the test journal having a velocity of about 30 miles per hour. The friction is weighed upon the double beam *h*, the smaller counterpoise reading pounds of friction at the surface of the journal, which is 3.75 inches diameter, and the larger weight reads to 10 lbs. The two weights combined weigh 350 lbs. of friction. Greater friction than this is provided for by weights hung upon the bob

A, which is suspended in a dash-pot of oil. In order exactly to subdivide the load on the journal so that precisely equal amounts of strains will act through each of the links *G*, the knife edges *J* are provided with lugs *I*, fastened with a set screw in a slot as shown. By means of these the knife edges can be slightly rotated nearer to or farther from the centre line of the test axle, and thereby adjust the relative strain in the two links *G*. It is impossible to construct machine work so that the knife edges *J* will be so accurately located, by measurement, as to equalize the strain in the links *G*. If the load upon the journal is only 5,000 lbs., an eccentricity of the test brass of one-half of one-hundredth of an inch will make the lever *h* read in error about 12 lbs., which is nearly twice the friction to be measured when the latter is near the minimum attainable with smooth bearings. Hence, all that can be accomplished with the lugs *I* is to adjust them when the lubrication is perfect and the pressures light, so that opposite directions of running give equal amounts of friction, and then determine the friction for greater loads by taking it equal to half the sum of the friction shown by running in opposite directions. When the coefficient of friction is to be determined under conditions giving minimum values of the latter, it is necessary to eliminate the friction between the test journal and any oiling pad which may rub against its under side in order to supply a lubricant. The friction of such a pad is found to be about 12 lbs., which is upward of the entire friction between the brass and the journal with 5,000 lbs. of load. To eliminate the pad friction, the pan *b*, which contains the latter, together with the oil to be tested, is supported free of the frame-work around the block *F* by a support like *c*. The rods *p* support a bar which carries a counterweight *Q*, by means of which the sensitiveness of the friction lever *h* is controlled. The apparatus does not differ in general plan from oil-testing machines now in use by others, such as the Tower machine, the machine in the United States Navy Yard in Brooklyn, and Goodman's and Stroudsley's apparatus. These machines use dead weight, however, to produce the load, instead of a steelyard. The latter is not, on the whole, as accurate as dead weights, but it permits the measurement of variations of friction, equal to a fraction of a pound, to be accurately made, which answers all purposes at present in view. The remaining features of difference are, *First*, the avoidance of the support of the test axle motion fixed at one quarter cut-off, no governor is needed and no manipulation of the throttle necessary. The boiler pressure is maintained between 125 and 135 lbs. above the atmosphere, and the speed at from 125 to 135 revolutions per minute. The oil to be tested is fed through a "Swift" sight feed lubricator *D*, Fig. 148, which is attached to the steam supply pipe in the usual manner. The cylinder is fitted with a bushing *B*, Fig. 148, which is removable. In commencing a test of oil the piston and bushing are removed, and submerged in a tank of naptha for upwards of twelve hours. They are then replaced without any oil upon them, and the engine run to speed. Oil is then fed at two drops per minute, and a pressure of 75 lbs. per square inch put upon the ring by the screw *F*, Fig. 146. Diagrams are then taken, and are generally about half an inch wide, and fairly rectangular. The rate of feeding is then gradually increased, each rate of feed prevailing for not less than three hours, until the width of the diagram is about three thirty-seconds of an inch. This corresponds to a coefficient of friction of about 1.5%. This rate of feeding is then maintained for two or three successive half-day runs of the engine, which is generally allowed to stand, with the full pressure upon the test ring, from noon of one day until about 7.30 A. M. of the following day. The test of an oil consists in determining the quantity required to be fed in order to steadily maintain a diagram three thirty-seconds of an inch wide. After the engine has been run about half an hour under any particular feed, the diagram is generally of fixed dimensions, and when the lubrication is satisfactory it is about identical with the one at the middle of Fig. 149. When this diagram is being produced with any oil there is no "groaning sound" emitted from the cylinder. Whenever there is the least sound resembling what engineers call "groaning," the pencil vibrates violently. Generally the groaning is omitted, only during the commencement of the stroke. Then the card shows the irregularities at the end, exhibited by the left diagram, Fig. 149. When steam is shut off and the engine gradually slows to a stop, vibrations occur during the last few turns, which give a diagram like that on the right of the figure, and the groaning sound accompanies them. A study of the "groaning" phenomenon indicates that it arises from a disturbance of a uniform condition of the film of oil upon the rubbing surfaces, and does not depend on the nature of the lubricant. In starting the engine in the morning, start, be about 6%, and remain at this figure long enough to cause several hundred degrees of temperature, and finally cut and scar the brass, as per Figs. 152 and 153. In this case, if the apparatus is continuously run for an hour, the journal may be at a *high red heat.* Such was actually the case with the brass in Fig. 152.

2d. The coefficient of friction may, at the start, be about 6%, and then may rapidly decrease, as shown in the following table, the brass being reciprocated five-sixteenths of an inch each way:

Duration of trial. Hours.	Temperature of brass.	Total load on journal. Lbs.	Area of contact. Sq. inches.	Pressure per square inch.	Coefficient of friction.
¼	75–130	5,000	1	5,000	6.0%
½	130–128	"	....	....	3.48
1	128–126	"	....	....	2.54
1½	126–95	"	....	....	0.84
2	126–95	"	....	....	0.36
1½	95–92	"	3.0	1,666	0.32

The area of bearing will have become a band about one-half an inch wide, as in Fig. 154. Further service slowly widens this band, and if a reciprocating motion is preserved, the surface of the brass becomes highly polished and is devoid of all rings or circular lines. With such surfaces any oil will lubricate, giving coefficients of friction of less than half of one per cent., and exhibiting only such differences of friction as are due to differences of viscosity. This is illustrated by the following table:

TABLE II.

FRICTION OF OILS FED BY PAD ON POLISHED SURFACES.

Each experiment of two or more hours' duration.

Character of Oil.	Total Load, 5,000 lbs. Press. per sq. inch, 1,000 lbs.			
	170 Revolutions.		320 Revolutions.	
	Temp. Fahr.	Coeff. of friction	Temp. Fahr.	Coeff. of friction
Pure sperm oil	75–91	0.14%	75–101	0.14
Paraffine oil	74–95	0.15	74–104	0.15
Pure petroleum cylinder stock, 550 flash	76–88	0.23	....	....

Character of Oil.	Total Load, 10,000 lbs. Press. per sq. inch, 1,000 lbs.			
	170 Revolutions.		320 Revolutions.	
	Temp. Fahr.	Coeff. of friction	Temp. Fahr.	Coeff. of friction
Pure sperm oil	76–100	0.11%	78–112	0.11%
Paraffine oil	88–106	0.11	106–113	0.11
Pure petroleum cylinder stock, 550 flash	76–97	0.18	....	....

It may be noticed that the experiments commence on September 20th, with tallow, and, after several weeks of experimenting with the other oils, tallow is again put through the apparatus with practically the same results, thus proving that the rubbing surfaces have not altered their condition. Between each test the apparatus was calibrated, and found unchanged in its scales. After each experiment with the tallow, an accumulation of black paste covered all parts of the interior of the engine to quite a tangible thickness; while, with the petroleum oils, only an oily film of the same consistency of the oils themselves was found on these surfaces.

While the subject is too little exhausted to venture upon final conclusions, the results suggest that the greater viscosity of the petroleum oils, under steam heat may be responsible for the economy shown as compared to the tallow, as this element certainly controls the rate of consumption in the case of external bearings, and must have a similar influence between the rubbing surfaces of the interior of steam-engines, provided the more viscous oil can precipitate itself out of the steam as readily as do the more fluid animal lubricants, like tallow or lard, etc.

The experiments afford no evidence of such superior powers of precipitation on the part of tallow, however.

Analysis of deposit made by tallow:

Animal oil	28.2
Carbon	11.1
Oxide of iron	4.2
Metallic iron	53.3
Lime	0.5
Water	0.9
Undetermined	1.8
	100.0

sperm or paraffine of one-sixth of one per cent., was planed out so that the five-eighths of an inch of width in bearing stand about one-sixteenth of an inch above the surrounding surfaces.

Fig. 157 represents the brass as thus prepared.

A load of 5,000 lbs. applied, and the machine run at 170 revolutions per minute, with paraffine oil applied with a pad.

The record was as follows :

Duration. Hours.	Temperature. Fahr.	Coefficient of Friction.
⅛	88°	5.5
¼	98	3.5
½	114	2.0
2	116	2.0
3	96	1.5
3½	96	1.25
3¾	96	1.10
4	95	0.60
4½	93	0.80
5	92	0.35
5½	92	0.35

Evidently the removal of the supporting effect of the metal planed out, allowed the bearing surface to alter, so that the friction was greatly increased at first. But from the final reduction of friction it is clear that the area of three square inches was quite sufficient to carry the load, and that the pressure per square inch, 1660 lbs., was not too great for the paraffine oil.

FRICTION OF LEAD-LINED BRASS.

The friction of concentric brasses like Fig. 151, supplied with the well-known "Hopkins lead lining," was found to be as follows:

Duration of Trial. Hours.	Revs. per minute.	Load in Pounds.	Temp. Fahr.	Oil.	Area of Contact.	Coef. of Friction.
⅓	170	5000	68–72°	Paraffine.		0.25%
½	170	5000	72–76	"		0.2%
1	170	5000	76–81	"		0%
2	170	5000	81–88	"		0%
2½	170	5000	88–90	"		0%
3	170	5000	90-90	"		0%
	Changed load to 10,000 lbs. on same day.					
1	170	10000	79–90	Paraffine.		0.18%
2	170	10000	90–94	"		0.15%
3	170	10000	94–98	"		0.12½%
5	170	10000	98–96	"	14	0.09%

Duration of Trial. Hours.	Revs. per minute.	Load in Pounds.	Temp. Fahr.	Oil.	Area of Contact.	Coef. of Friction.
	After standing two days.					
$\frac{1}{2}$	170	5000	72–80	Paraffine.	14	0.20%
1	170	5000	80–82	"	14	0.20%
2	170	5000	82–88	"	14	0.20%
	Cooling Compound applied, friction immediately rose for an instant to 0.50%, and then fell to 0.26%, as follows:					
2^{1}	170	5000	88–90	Paraffine.	14	0.26%
$2\frac{1}{2}$	170	5000	88–90	"	14	0.26%
3	170	5000	88–90	"	14	0.26%

With concentric brasses the lead does not "flow" under pressure to any perceptible extent. But under the irregularly distributed pressure of a M. C. B. brass there is a very considerable "flow." See Fig. 158.

FRICTIONAL EFFECT OF APPLYING A COOLING COMPOUND ON BARE BRASSES.

The cooling compound used consisted of about 60% of powdered soapstone mixed with 40% of grease. Between the fingers it has a sharp, gritty feeling, but contains no lumps.

The following are instances of its effect applied to paraffine oil running steadily with a coefficient of 0.25% on a bare brass:

Duration of Trial. Hours.	Revs. per min.	Load, Pounds.	Temp. Fahr.	Oil.	Coef. of Fric.
1	170	5000	86–95°	Paraffine.	0.25%
2	170	5000	95–97	"	0.25%
	Applied spoonful of cooling compound to Pad.				
	Friction increased instantly to.				1.1%
$2\frac{1}{30}$	170	5000	97–100	Paraffine.	0.9%
$2\frac{1}{20}$	170	5000	97–100	"	0.5%
$2\frac{1}{12}$	170	5000	97–100	"	0.4%
$2\frac{1}{4}$	170	5000	97–100	"	0.3%
3	170	5000	97–100	"	0.3%
$3\frac{1}{2}$	170	5000	100–98	"	0.3%

At the point where the cooling compound was applied the surface of the brass and journal showed a band of fine parallel incisions about one one-hundredth of an inch deep, but there was no appearance of a high temperature having been produced, as will be seen to be the case when a grain of emery cut its way across the bearings.

A, which is suspended in a dash-pot of oil. In order exactly to subdivide the load on the journal so that precisely equal amounts of strains will act through each of the links *G*, the knife edges *J* are provided with lugs *I*, fastened with a set screw in a slot as shown. By means of these the knife edges can be slightly rotated nearer to or farther from the centre line of the test axle, and thereby adjust the relative strain in the two links *G*. It is impossible to construct machine work so that the knife edges *J* will be so accurately located, by measurement, as to equalize the strain in the links *G*. If the load upon the journal is only 5,000 lbs., an eccentricity of the test brass of one-half of one-hundredth of an inch will make the lever *h* read in error about 12 lbs., which is nearly twice the friction to be measured when the latter is near the minimum attainable with smooth bearings. Hence, all that can be accomplished with the lugs *I* is to adjust them when the lubrication is perfect and the pressures light, so that opposite directions of running give equal amounts of friction, and then determine the friction for greater loads by taking it equal to half the sum of the friction shown by running in opposite directions. When the coefficient of friction is to be determined under conditions giving minimum values of the latter, it is necessary to eliminate the friction between the test journal and any oiling pad which may rub against its under side in order to supply a lubricant. The friction of such a pad is found to be about 12 lbs., which is upward of the entire friction between the brass and the journal with 5,000 lbs. of load. To eliminate the pad friction, the pan *b*, which contains the latter, together with the oil to be tested, is supported free of the frame-work around the block *F* by a support like *c*. The rods *p* support a bar which carries a counterweight *Q*, by means of which the sensitiveness of the friction lever *h* is controlled. The apparatus does not differ in general plan from oil-testing machines now in use by others, such as the Tower machine, the machine in the United States Navy Yard in Brooklyn, and Goodman's and Stroudsley's apparatus. These machines use dead weight, however, to produce the load, instead of a steelyard. The latter is not, on the whole, as accurate as dead weights, but it permits the measurement of variations of friction, equal to a fraction of a pound, to be accurately made, which answers all purposes at present in view. The remaining features of difference are, *First*, the avoidance of the support of the test axle at *A* in an ordinary bearing, whose friction would affect the temperature of the thermometer inserted in the test brass so that this temperature cannot be used as an index of the friction of the test brass, as it must be in using ordinary car brasses, which do not permit the bearing of the test brass to be sufficiently concentric for the accurate use of the friction lever *h*. Supporting the axle on the rollers *B* also enables a violent jar to be created by the irregular periphery of the wheels, as is the case in railroad practice, and this constitutes a feature which is ascribable to the fact that no measurable difference of friction is found when the direction of the machine is reversed, as was the case in the experiments of Mr. Tower and with the machine at the United States Navy Yard.

Second. A positive power connection is made with an engine of sufficient capacity to drive the apparatus to any speed when the test bearings are creating an abnormal amount of friction.

Third. The current of air provided to maintain low temperatures copies practical conditions a little more closely than does the use of cored brasses with water circulation.

COEFFICIENTS OF FRICTION WITH SPECIAL BRASSES.

Brasses like Fig. 151 are obtained from the Hopkins Car Brass Foundry, Jersey City, and bored with a bar, while held firmly in the block *F*, to which they have been previously fitted.

The brasses are then sent back to the foundry, and the concave bearing surface dressed with an emery wheel so as to give the rough surface shown in the figures, but the curvature of the surface is not altered. The object of dressing with the emery wheel is to enable the friction of the brass to be studied under the conditions of surface common to a large division of railroad car service using bare brasses, obtained from the Hopkins Foundry, and dressed in the above manner. When such a brass is loaded with 5,000 lbs. it may bear only over an area of about an inch.

Fig. 151 shows such a case. The two spots show the bearing after the axle had been turned a few times under 5,000 lbs. load. With a bearing surface of this character aggregating about one square inch, making the pressure per square inch fully 5,000 lbs., it appears that with any oil applied with a pad either of the following things may transpire :

1st. The coefficient of friction at 170 revolutions may, at the

BEHAVIOR OF SIX M. C. B. BRASSES UNDER CAR SERVICE CONDITIONS.

These brasses were obtained from the Hopkins Car Brass Foundry, without lead lining, and were dressed upon their rubbing surfaces with an emery wheel, as explained above. They were of hexagonal form on their upper surface, which roughly fitted the ordinary rough cast-iron saddle, inserted between the brass and the inner side of the top of the ordinary railway car box, which in these experiments took the place of the pressure block *K'*, Fig. 150.

As explained in connection with the description of the machine, an eccentricity of bearing in the brass of 0.005 inches defeats any attempt to measure friction directly in any apparatus in which the pressure is all applied to the top brass.

Hence, with so rough and indeterminate a line of contact as exists between a car box and the hexagonal brass, it is not feasible to determine the friction except by observation of the temperatures of the bearings. While the latter is not a sufficiently accurate index of the coefficient of friction to determine variations of the latter, when it falls below one-half of one per cent., yet a study of the data given for the concentric brasses will make it clear that the following basis of interpretation of the friction, from temperature, is practically acceptable.

1. At 170 revolutions per minute, and any load up to 10,000 lbs., if the temperature increases at the rate of from one-third to one-half degree per minute, the coefficient of friction is less than one-half of one per cent., and there is no danger of overheating in any length of trial. A thirty-minute run will then give the maximum temperature of the brass, which will be about 90° Fahr., in a still atmosphere.

2. At 170 revolutions per minute, and any load up to 10,000 lbs. load, if the temperature increases at the rate of one or more degrees per minute, so that at the end of thirty minutes' trial the temperature in a still atmosphere is upward of 120°, then the coefficient of friction is three or more per cent., and continued running would result in a "hot box."

The following tables give the record of each of the six brasses and one lead-lined brass.

The general programme followed is:

1st. To run ten minutes under 5,000 lbs. load and then note the extent and character of the bearing.

2d. To make two or more trials of thirty minutes under 5,000 and 10,000 lbs. load, to determine the liability of the brass to overheat, or to run at minimum friction.

3d. To use paraffine oil first and then follow with sperm under the most severe conditions to which the paraffine had been subjected, in order to discover any superior quality of sperm over the paraffine.

4th. To artificially create heating with emery dust, so as to note to what extent grit, accidentally entering between bearings, could cause "hot boxes."

The method of lubrication was by a pad 3 inches by 6 inches, pressed against the journal by springs, and taking its supply of oil with wicks. A reciprocating motion of five-sixteenths of an inch each way was maintained at the rate of about thirty-five double motions per minute.

HISTORY OF BRASS No. 1.—Fig. 159.

88 miles at 5,000 lbs. load; 34 miles at 10,000 lbs. load.

Date.	Test No.	Revs. per Minute.	Total Load. Lbs.	Duration of Trial. Hours.	Temp. of Brass. Fahr.	Area of Contact. Sq. Ins.	Press. per Sq. In. Lbs.	Kind of Oil.	Shape of Area of Contact, and Remarks.
	Prelim.	170	5000	$\frac{1}{6}$	....	2.25	4300	Paraffine.	$\frac{1}{2}$" wide × $4\frac{1}{2}$" long, uniform width.
Aug. 18	1	170	5000	$\frac{1}{4}$	74–101	5.25	2250	"	1" " × $5\frac{1}{4}$" " " "
" "	1	170	5000	$\frac{1}{4}$	101–108	5.25	2250	"	
" 20	2	170	5000	$\frac{1}{2}$	74–82	5.25	2250	"	Same as in [1], highly polished.
" 28	3	170	10000	$\frac{1}{2}$	76–84	5.25	4500	"	" " " " "
" 29	4	315	10000	$\frac{1}{2}$	74–120	5.25	4500	"	" " " " "
		Stopped and cooled with blower, and started again.							
"	"	315	10000	$\frac{1}{2}$	76–120	5.25	4500	"	" " " " "
		Stopped and changed oil to pure sperm.							
"	"	315	10000	$\frac{1}{2}$	110–120	5.25	4500	Sperm.	" " " " "
		Reduced speed.							
"	"	170	10000	$1\frac{1}{4}$	82–84	8.25	1200	"	$1\frac{3}{8}$" × 6", uniform width, high pol.
Sept. 5	5	170	10000	$1\frac{1}{4}$	70–78	8.25	1200	"	
		Three grains of emery thrown between bearings.							
"	"	170	10000	$\frac{1}{3}$	78–114	....	....	"	Slightly smoking.
		Added sperm with syringe to reinforce pad.							
"	"	170	10000	1	114–170	9.0	1100	"	Smoking heavily. Area of contact as per Fig. 159. Emery caused scratches at .... Elsewhere high polish.

HISTORY OF BRASS No. 2.—Fig. 160.

17 miles at 5,000 lbs. load ; 76 miles at 10,000 lbs. load.

Date.	Test. No.	Revs. per Minute.	Total Load. Lbs.	Duration of Trial. Hours.	Temp. of Brass. Fahr.	Area of Contact. Sq. Ins.	Press. per Sq. In. Lbs.	Kind of Oil.	Shape of Area of Contact, and Remarks.
August 18	Prelim.	170	5000	1/6	....	1.25	4000	Heavy Par-	½" × 2½". Uniform width at centre of crown.
" 18	1	170	5000	1/2	Smoking. 78–130	3.00	1660	affine.	1" × 3". " " " "
" 21	2	170	5000	1/2	76–88	3.00	1660	"	1" × 3". " " " "
" 28	3	170	10000	1/2	76–91	3.25	3100	"	1" × 3¼". " " " "
" 30	4	170	10000	4.0	70–86	3.50	2900	Sperm.	1" × 3½". " " " " Brass as per Fig. 160.

BRASS No. A.—Fig. 161.

17 miles at 5,000 lbs. load ; 93 miles at 10,000 lbs. load.

Date.	Test. No.	Revs. per Minute.	Total Load. Lbs.	Duration of Trial. Hours.	Temp. of Brass. Fahr.	Area of Contact. Sq. Ins.	Press. per Sq. In. Lbs.	Kind of Oil.	Shape of Area of Contact, and Remarks.
August 18	Prelim.	170	5000	1/6	....	2.6	2000	Paraffine.	5¼" long, dumb-bell shape; avg. ½" wide.
" 19	1	170	5000	1/2	74–84	4.0	1250	"	" " " " " ¾" "
" 20	2	170	5000	1/2	76–84	4.0	1250	"	" " " " " " "
" 28	3	170	10000	1/2	79–90	4.5	2200	"	" " " " " " "
Sept. 1	4	170	10000	4	66–83	5.0	2200	Sperm.	" " " " " " "
" 9	5	170	10000	1/2	73–98	5.0	2200	"	" "
			Put three grains of emery in bearings.						
" 9	5	170	10000	3/4	78–93	5.0	2200	"	
" 9	5		Put second dose of emery in bearings.						
" 9	5	170	10000	1/6	Smoking. 92–154	9.0	1100	....	6" × 1½", uniform width.
" 9	5	170	10000	1/6	154–227 Heavy Smoke.	9.0	1100	....	Brass as Fig. 161.

HISTORY OF BRASS No. C.—FIG. 162.

27 miles at 5,000 lbs. load ; 158 miles at 10,000 lbs. load.

Date.	Test No.	Revolutions per Minute.	Total Load. Lbs.	Duration of Trial. Hours.	Temperature of Brass. Fahr.	Area of Contact. Sq. In.	Pressure per Sq. In. Lbs.	Kind of Oil.	Shape of Area of Contact, and Remarks.
Aug. 18	Prelim.	170	5000	$\frac{1}{6}$	....	3.75	1330	Paraffine.	5" × $\frac{3}{4}$" uniform width.
" 19	1	170	5000	$\frac{1}{4}$	74–113	....	....	"	Smoking. Centre of journal, 244°.
" 19	1	170	5000	$\frac{1}{4}$	113–146	4.2	1200	"	5$\frac{1}{4}$" × $\frac{7}{8}$".
" 20	2	170	5000	$\frac{1}{2}$	76–120	....	....	"	Smoking. Centre of journal, 180°
" 21	3	170	5000	$\frac{1}{2}$	76–94	....	....	"	" " " 114°
" 28	4	170	10000	$\frac{1}{2}$	70–97	4.2	2400	"	5$\frac{1}{4}$" × $\frac{7}{8}$" " " 123°.
Sept. 2	5	170	10000	4	66–84	5.0	2000	Sperm.	5$\frac{1}{2}$" × $\frac{15}{16}$".
" 3	5	170	10000	2.5	70–90	....	....	Paraffine.	
		Three grains of emery between bearings.							
" 3	6	170	10000	$\frac{1}{2}$	90–128	....	....	"	
		Changed to sperm oil.							
" 3	6	170	10000	1	128–140	....	....	Sperm.	
" 3	6	170	10000	$\frac{1}{2}$	140–130	6.8	1500	"	6" × 1$\frac{1}{4}$". Fig. 162.

2d. To make two or more trials of thirty minutes under 5,000 and 10,000 lbs. load, to determine the liability of the brass to overheat, or to run at minimum friction.

3d. To use paraffine oil first and then follow with sperm under the most severe conditions to which the paraffine had been subjected, in order to discover any superior quality of sperm over the paraffine.

4th. To artificially create heating with emery dust, so as to note to what extent grit, accidentally entering between bearings, could cause "hot boxes."

The method of lubrication was by a pad 3 inches by 6 inches, pressed against the journal by springs, and taking its supply of oil with wicks. A reciprocating motion of five-sixteenths of an inch each way was maintained at the rate of about thirty-five double motions per minute.

HISTORY OF BRASS NO. 1.—FIG. 159.

88 miles at 5,000 lbs. load; 91 miles at 10,000 lbs. load.

Date.	Test No.	Revs. per Minute.	Total Load. Lbs.	Duration of Trial. Hours.	Temp. of Brass. Fahr.	Area of Contact. Sq. Ins.	Press. per Sq. In. Lbs.	Kind of Oil.	Shape of Area of Contact, and Remarks.
	Prelim.	170	5000	$\frac{1}{6}$	....	2.25	4300	Paraffine.	$\frac{1}{2}''$ wide × $4\frac{1}{2}''$ long, uniform width.
Aug. 18	1	170	5000	$\frac{1}{4}$	74–101	5.25	2250	"	1" " × $5\frac{1}{4}''$ " " "
" "	1	170	5000	$\frac{1}{4}$	101–108	5.25	2250	"	
" 20	2	170	5000	$\frac{1}{2}$	74–82	5.25	2250	"	Same as in [1], highly polished.
" 28	3	170	10000	$\frac{1}{3}$	76–84	5.25	4500	"	" " " " "
" 29	4	315	10000	$\frac{1}{2}$	74–120	5.25	4500	"	" " " " "
Stopped and cooled with blower, and started again.									
" "	"	315	10000	$\frac{1}{2}$	76–120	5.25	4500	"	" " " " "
Stopped and changed oil to pure sperm.									
" "	"	315	10000	$\frac{1}{2}$	110–120	5.25	4500	Sperm.	" " " " "
Reduced speed.									
" "	"	170	10000	$1\frac{1}{4}$	82–84	8.25	1200	"	$1\frac{3}{8}''$ × 6", uniform width, high pol.
Sept. 5	5	170	10000	$1\frac{1}{4}$	70–78	8.25	1200	"	
Three grains of emery thrown between bearings.									
" "	"	170	10000	$\frac{1}{3}$	78–114	....	....	"	Slightly smoking.
Added sperm with syringe to reinforce pad.									
" "	"	170	10000	1	114–170	9.0	1100	"	Smoking heavily. Area of contact as per Fig. 159. Emery caused scratches at .... Elsewhere high polish.

Figs. 159 to 164 are reproductions of photographs of the final condition of the brasses.

The following is a résumé of the results in the tables, not including the cases where emery is used:

	Range of Press. per sq. in., Lbs.	Times Tried.	Times Overheated.
Brass 1 at 5000 lbs.	2250–4800	2	1
" " " 10000 "	1200–4500	3	0
" 2 " 5000 "	1600–4000	2	1
" " " 10000 "	2900–3100	2	0
" A " 5000 "	1250–2000	2	0
" " " 10000 "	1100–2250	3	0
" F " 5000 "	1400–2250	2	0
" " " 10000 "	2100–1200	5	0
" B " 5000 "	1800–3500	2	1
" " " 10000 "	1800–3300	3	2
" C " 5000 "	1200–1300	3	2
" " " 10000 "	2000–2400	3	0
	Totals........	32	7

These figures show how purely a matter of chance is the overheating, as a brass which ran hot at 5,000 lbs. load on one day would run cool on a later date at the same or higher pressure. The explanation of this apparently arbitrary difference of behavior is that the *accidental variations of the smoothness of the surfaces, almost infinitesimal in their magnitude, cause variations of friction which are always tending to produce overheating, and it is solely a matter of chance when these tendencies preponderate over the lubricating influence of the oil.* There is no appreciable advantage shown by the sperm oil, when there is no tendency to overheat—that is, the paraffine can lubricate under the highest pressures which occur, as well as the sperm, when the surfaces are within the conditions affording the minimum coefficients of friction.

But when a few grains of emery are thrown between the bearings, intense heat is generated at the point of introduction. This is shown by the burnt oil making the black irregular stains in Figs. 159, 160, etc., due to the emery cutting its way through the metals of the bearings. *Under these circumstances the paraffine volatilizes, and utterly destroys lubrication, while the sperm resists volatilization, and makes the heating of the whole journal take place more slowly.* This appears by comparing the record of Brass A, which took two doses of emery to overheat it with the sperm, with that of Brass F.

In other words, the sperm and other oils of high heat-resisting qualities, like vegetable oil and petroleum cylinder stocks, only differ from the more volatile lubricants, like paraffine, in their ability to reduce the chances of the continual accidental infinitesimal abrasion producing overheating. To exhibit thoroughly such differences, a considerable number of brasses must be tried, and as soon as one of them promises to overheat with one oil, the other oil must be substituted, and its behavior noted. From a number of such trials, differences of lubricating property of oils may be found. But such differences cannot be developed when the bearing surfaces are in condition to give the minimum coefficients of friction. This idea is discussed more at length in a paper* read at the meeting of the American Association for Advance of Science, 1890.

The lead-lined brass's record follows that of Brass B, which had sensibly roughened the journal. These figures show that the lead lining automatically adjusts itself to any irregularities in the journal, and finally secures minimum friction. The resistance of the lead to being scored by an excrescence, on the journal, is much less than that of the bare brass, so that while there is a temporary increase of friction, no very intense heat is created.

REMARKABLE CASE OF OVERHEATING OF LEAD-LINED BRASSES.

Fig. 165 shows one end of a lot of lead-lined brasses which in the Pullman service of a neighboring railroad gave so much trouble from overheating that they were withdrawn from use, and broken so as to expose the structure of the brass. It then appeared that the centre of the brass, as shown by the darker portion of the cross-section in Fig. 165, differed essentially from the outer portions of its body. The attention of the writer being called to the matter, a chemical analysis was made of the different portions of the brass, the drill holes shown in the latter being made to obtain the samples for examination. It was found that the inner portion of the brass was devoid of phosphorus, while the outer portion contained a little more of this element than belongs to the toughest variety of phosphor bronze.

This irregularity of structure was apparently the cause of the lead lining wearing so that the brass near the centre of the bear-

* Am. Mach., Oct. 24, 1890.

HISTORY OF BRASS No. B.—Fig. 163.

21 miles at 5,000 lb. load; 94 miles at 10,000 lbs. load.

Date.	Test No.	Revs. per Minute.	Total Load. Lbs.	Duration of Trial. Hours.	Temp. of Brass. Fahr.	Area of Contact. Sq. Ins.	Press. per Sq. In. Lbs.	Kind of Oil.	Shape of Area of Contact, and Remarks.
August 18	Prelim.	170	5000	$\frac{1}{6}$	....	1.38	3500	Paraffine.	One spot $\frac{1}{2}'' \times \frac{1}{4}''$. " " $\frac{5}{8}'' \times 2''$.
" 19	1	170	5000	$\frac{1}{3}$	75–110	2.8	1800	"	Smoking, $\frac{1}{2}'' \times 1\frac{3}{4}''$.
" 19	1	170	5000	$\frac{1}{3}$	110–147	2.8	1800	"	Spots joined, $1'' \times 1\frac{3}{4}''$.
" 21	2	170	5000	$\frac{1}{2}$	76–88	2.8	1800	"	" " "
" 28	3	170	10000	$\frac{1}{2}$	78–126	3.0	3300	"	Smoking " "
September 1	4	170	10000	4	78–92	4.3	2400	Sperm.	$3\frac{1}{2}'' \times 1\frac{1}{4}''$ uniform width.
" 4	5	170	10000	$\frac{1}{2}$	70–110	....	....	Paraffine.	
Changed to sperm oil.									
" 4	5	170	10000	$\frac{1}{6}$	110–128	....	....	....	
" 4	5	170	10000	$\frac{1}{2}$	128–180	5.6	1800	....	$4'' \times 1\frac{1}{2}''$, Fig. 163.
Lead-lined brass applied.									

LEAD-LINED BRASS.—Fig. 158.

17 miles at 5,000 lbs. load; 29 miles at 10,000 lbs. load.

Date.	Test No.	Revs. per Minute.	Total Load. Lbs.	Duration of Trial. Hours.	Temp. of Brass. Fahr.	Area of Contact. Sq. Ins.	Press. per Sq. In. Lbs.	Kind of Oil.	Shape of Area of Contact, and Remarks.
September 4	1	170	10000	$\frac{1}{2}$	76–112	....	....	Sperm.	Smoking.
" 4	1	170	5000	$\frac{1}{4}$	112–122	....	....	....	Smoking stopped.
" 4	1	170	5000	$\frac{1}{4}$	122–112	....	....	....	
" 4	1	170	5000	$\frac{1}{4}$	112–102	....	....	....	
Load increased to 10,000 lbs.									
" 4	1	170	10000	$\frac{1}{4}$	102–100	....	....	....	Lead flowed as per Fig. 158.
" 4	1	170	10000	$\frac{3}{4}$	100–92	24	420	....	Bearing $8'' \times 6''$.

under full steam would cause such mutilation whatever the lubricant.

REDUCTION OF FRICTION BY EXCESSIVE APPLICATION OF GRITTY MATERIAL TO A BEARING.

The phenomenon of apparently cooling excessively over-heated bearings, by applying grinding material such as emery or sand, is well known. To explain how this paradoxical treatment can afford relief to a hot journal, the following experiments are related:

A journal about $1\frac{1}{2}$ inches square is fitted with double brasses, and the surfaces finished with only the ordinary care bestowed in turning shafting. The brasses are clamped upon the journal by a spring affording 1,000 lbs. pressure per square inch; the journal is mounted in a lathe so that, by means of back gearing, any amount of friction can be overcome at steady speed. An arm extends from the brasses and rests upon a platform scale so that the friction can be measured. Oil is supplied through a vertical oil hole kept full with a squirt can, and the brasses are provided with liberal oil grooves for distributing the lubricant, and precautions are taken to bevel the edges of the brasses, etc., so as to afford all ordinary opportunities for good lubrication. The oil used is pure sperm. The following is a typical record:

Duration of Trial. Minutes.	Temperature, Fahrenheit.	Revolutions per Minute.	Coefficient of Friction, per cent.
5	122°	35	16
10	140	35	18
15	167	35	22
20	194	35	21
25	208	35	20
30	212	35	18
35	206	35	16

Change pressure to 200 lbs. per square inch, and revolutions to 330 per minute, and record may be as follows:

Duration of Trial. Minutes.	Temperature, Fahrenheit.	Revolutions per Minute.	Coefficient of Friction, per cent.
5	122	330	$7\frac{1}{2}$
10	140	330	6
15	145	330	5
20	150	330	5

Duration of Trial. Minutes.	Temperature, Fahrenheit.	Revolutions per Minute.	Coefficient of Friction, per cent.
Applied thick paste of emery and oil.			
25	....	330	9
30	....	330	16
35	....	330	12
40	....	330	13
50	....	330	10
Cleaned bearings thoroughly and applied pure sperm oil.			
5	....	330	10
10	....	330	12
Reapplied one thousand pounds pressure.			
5	....	330	16
10	....	330	14
20	....	330	14
30	....	330	14
Covered surface of bearings with paste of emery and sperm oil.			
5	....	330	16
15	....	330	13
25	....	330	13

The effect of the emery upon the surfaces of the bearings is to cover the latter with a series of parallel grooves, and, apparently, after such grooves are made the presence of the emery does not practically increase the friction over the amount of the latter when pure oil only is between the surfaces. The infinite number of grooves constitute a very perfect means of insuring a uniform oil supply at every point of the bearings. As long as grooves in the journal match with those in the brasses the friction appears to amount to only about 10% to 15% of the pressure. But if a smooth journal is placed between a set of brasses which are grooved, and pressure be applied, the journal crushes the grooves and becomes brazed or coated with brass, and then the coefficient of friction becomes upward of 40%. If then emery is applied, the friction is made very much less by its presence, because the grooves are made to match each other, and a uniform oil supply prevails at every point of the bearings, whereas before the application of the emery many spots of the latter receive no oil between them.

In other words, the sperm and other oils of high heat-resisting qualities, like vegetable oil and petroleum cylinder stocks, only differ from the more volatile lubricants, like paraffine, in their ability to reduce the chances of the continual accidental infinitesimal abrasion producing overheating. To exhibit thoroughly such differences, a considerable number of brasses must be tried, and as soon as one of them promises to overheat with one oil, the other oil must be substituted, and its behavior noted. From a number of such trials, differences of lubricating property of oils may be found. But such differences cannot be developed when the bearing surfaces are in condition to give the minimum coefficients of friction. This idea is discussed more at length in a paper* read at the meeting of the American Association for Advance of Science, 1890.

The lead-lined brass's record follows that of Brass B, which had sensibly roughened the journal. These figures show that the lead lining automatically adjusts itself to any irregularities in the journal, and finally secures minimum friction. The resistance of the lead to being scored by an excrescence, on the journal, is much less than that of the bare brass, so that while there is a temporary increase of friction, no very intense heat is created.

REMARKABLE CASE OF OVERHEATING OF LEAD-LINED BRASSES.

Fig. 165 shows one end of a lot of lead-lined brasses which in the Pullman service of a neighboring railroad gave so much trouble from overheating that they were withdrawn from use, and broken so as to expose the structure of the brass. It then appeared that the centre of the brass, as shown by the darker portion of the cross-section in Fig. 165, differed essentially from the outer portions of its body. The attention of the writer being called to the matter, a chemical analysis was made of the different portions of the brass, the drill holes shown in the latter being made to obtain the samples for examination. It was found that the inner portion of the brass was devoid of phosphorus, while the outer portion contained a little more of this element than belongs to the toughest variety of phosphor bronze.

This irregularity of structure was apparently the cause of the lead lining wearing so that the brass near the centre of the bear-

* Am. Mach., Oct. 24, 1890.

VALUE OF ANTI-FRICTION METALS.

The various white metals available for lining brasses dc afford coefficients of friction lower than can be obtained bare brass, but they are less liable to "overheating," becau the superiority of such material over bronze in ability to pe of abrasion or crushing, without excessive increase of frictic

For example, several brasses similar to those used in experiments under the last head being lined with one ol most successful anti-friction metals, the following is typic their behavior.

Pressure per square inch, 1,000 lbs. Lubricant, sperr and emery.

Duration of Trial. Minutes.	Temperature. Fahr.	Revolutions per Minute.	Coefficient of Friction.
5	129	35	16%
15	214	35	23
30	208	35	30
45	208	35	20

Bearings were thoroughly grooved at the end of this trial new smooth journal was then placed between the gro brasses, and 1,000 lbs. pressure per square inch again app with results as follows :

Duration of Trial. Minutes.	Revolutions per Minute.	Temperature. Fahr.	Coefficient of Friction.
5	35	149	23%
15	35	212	23
30	35	220	20
45	35	198	14
60	35	157	8
70	35	149	8½
80	35	150	9

On examination the grooves were found crushed d smoothly and the journal smooth, showing that the white n refuses to be provoked into running with excessive fric under circumstances which would always cause overhea with bare brasses. In other words, the use of soft white m reduces the *chances* of overheating. The above bearings, b cleansed of oil in ether and run dry at 35 revolutions and 1,000 lbs. pressure per square inch, gave a coefficient of friction during half an hour of .16%, and the surfaces remained smooth. Sperm oil was then supplied and the speed doubled. After 20 minutes the temperature was 390° Fahr., and the metal collapsed by melting, as per Fig. 172.

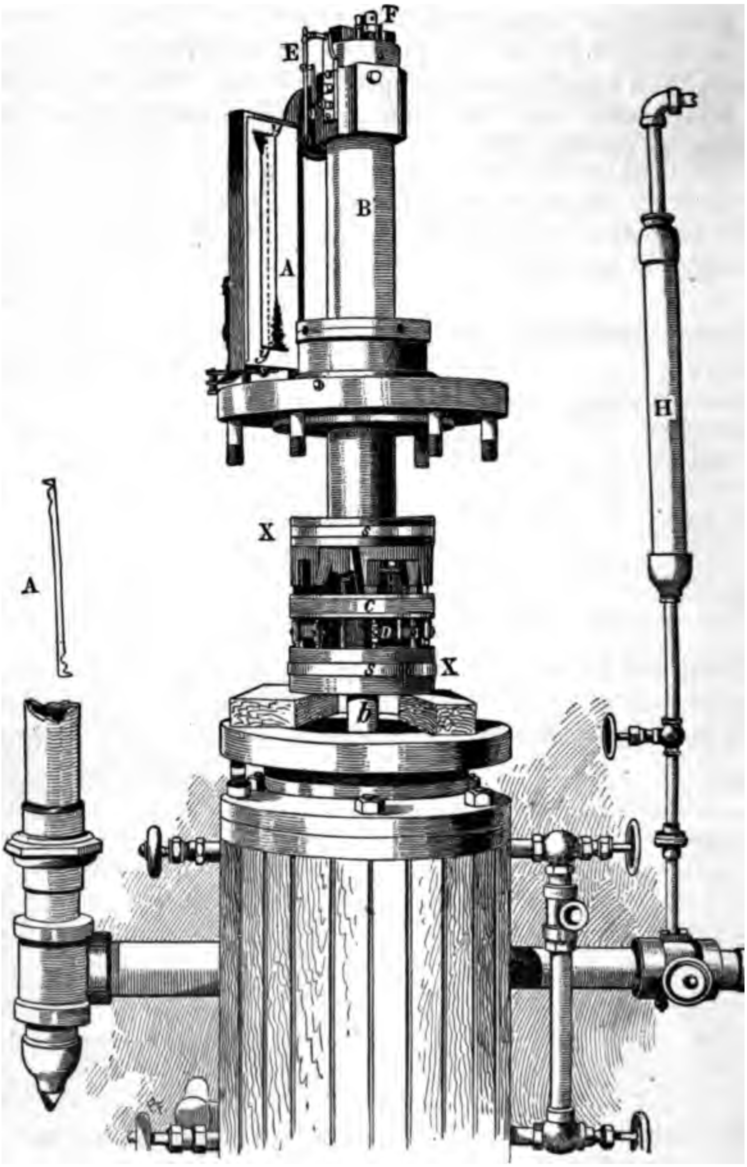


FIG. 146.
Enlarged view of cylinder oil tester.

The following is a typical record:

NEW SMOOTH JOURNAL PLACED BETWEEN BRASSES GROOVED BY USE OF EMERY AND LUBRICATED WITH PURE SPERM.

Duration of Trial. Minutes.	Temperature, Fahr.	Revolutions per Minute.	Coefficient of Friction, per cent.
5	.....	35	40
10	......	"	39
15	.....	"	40
20	......	"	35
25	......	"	40
Applied emery.			
5	......	35	20
10	......	"	19
15	......	"	14
20	......	"	14
Cleaned bearings and applied pure sperm.			
5	......	35	20
10	......	"	16
15	......	"	16

Pressure, 1,000 lbs. per square inch.

The appearance of bearings as grooved by emery is shown in Fig. 171. The conclusion is, that the relief afforded by a grinding material applied to an overheated bearing is due to the fact that the surfaces are changed from an irregular condition of contact to a series of parallel grooves, whereby oil is carried to every point of the bearing surfaces, so that the friction resulting from the application of the grinding material is much less than that existing before its application; but, nevertheless, such friction is from two to three times the amount which permits bearings to run thoroughly cool. The latter fact prevents the explanation offered from conflicting with the fact that grit added to bearings, which are giving minimum coefficients of friction, creates an amount of overheating, which bears the same relation to a perfectly cool condition as the condition which the emery relieves bears to the lubrication finally produced by the emery. Cooling compounds produce relief on the same principles. But their cutting power for the production of parallel grooves is much feebler than that of emery.

Hence, unless applied in the early stages of heating—that is, before the abrasion of the bearings is such as to make the bearings heat excessively in a few moments—they are unable to produce a reduction of friction.

VALUE OF ANTI-FRICTION METALS.

The various white metals available for lining brasses do not afford coefficients of friction lower than can be obtained with bare brass, but they are less liable to "overheating," because of the superiority of such material over bronze in ability to permit of abrasion or crushing, without excessive increase of friction.

For example, several brasses similar to those used in the experiments under the last head being lined with one of the most successful anti-friction metals, the following is typical of their behavior.

Pressure per square inch, 1,000 lbs. Lubricant, sperm oil and emery.

Duration of Trial. Minutes.	Temperature. Fahr.	Revolutions per Minute.	Coefficient of Friction.
5	129	35	$16\frac{7}{8}$
15	214	35	23
30	208	35	30
45	208	35	20

Bearings were thoroughly grooved at the end of this trial. A new smooth journal was then placed between the grooved brasses, and 1,000 lbs. pressure per square inch again applied, with results as follows:

Duration of Trial. Minutes.	Revolutions per Minute.	Temperature. Fahr.	Coefficient of Friction.
5	35	149	$23\frac{7}{8}$
15	35	212	23
30	35	220	20
45	35	198	14
60	35	157	8
70	35	149	$8\frac{1}{2}$
80	35	150	9

On examination the grooves were found crushed down smoothly and the journal smooth, showing that the white metal refuses to be provoked into running with excessive friction, under circumstances which would always cause overheating with bare brasses. In other words, the use of soft white metals reduces the *chances* of overheating. The above bearings, being cleansed of oil in ether and run dry at 35 revolutions and 1,000 lbs. pressure per square inch, gave a coefficient of friction during half an hour of .16%, and the surfaces remained smooth. Sperm oil was then supplied and the speed doubled. After 20 minutes the temperature was 390° Fahr., and the metal collapsed by melting, as per Fig. 172.

FIG. 151.

New brass run for 10 minutes to show bearing.

FIG. 154.

Brass affording minimum coefficients of friction, Table II.

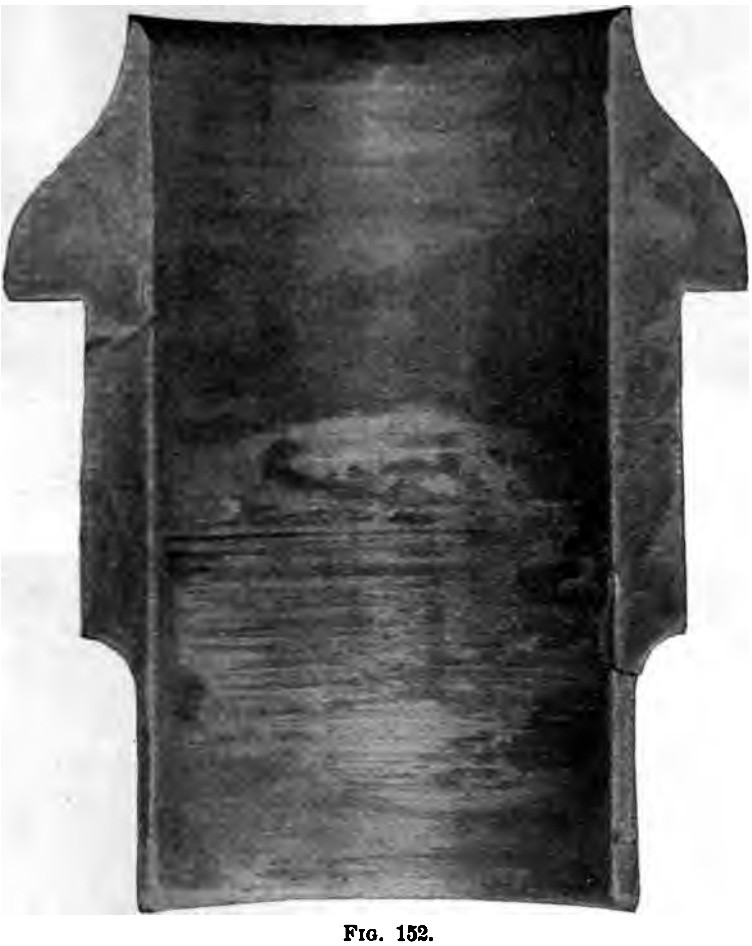


FIG. 152.

Brass run on testing machine until journal was red hot.

FIG. 153.

Sections of Pullman car brasses overheated in service.

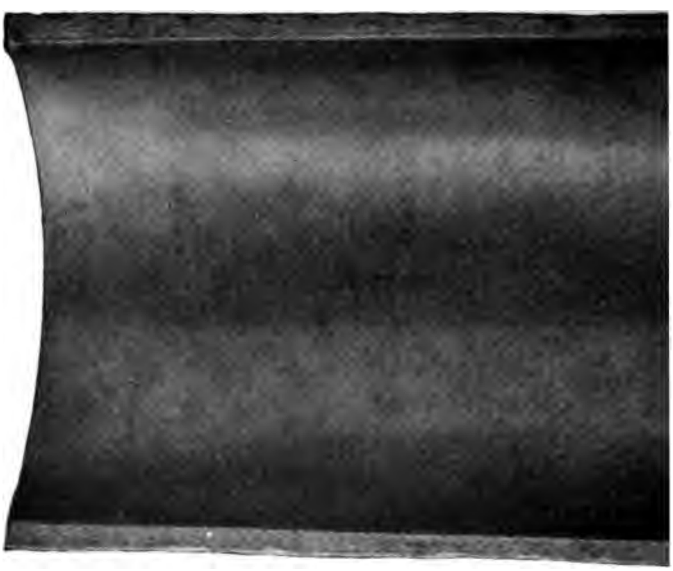


FIG. 155.

Pullman car brass worn out in service without overheating.

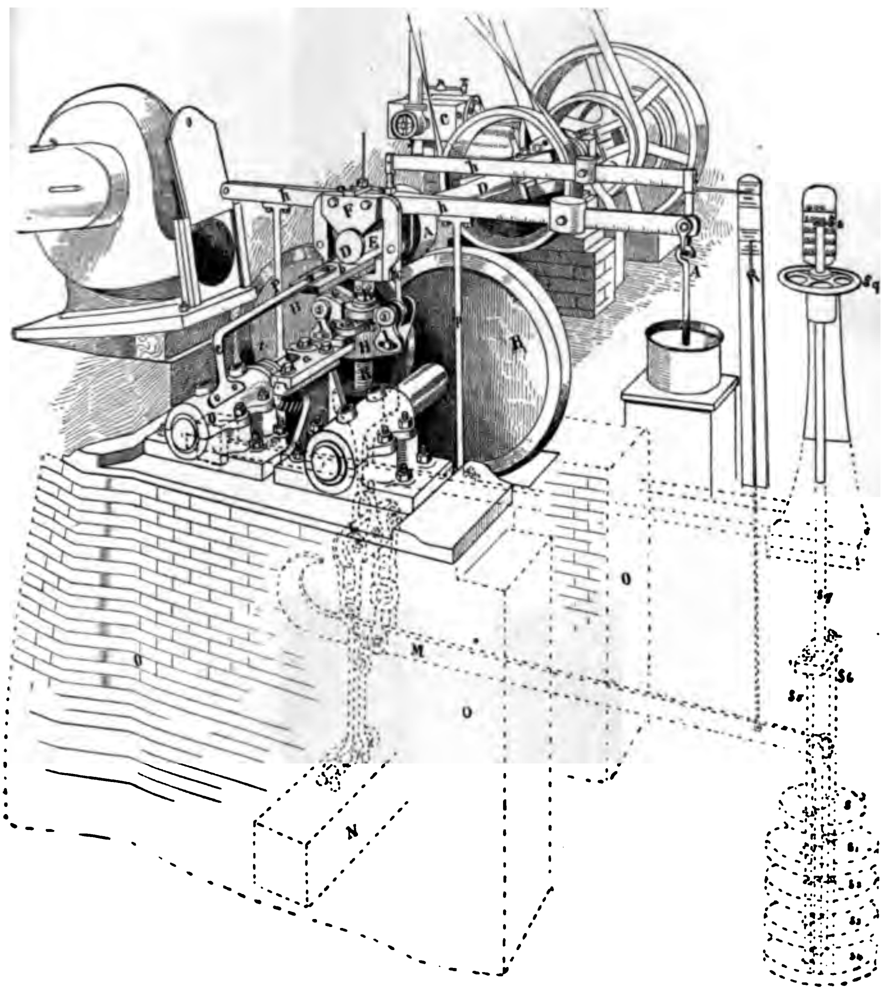


Fig. 150.

General view of oil tester for heavy bearing pressures.

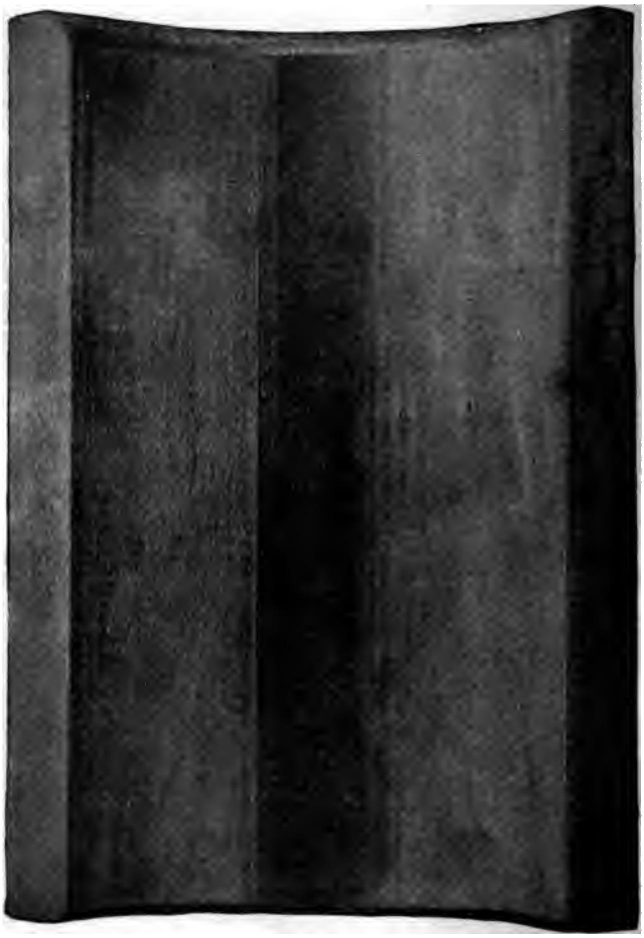


FIG. 157.

Brass cut away to reduce bearing to ½ inch by 6 inches.

FIG. 158.
Lead-lined brass, showing flow of lead under pressure.

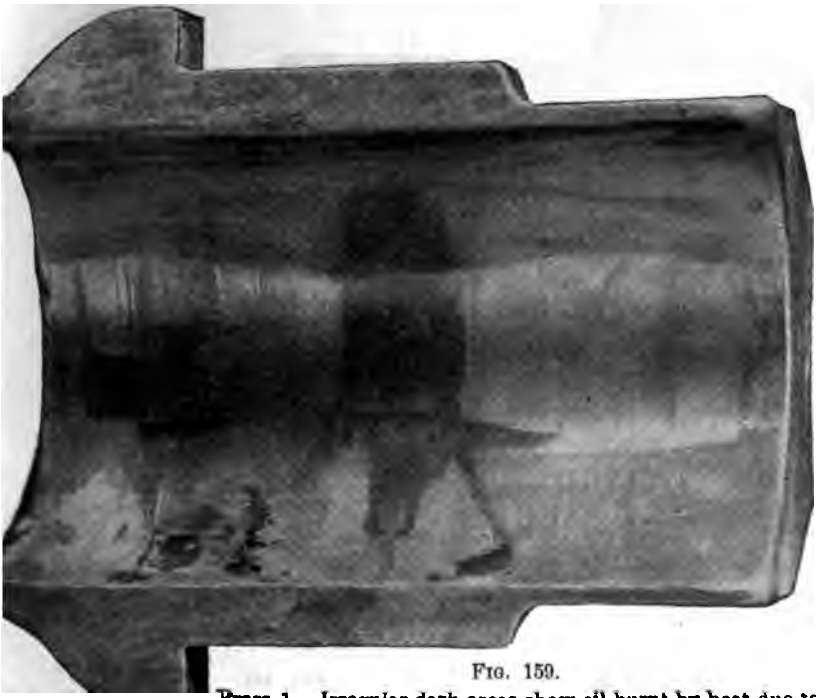


FIG. 159.
Brass 1.—*Irregular* dark areas show oil burnt by heat due to action of emery.

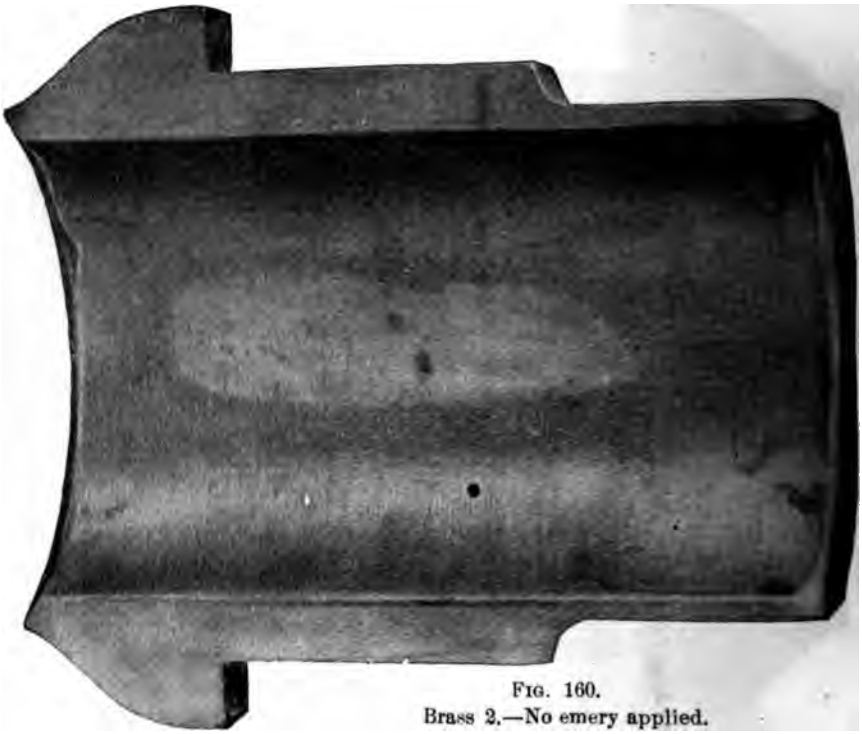


FIG. 160.
Brass 2.—No emery applied.

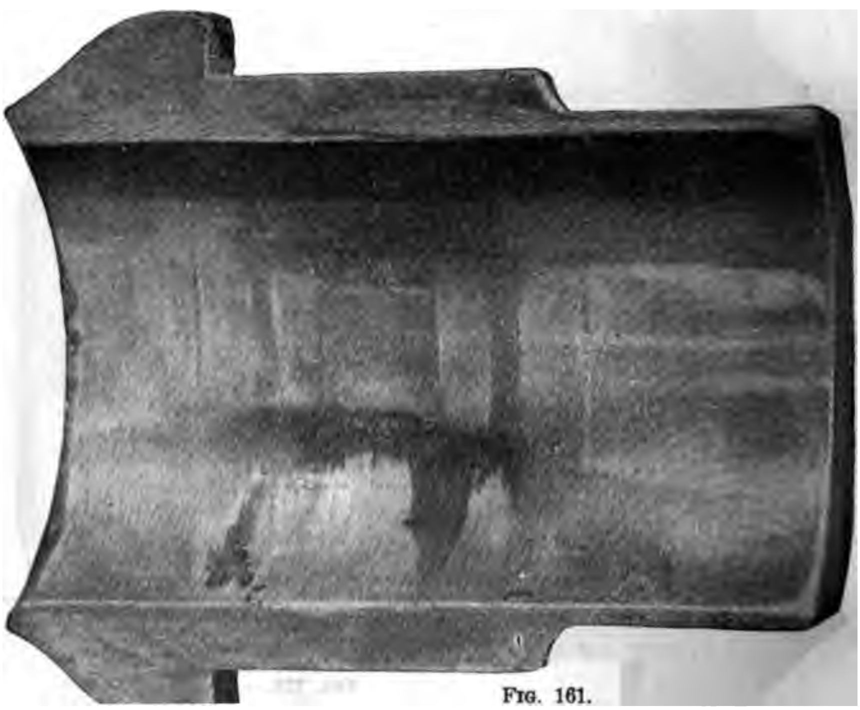


FIG. 161.
Brass A.—Irregular dark areas show oil burnt by heat due to action of emery.

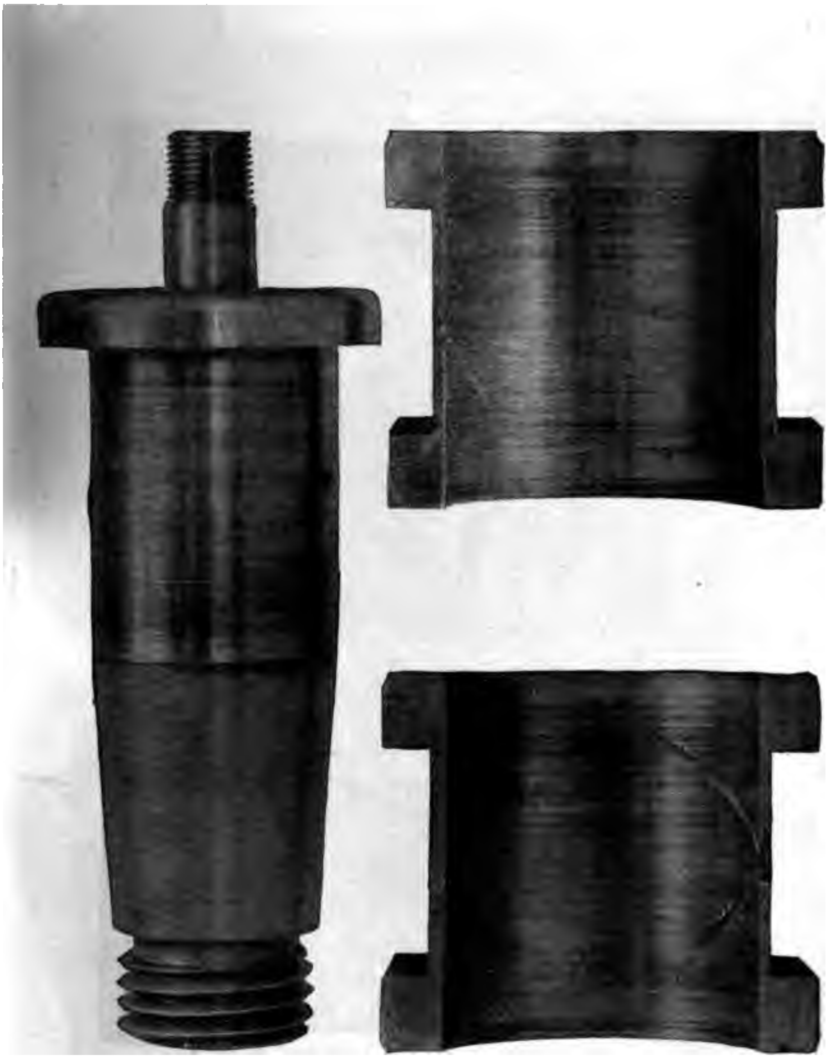


FIG. 156.
Crank-pin brasses worn in grooves by absence of lateral motion.

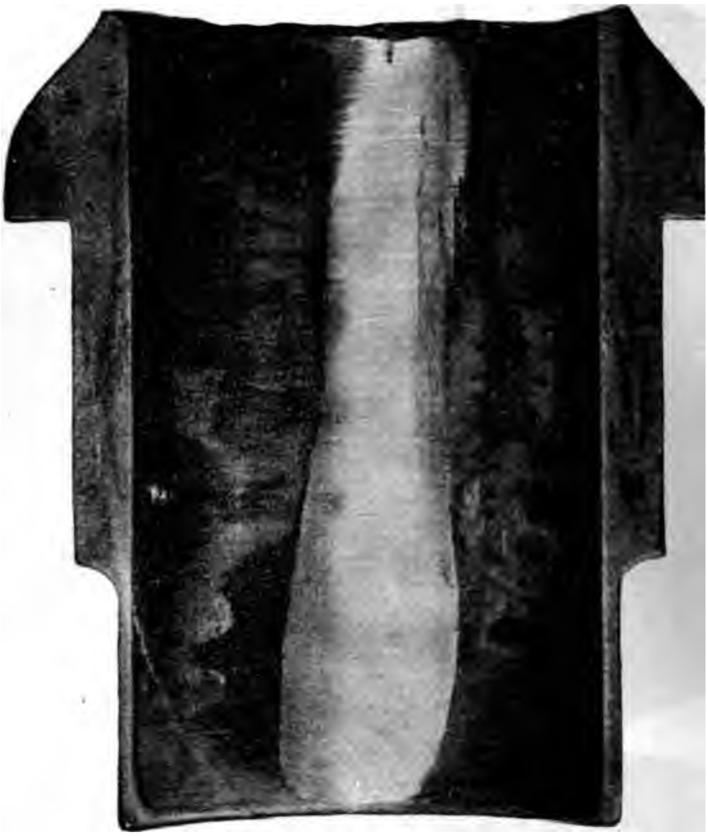


FIG. 164.
Brass F.—Blackened surface caused by oil taking fire throughout the bo

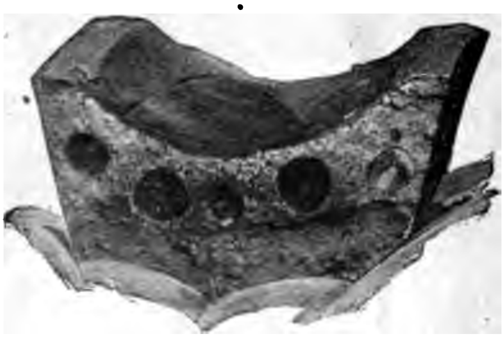


FIG. 165.
Section of lead-lined brass overheated in Pullman service.

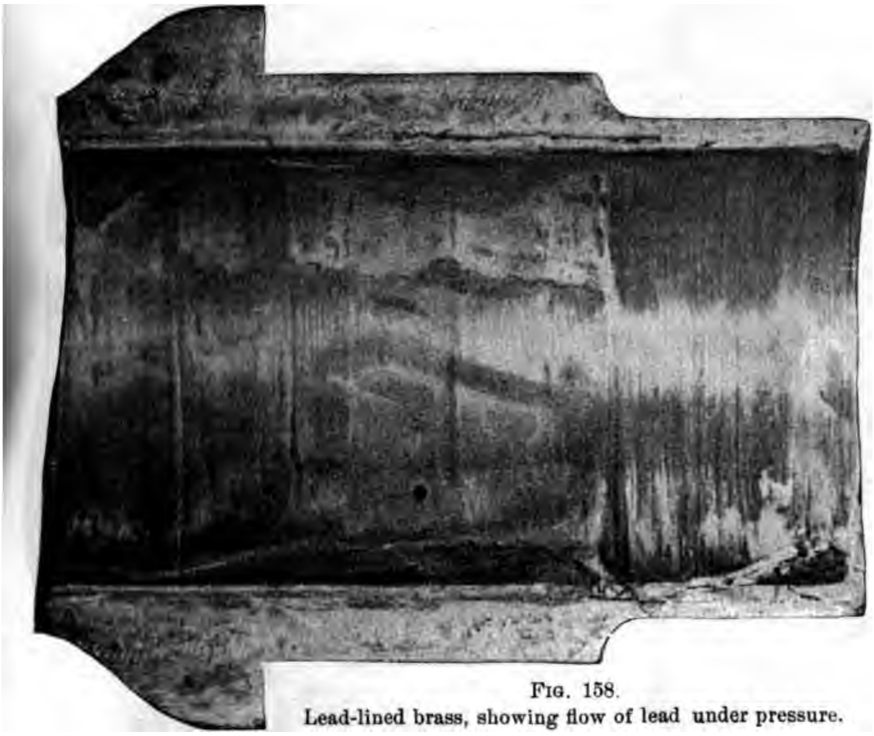


FIG. 158.
Lead-lined brass, showing flow of lead under pressure.

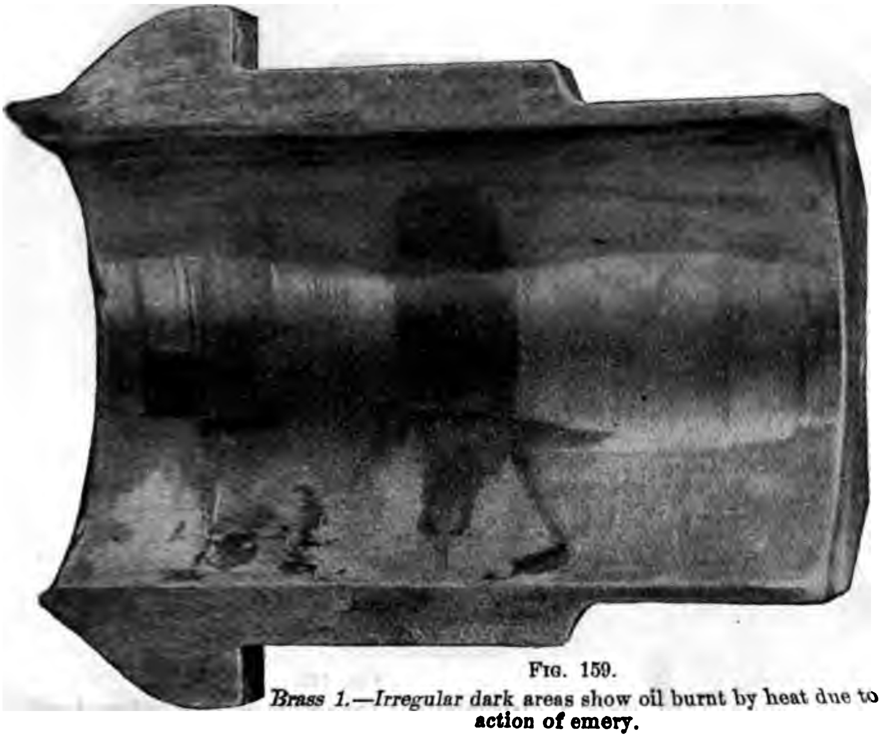


FIG. 159.
Brass 1.—Irregular dark areas show oil burnt by heat due to **action of emery.**

FIG. 167.
Mutilated crank-pin due to overheating.

FIG. 169.
Wrought-iron journals with cinder cracks.

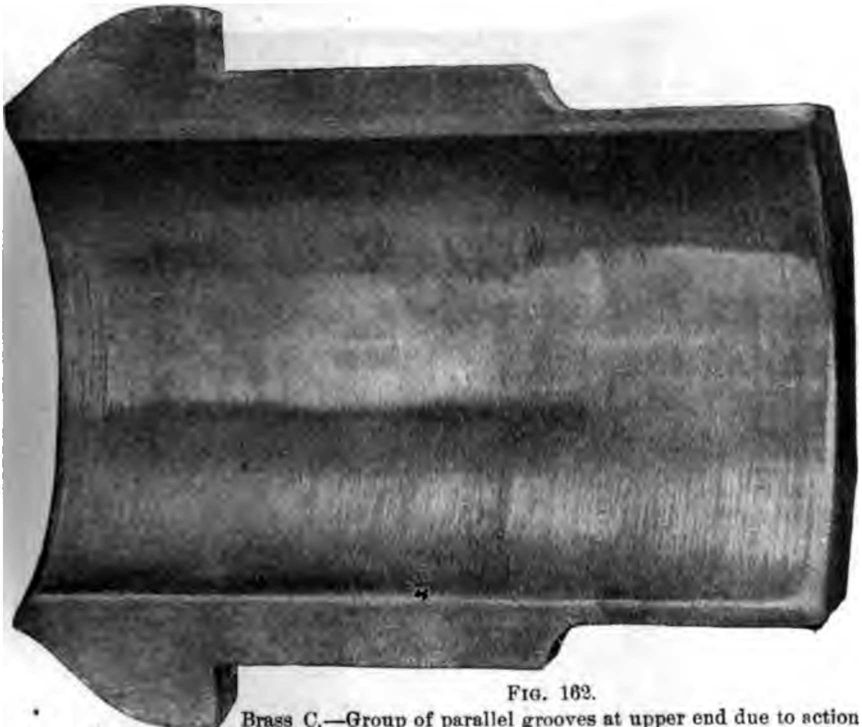


FIG. 162.
Brass C.—Group of parallel grooves at upper end due to action of emery.

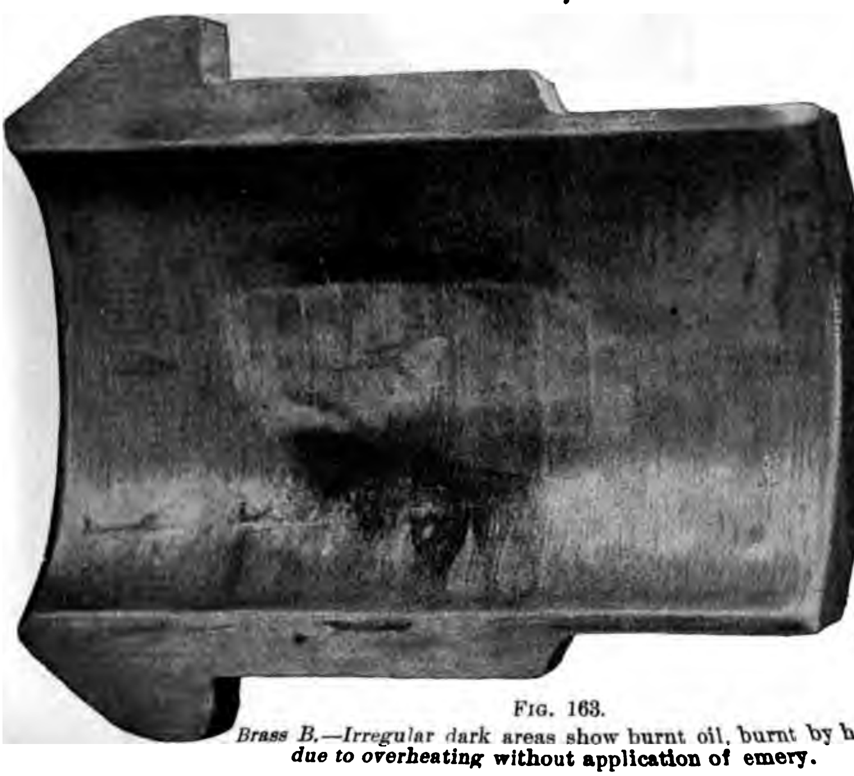


FIG. 163.
Brass B.—Irregular dark areas show burnt oil, burnt by heat due to overheating without application of emery.

DISCUSSION.

Mr. Carleton W. Nason.—Travelling a few years ago in a new car over the Lehigh Valley Railroad, we were much delayed by the heating of the brasses. Finally the trouble was radically checked by a brakeman, who picked up a handful of gravel from the road-bed, and threw it into the box. It ran cool for the rest of the trip.

Mr. Jas. McBride.—I understand Prof. Denton to say that, wherever it is possible, it is good practice to give a journal end or lateral motion in any machinery that would admit of it?

Prof. Denton.—Mr. Nason's case is discussed at the back of the paper, where I put emery in to reduce the friction, just as though it were sand.

The answer to Mr. McBride's question is, that the lateral motion does produce a surface unattainable in any other way.

CCCCXXXIV.*

AN ENGINEERING PROBLEM AT RICHMOND, VA.

BY W. H. ADAMS, NEW YORK.

(Member of the Society.)

THE Southern States are well launched on a tide of prosperity, resulting from a business development of natural resources, which is attracting the attention of engineers from all climes and from nearly every branch of the profession. They have in their special work already revolutionized many of the seemingly fixed ideas of the century, and it is daily becoming manifest to careful observers that nothing can stay the progress of events which shall place the control of leading staple products almost entirely in the hands of Southern capitalists, and within a comparatively short time.

The multitudinous industries which are springing up on every hand, and in directions only appreciated within the last few years, depend for their well-being greatly, and in some cases wholly, upon such lines of transportation as find their terminals upon the Atlantic coast, or upon the rivers which lead to the ocean. Everywhere throughout the South railroads keep pace with the advancing wave of development, and combinations are constantly being formed from integral parts of the inland systems of railway, which are intended to secure to them the advantages of natural outlets on tide-water.

There is, however, a manifest lack of comprehensive measures for developing the outlets on the Atlantic to the enormous growth of the interior trade, and while, perhaps, this can be easily shown to be due almost entirely to local circumstances and conditions which have prevailed during the past twenty years, yet, in view of the surprises in store for us when the facts of the late census are gathered, and the changes are admitted sure to take place in the next decade, we should be quick to study and comprehend any scheme of improvement, now pre-

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

Fig. 167.
Mutilated crank-piu due to overheating.

Fig. 169.
Wrought-iron journals with cinder cracks.

150 miles, equal to 6,000 square miles of territory—are not only practicable from the nature of the bottom lands of the upper country, but inexpensive (considering the advantages to be gained) as compared with public improvements of like nature in other parts of the world. The measured discharge of the James River, taken at Richmond under normal conditions, varies from 2,000 cubic feet per second to 10,000 cubic feet per second—a fair average fall of, say, 4,000 cubic feet per second. Freshets are of a character difficult to understand without close study of the topography of the country about Richmond, but at such periods the estimated discharge of water past the city of Richmond is as high as 200,000 cubic feet per second. To grasp fully the economic value of such a volume of water as is here obtainable, if evenly distributed throughout the season and made use of by industrial cities along the river, it should be understood that within thirteen miles from the city (Richmond being at the head of navigation and on tide-water) there is a fall of over 115 feet, and from tide-water to the gap in the Alleghanies, from which the James River takes its source, the elevation is about 585 feet. By any reasonable system of impounding reservoirs within a distance of 50 miles of the city there could be guaranteed at least 4,000 cubic feet per second during summer months, and under a head of 115 feet this would average, say, 40,000 H.P. To enable the reader to compare this measured value with well-known sections where the question has been answered by the expenditure of millions of money to secure the advantages of power of this character, the following table of the horse-power for the summer months alone has been compiled:

City.	*Horse-power.*
James River at Richmond, Va., with reservoirs	40,000
James River at Richmond, Va., at present	25,000
Holyoke, Mass.	12,260
Manchester, N. H.	12,000
Lowell, Mass.	11,845
Lewiston, Me.	11,000
Lawrence, Mass.	10,902
Cohoes, N. Y.	6,556

It is thus evident that the ordinary summer conditions compare favorably with the well-known examples above, while great freshets furnish over two million horse-power to this exceptionally favored location at Richmond.

A fair valuation of the average horse-power per year is stated as $28, and it follows, therefore, that Richmond could easily control an annual valuation for her contiguous water-privileges of over one million dollars—a dormant value to a large extent at present, but one so clearly within the scope of engineering talent and science, and so essential to the future grand development of the city, that only the means need now be secured to insure the successful inauguration of plans and operations already outlined or completed. It may seem very strange to engineers whose attention is but just now called to these facts that operations on a gigantic scale have not already been inaugurated with a view to distribute evenly the immense water-power of the James River, a work which would do so much to place the city of Richmond in her rightful position as the foremost manufacturing centre of the Southern States.

Geographically, Richmond is at the end of the deepest western indentation of the Atlantic, 127 miles from the sea and 74 miles in a direct line from Newport News, on Hampton Roads.

It commands a far wider territorial area within the limits of the United States than any city north of it. Thus, all points south of a straight line projected from the mouth of the Delaware River to the Lake of the Woods are nearer to Richmond than to New York. It is 241 miles nearer to San Francisco than New York. It is nearer to Chicago than New York. It is 130 miles nearer St. Louis than New York. It is 197 miles nearer to Louisville than Baltimore, 218 miles nearer than Philadelphia, and 308 miles nearer than New York. It is now the focus of seven railroads from south, north, east, and west, with their widely ramifying connections, namely: The Atlantic Coast Line, the Richmond & Danville, the Richmond & Alleghany, the Richmond, Fredericksburg & Potomac, Chesapeake & Ohio, with Newport News extension, and the Richmond & York River road. As nearly all these roads reach out to the West by shortest distances, so they come by routes of lower grade and through more temperate climes to James River, which is without question the best outlet for them all.

Of this distance of 127 miles by river, as stated, from Richmond to the Capes, 82$\frac{3}{8}$ thereof has the requisite depth of 22 feet at mean low tide—generally very much more, and with no bar at the mouth of the river, as there is at the

CCCCXXXIV.*

AN ENGINEERING PROBLEM AT RICHMOND, VA.

BY W. H. ADAMS, NEW YORK.

(Member of the Society.)

THE Southern States are well launched on a tide of prosperity, resulting from a business development of natural resources, which is attracting the attention of engineers from all climes and from nearly every branch of the profession. They have in their special work already revolutionized many of the seemingly fixed ideas of the century, and it is daily becoming manifest to careful observers that nothing can stay the progress of events which shall place the control of leading staple products almost entirely in the hands of Southern capitalists, and within a comparatively short time.

The multitudinous industries which are springing up on every hand, and in directions only appreciated within the last few years, depend for their well-being greatly, and in some cases wholly, upon such lines of transportation as find their terminals upon the Atlantic coast, or upon the rivers which lead to the ocean. Everywhere throughout the South railroads keep pace with the advancing wave of development, and combinations are constantly being formed from integral parts of the inland systems of railway, which are intended to secure to them the advantages of natural outlets on tide-water.

There is, however, a manifest lack of comprehensive measures for developing the outlets on the Atlantic to the enormous growth of the interior trade, and while, perhaps, this can be easily shown to be due almost entirely to local circumstances and conditions which have prevailed during the past twenty years, yet, in view of the surprises in store for us when the facts of the late census are gathered, and the changes are admitted sure to take place in the next decade, we should be quick to study and comprehend any scheme of improvement, now pre-

---

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

until the development of iron and coal industries sent Liverpool to the front, the spring-tide in the Avon rises to a height of 40 to 48 feet.

Comparison in natural resources with the enormously wealthy and progressive inland cities of England, or of our own country, will not show to the disadvantage of Richmond; and, if possibilities are to be considered, there is more in the future growth of this inland centre of trade—with its mineral and agricultural wealth so closely allied, the combination of complex problems in water-storage and delivery, utilization, etc.—than can be grasped at a sitting.

I cannot do better than quote freely from the able articles which have been written by Mr. Burgwyn, on the special subject of the inland basin to be located on the west bank of the James River at Richmond.

"There are few localities in the world where the water-facilities or frontage are better. While the present official plan is to obtain 22 feet at low water, it is quite practical, within reasonable limits as to cost, to obtain a channel 30 feet deep at low water from the sea as far as Drewry's Bluff. Arriving at Drewry's Bluff, the project is to have lock-gates, and a basin and canal fed by water from the James River. As a series of basins could be constructed at a minimum cost by the placing of the excavated materials upon the levees between them and the river, and as the level of the water in the canal at Manchester is just about what is needed for the elevation of the water in the grand basins above low tide in the river, it follows that the construction of such basins are feasible, are economical, and are possessed of many advantages. Thus the water discharged from the opening of the locks can be utilized in charging hydraulic accumulators, storage batteries, and for other purposes. The level of the water in the basin being higher than the river, dry-docks can be constructed which will empty themselves by gravity, while the water in running out can be utilized for many purposes. The water in the high-level basins will also afford a cheap power for manufacturing purposes. Around such basins, elevators can be constructed with the necessary trackage. Cotton compresses, coal pockets, etc., etc., can also be provided for. Back of these basins model homes can be constructed for artisans, mechanics, and laborers, and upon the level plateau above all would be a city of magnificent size, discharging its sewage under the basins

150 miles, equal to 6,000 square miles of territory—are not only practicable from the nature of the bottom lands of the upper country, but inexpensive (considering the advantages to be gained) as compared with public improvements of like nature in other parts of the world. The measured discharge of the James River, taken at Richmond under normal conditions, varies from 2,000 cubic feet per second to 10,000 cubic feet per second—a fair average fall of, say, 4,000 cubic feet per second. Freshets are of a character difficult to understand without close study of the topography of the country about Richmond, but at such periods the estimated discharge of water past the city of Richmond is as high as 200,000 cubic feet per second. To grasp fully the economic value of such a volume of water as is here obtainable, if evenly distributed throughout the season and made use of by industrial cities along the river, it should be understood that within thirteen miles from the city (Richmond being at the head of navigation and on tide-water) there is a fall of over 115 feet, and from tide-water to the gap in the Alleghanies, from which the James River takes its source, the elevation is about 585 feet. By any reasonable system of impounding reservoirs within a distance of 50 miles of the city there could be guaranteed at least 4,000 cubic feet per second during summer months, and under a head of 115 feet this would average, say, 40,000 H.P. To enable the reader to compare this measured value with well-known sections where the question has been answered by the expenditure of millions of money to secure the advantages of power of this character, the following table of the horse-power for the summer months alone has been compiled:

City.	*Horse-power.*
James River at Richmond, Va., with reservoirs	40,000
James River at Richmond, Va., at present	25,000
Holyoke, Mass.	12,260
Manchester, N. H.	12,000
Lowell, Mass.	11,845
Lewiston, Me.	11,000
Lawrence, Mass.	10,902
Cohoes, N. Y.	6,556

It is thus evident that the ordinary summer conditions compare favorably with the well-known examples above, while great freshets furnish over two million horse-power to this exceptionally favored location at Richmond.

tracks. It is estimated that the cost of the entrance lock and the 8 dry-docks would amount to $3,245,000, the excavation of the interior basin would amount to about $750,000, and the interior wharfage line to about $250,000, and the outside wharf-line $430,000. There are about 26 miles of track, which, including the switches, etc., would cost about $128,000. The buildings, etc., would be a cost determined upon the machinery required. It is to be observed, however, that almost all of this construction could be done *seriatim*, and therefore be developed as the commerce or necessity requires. The interior basin here shown lies between Manchester and Goode's Creek. The next one would be along the river between Goode's Creek and the bluffs at Warwick, and the third basin would lie between the bluffs at the old town of Warwick and Drewry's Bluff.*

"The total acreage in these three basins would amount to over 1,500 acres, and the wharfage frontage about 35 miles. When one compares these possibilities here with the data of existing works in the acompanying tables, he would be astounded at the advantages of this location and be convinced more than ever that 'Chesterfield County is as richly endowed by nature as any place in the world.'"

TABLE SHOWING NUMBER OF VESSELS OF GIVEN TONNAGE ACCOMMODATED PER ACRE OF DOCK AREA.

Tonnage.	No. of Vessels per Acre.	Tonnage.	No. of Vessels per Acre.
100	14.0	350	6.9
150	10.6	400	6.5
200	9.0	450	6.2
250	8.0	500	6.0
300	7.3		

* See Plate, Fig. 176.

mouth of the Mississippi, the Danube, and the Nile. There is, however, in the magnificent harbor of Hampton Roads an entrance which ranks the James River with the great harbors of the world, like the Amazon. The river itself is one continuous harbor, reaching into the country 98 miles—the upper 50 miles being fresh water.

This whole distance of 98 miles consists of long, deep, capacious pools, separated by shoals, which are now being systematically worked upon to produce a uniform depth of 22 feet at low water. (See Plate, Fig. 174.)

With such a depth, Richmond can compete with any of the Northern cities, as she will soon be 67 miles nearer the sea than Baltimore, but 17 miles farther than Philadelphia.

New York has now 21 to 25 feet at low tide, with a mean rise of 4 feet 3 inches = 29 feet 3 inches.

Philadelphia has on her bar 19 feet at low tide, with a rise of 6 feet = 25 feet.

Norfolk has at her wharves about 25 feet at low tide, with a rise of 3 feet = 28 feet.

Baltimore has 27 feet at low tide, with a rise of 1½ feet = 28½ feet.

Richmond is promised 22 feet at low tide, with a rise of 3½ feet = 25½ feet.

Some fears have been expressed that the floods in James River might be fatal to ship-building, from the overflow of the banks.

Twice within the last one hundred years, to wit: in 1870 and 1877, the flood rose in one case to 24, and in the other to 25, feet above mean high tide. But twice daily the Clyde rises 10½ feet; the Tyne, 15 feet; the Thames—at Sheerness—16 feet; at London Bridge, 21 feet 7 inches; and the Mersey, at Liverpool, rises 21 feet—or several feet more, twice daily, from ordinary tides than the James does in a year from only occasional freshets. Though the spring-tides of the Mersey rise ordinarily to 26 feet, and at the equinox to 31 feet, or six feet higher than our rarest freshets have ever risen, and on the Thames nearly as high, yet London has grown to be the largest city in the world, and Liverpool to be the first in commercial importance in Great Britain.

At Bristol, which was the foremost sea-port in Great Britain tracks. It is estimated that the cost of the entrance lock and the 8 dry-docks would amount to $3,245,000, the excavation of the interior basin would amount to about $750,000, and the interior wharfage line to about $250,000, and the outside wharf-line $430,000. There are about 26 miles of track, which, including the switches, etc., would cost about $128,000. The buildings, etc., would be a cost determined upon the machinery required. It is to be observed, however, that almost all of this construction could be done *seriatim*, and therefore be developed as the commerce or necessity requires. The interior basin here shown lies between Manchester and Goode's Creek. The next one would be along the river between Goode's Creek and the bluffs at Warwick, and the third basin would lie between the bluffs at the old town of Warwick and Drewry's Bluff.*

"The total acreage in these three basins would amount to over 1,500 acres, and the wharfage frontage about 35 miles. When one compares these possibilities here with the data of existing works in the acompanying tables, he would be astounded at the advantages of this location and be convinced more than ever that 'Chesterfield County is as richly endowed by nature as any place in the world.'"

TABLE SHOWING NUMBER OF VESSELS OF GIVEN TONNAGE ACCOMMODATED PER ACRE OF DOCK AREA.

Tonnage.	No. of Vessels per Acre.	Tonnage.	No. of Vessels per Acre.
100	14.0	350	6.9
150	10.6	400	6.5
200	9.0	450	6.2
250	8.0	500	6.0
300	7.8		

* See Plate, Fig. 176.

into the river, when it would be carried off without further trouble or cost.

"Careful consideration is called to the accompanying maps, where it is shown in detail how such a plan is feasible, how readily it can be carried out, and what a legacy nature has bestowed upon the locality. By an examination of the profile of the channel of the James River, it will be seen that the carrying of the depth to 30 feet at low tide will, in almost all instances, merely necessitate the cutting through of mud bars. The mud in these bars is so soft that iron rods have been run into them, sinking from 15 to 20 feet into the mud, simply from their own weight. The procuring of a depth of 30 feet at low tide, instead of 22 feet, the present plan, would require the removal of only about 29,000,000 cubic yards of material. By skilfully locating brush dikes, these channels, once obtained, can be maintained at little cost on account of the scouring action of the tidal currents. With high-level basins, as above described, and with a highway to the sea of 30 feet at low tide, it is held, as proved, that few localities surpass the County of Chesterfield in its physical location as regards deep-water transportation. It is stated that at the Panama Canal the machines have moved, since the 1st of January, 1888, material at the rate of a million and a half cubic yards per month. At this rate it would take less than two years to make the excavation. Material of this nature in the vicinity of Norfolk has been let to contract as low as ten cents per cubic yard. It is believed that this extra depth can be procured at a cost of six million dollars.

"Comparing this work with others, it has been stated that the cost of the improvement of the Clyde was $47,845,000, and that of the Tyne was $17,280,000. In the latter instance 40,000,000 cubic yards were removed, but the material was much harder than the soft mud of the James, and the dikes were of stone instead of brush, as is proposed here.

"One of these basins is shown in detail on the map.* It contains about 400 acres of land. The dockage or wharf-length is about 9 miles, and it contains 8 dry-docks, which can empty themselves by gravity. Two of these dry-docks are sufficiently large to take in any vessel except, perhaps, one of the size of the *Great Eastern*, and the remaining six are of smaller, graduated sizes. Each dry-dock can be entered by itself and has its own railroad

* See Plate, Fig. 175.

TABLE SHOWING SIZE OF VESSEL THAT CAN ENTER A DOCK OR CHANNEL OF GIVEN DEPTH (STEPHENSON).

Tonnage.	Draught.	Tonnage.	Draught.	Tonnage.	Draught.	Tonnage.	Draught.
50	7.8	250	13.5	500	15.6	1,300	21.4
60	8.4	300	14.5	600	16.5	1,400	21.9
70	8.8	350	15.1	700	17.4	1,500	22.5
80	9.2	400	15.8	800	18.2	1,600	23.0
90	9.6	450	16.4	900	18.9	1,700	23.4
100	10.0			1,000	19.6	1,800	23.9
150	11.4			1,100	20.2	1,900	24.3
200	12.5			1,200	20.8	2,000	24.9

DOCKS IN THE PRINCIPAL CITIES OF ENGLAND.

Ports.	No.	Area in Acres.
London	28	350
Liverpool	38	206
Birkenhead	4	142
Bristol	4	79
Hull, exclusive of timber pounds	7	46½
Great Grimsby	2	51
West Hartlepool, exclusive of timber pounds	8	82
Hartlepool	1	20
River Wear	2	41
River Tyne	4	107
Leith	3	15½
Dundee	4	34
Aberdeen	1	35

COMPARISON OF DOCKS.

	Length of Dock.		Breadth.		Depth over sill at High-water Springs.	
	Ft.	In.	Ft.	In.	Ft.	In.
Leith (East)	160	..	36	..	18	5
Dundee (Victoria)	230	..	60	..	21	..
Aberdeen (Victoria)	250	..	60	..	21	..
Dublin (large dock)	180	..	36	..	18	..
Cork	180	..	45	..	18	6
Bristol (Cumberland)	260	..	54 & 67		30	..
Plymouth	250	..	55	..	24	..
Newport	225	..	61	..	25	..
Cardiff	152	..	36	..	19	..
Swansea	165	..	56	..	21	6
Ipswich	150	..	45	..	16	6
Hull (Humber)	158	6	42	..	24	..
Great Grimsby	300	..	70	..	26	..
Goole (railway)	216	..	58	..	19	..
Goole (barge dock)	72	6	19	6	9	..
Middleborough	132	..	30	..	19	..
London (Victoria)	326	6	80	..	28	..

TABLE SHOWING DIMENSIONS OF DIFFERENT DOCKS.

Cities.	Length.	Breadth.
	Feet.	Feet
Sunderland	645	147
Leith (East)	250	100
Leith (Victoria)	238	200
Aberdeen	950	175
Dublin (large)	217	100
Galway	239	193
Limerick	270	130
Bristol (Cumberland)	245	90
Plymouth	420	150
Newport	270	73
Swansea	760	80
Great Grimsby	600	167
Hull (Victoria)	480	126
Goole (barge)	290	50
Goole (ship)	234	67
Goole (steamer)	120	164
Middleborough	400	130

In conclusion it need only be stated that if we enlarge the scope of the possible utilization of the water-power of the rivers within 20 miles of the city of Richmond, it will be found that this single district compares favorably in water frontage and power with many of the Northern States in their entirety, and it will strike engineers as peculiarly favorable that so certain a future, so grand a possibility as the unity of all interests, which shall eventually join to make up an industrial centre at this locality, will be planned and executed from original premises, and not hampered by ill-advised attempts of a former generation.

What has here been said as regards the advantages of the river frontage on the west bank, specially reported on by Mr. Burgwyn within the past year, is applicable to the territory which borders on the east bank of the James, nearer to the city.

Taken as a whole, there is no grander basis for the exercise of talent and energy in development of natural resources than can here be found, and the general outlines are presented to the Society of Mechanical Engineers with the assurance that through them there will be such interest excited in the subject as will eventually lead to the solution of so gigantic a problem.

During the years which shall come to us in celebrating the Columbian anniversary, there will be opportunities offered for presentation of the great engineering problems which are to be solved in this country during the next fifty years, among them the one herein brought to notice of the Society for the first time. Whether the subject be taken up from the special commercial basis, as clearly shown, or on broader lines, which shall make the work a national one, no engineer, whatever his training, can look upon the virtually unoccupied territory, note the possibilities of a near future which shall see the vast agricultural and manufacturing regions of the West and South brought into closer contact with this centrally land-locked sea-port city, without feelings of pride that so much, in a strictly engineering sense, has now to be accomplished, from original premises, with modern ideas, and with modern means at command.

TABLE SHOWING DIMENSIONS OF DIFFERENT DOCKS.

Cities.	Length.	Breadth.
	Feet.	Feet
Sunderland	645	147
Leith (East)	250	100
Leith (Victoria)	233	200
Aberdeen	950	175
Dublin (large)	217	100
Galway	239	193
Limerick	270	130
Bristol (Cumberland)	245	90
Plymouth	420	150
Newport	270	78
Swansea	760	80
Great Grimsby	600	167
Hull (Victoria)	480	126
Goole (barge)	290	50
Goole (ship)	234	67
Goole (steamer)	120	164
Middleborough	400	130

In conclusion it need only be stated that if we enlarge the scope of the possible utilization of the water-power of the rivers within 20 miles of the city of Richmond, it will be found that this single district compares favorably in water frontage and power with many of the Northern States in their entirety, and it will strike engineers as peculiarly favorable that so certain a future, so grand a possibility as the unity of all interests, which shall eventually join to make up an industrial centre at this locality, will be planned and executed from original premises, and not hampered by ill-advised attempts of a former generation.

What has here been said as regards the advantages of the river frontage on the west bank, specially reported on by Mr. Burgwyn within the past year, is applicable to the territory which borders on the east bank of the James, nearer to the city.

Taken as a whole, there is no grander basis for the exercise of talent and energy in development of natural resources than can here be found, and the general outlines are presented to the Society of Mechanical Engineers with the assurance that through them there will be such interest excited in the subject as will eventually lead to the solution of so gigantic a problem.

During the years which shall come to us in celebrating the Columbian anniversary, there will be opportunities offered for presentation of the great engineering problems which are to be solved in this country during the next fifty years, among them the one herein brought to notice of the Society for the first time. Whether the subject be taken up from the special commercial basis as clearly shown, or on broader lines, which shall make the work a national one, no engineer, whatever his training, can look upon the virtually unoccupied territory, note the possibilities of a near future which shall see the vast agricultural and manufacturing regions of the West and South brought into closer contact with this centrally land-locked sea-port city, without feelings of pride that so much, in a strictly engineering sense, has now to be accomplished, from original premises, with modern ideas, and with modern means at command.

CCCCXXXV.*

AUTHORITIES ON THE STEAM-JACKET; FACTS, AND CURRENT OPINION.

BY R. H. THURSTON, ITHACA, N. Y.
(Member of the Society, and Past President).

"J'ai déjà dit que les effets de l'enveloppe, ou chemise à vapeur de Watt, par exemple, avaient été alternativement prouvés ou niés sans que le public parvint à savoir à quoi s'en tenir sur la réalité des choses."†—*Hirn.*

THE discussion of a paper on the Compound Engine, and especially on the importance of the steam-jacket in that connection,‡ and a somewhat extensive correspondence to which it incidentally gave rise, has revealed such unexpected differences of opinion as to the value of the jacket, and its relative efficiencies under different circumstances in ordinary practice, that the writer has been led to make a somewhat extended examination of the authorities on this subject, so far as they exist, and to obtain from those who have had more extensive experience than himself their ideas of the character and extent of the benefit to be derived from its application under the usual conditions of steam-engine construction and operation. The result has been to throw some light on the subject, but yet to show such uncertainty and diversity of view where concordance was expected as to make it seem probable that a summary of current opinion might prove a very useful and instructive introduction to a more formal discussion of the philosophy of the steam-jacket; in which discussion it may be endeavored to show that a correct theory of the action and effects of the jacket will probably very largely reconcile these apparently conflicting opinions by exhibiting the fact that this accessory may, under different circum-

---

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

† *Théorie Mécanique de la Chaleur*, 876, vol. ii., p. 81.

‡ *Transactions American Society Mechanical Engineers*, New York meeting, November, 1889; *The Philosophy of the Multiple-Cylinder Engine*, No. CCCLXII.

stances, prove a decided advantage, an unimportant appurtenance of the engine, or even, under exceptional conditions, deleterious to efficiency.

In the paper to which reference is here made, it was remarked:

"*Steam-jacketing* is a common partial remedy for waste. By surrounding the steam-cylinder with the steam-jacket, it is possible to produce, in part, the effect of superheating; that is, to secure drier steam in the engine throughout the stroke. The amount of reëvaporation during the period succeeding cut-off and up to the closure of the exhaust-valve, and the quantity of heat of which the cylinder is thus robbed, measures the amount of initial condensation and waste and fixes the weight of steam which must be supplied, in excess of the thermodynamic demand, to compensate that loss. The effect of the addition of a steam-jacket depends upon the conditions of operation of the engine, largely, and may be productive of marked advantage, or, under unfavorable conditions, of no important useful effect. . . . High-speed engines derive less advantage from its application than slow-moving machines; and compound or multi-cylinder engines are less dependent upon it for economy than are simple engines. The saving effected in ordinary cases,* by its use, may be taken as averaging about 20%; and about the same gain is attained by effective superheating within the usually practicable range."

. . . .

"We are interested in the answer to the question: To what extent and in what manner is the jacket advantageous in the compound or multi-cylinder engine? Authorities disagree, even where they have themselves had large practical experience. It is sometimes advised to jacket only the high-pressure cylinder, sometimes to jacket only the low-pressure cylinder; and sometimes to jacket the whole series, whether one, two, or three, or more. The philosophy of the multi-cylinder engine, as above outlined, would obviously indicate that, to secure maximum good effect, assuming the jacket on the whole desirable at all, the best system is the latter; and that, since the waste of the engine is measured by the waste of its most wasteful member, to omit the jacket from any one cylinder insures that the aggregate loss of heat in the whole engine will be increased by

* In simple mill engines, is here understood.

at the initial temperature of the steam," * thus paraphrasing Watt's earlier statement.

The following quotations and abstracts of papers, collected by the writer, represent but a fraction of the material which he has gathered in the course of a very extensive correspondence and research among files of periodicals and transactions of societies. They will, however, give a fair idea of the character of evidence at present available on this subject. Should time and opportunity permit, a later paper may be given in which the philosophy of the subject may be more systematically discussed. The appendix contains an enormously valuable collection of data reported by Mr. Donkin to a committee of the British Institution of Mechanical Engineers, appointed especially to study this subject, and which is published most opportunely for our purpose.

That Watt understood the phenomena of cylinder-wastes better than has been commonly supposed, and better than many later writers and practitioners, is evident from a perusal of some of his correspondence. Thus, he says, in a letter to Dr. Small in 1769:

"I have misimproved the cylinder by making the bottom of cast-iron, which . . . does not communicate heat fast enough to evaporate the water left in the cylinder the first stroke, without being considerably cooled by it, and consequently condensing as much the next. I have also done wrong in removing the outer cylinder, which kept the inner one always surrounded by steam, ready to supply any loss of heat that might happen by evaporation from the inside surface."

He proceeded to correct these mistakes, and later, writing to Dr. Roebuck (1769), he explains his ideas more fully. He says:

"At first, when the cylinder is cold, there is a small quantity of water left in the cylinder. Immediately on producing the vacuum, the water is converted into steam, and thereby cools the cylinder considerably. Now, if there is an external cylinder, and the bottom of the internal one is thin, the steam on the outside will warm it again instantly, and no steam will be condensed the next time. (This was the case in my last model.)"

Watt probably had not more definite ideas of the theory of the jacket; but he was firmly convinced, by his own experience and by direct experiment, of its value. He usually adopted very

* *Railway Machinery.* *Vide*, also, *Development of the Philosophy of the Steam-engine*, R. H. Thurston; N. Y., J. Wiley & Sons, 1889, p. 83.

It is necessary to keep the cylinder "always as hot as the steam which enters it."*

This he sought to do by the use of his greatest invention, the separate condenser; but he then went further and enclosed the cylinder in an exterior vessel in which was confined steam at the pressure and temperature of the boiler.

In describing the invention, as covered by his patent of 1769, he says: †

"My method of lessening the consumption of steam consists in the following principles:

"(1) That the vessel in which the powers of steam are to be employed to work the engine—which is called the 'cylinder' in common fire-engines, and which I call the 'steam-vessel'—must, during the whole time that the engine is at work, be kept as hot as the steam which enters it; first, by enclosing it in a case of wood or any other materials that transmit heat slowly; secondly, by surrounding it with steam or other heated bodies; and, thirdly, by suffering neither water nor other substances colder than the steam to enter or touch it during that time."

He then goes on to enunciate other principles, and his now familiar method of accomplishing his objects.

From that day to the present time, the steam-jacket has continued in more or less general use on the more economical types of engine; and the Cornish engine, the modern representative of the first engine of James Watt, invariably exhibits its use, precisely as first employed by the great engineer; while the reason of its use has now become generally understood by well-read engineers. Writing eighty years after the above date, Daniel Kinnear Clark, the first among later writers to describe clearly and fully the methods of waste of heat in the heat engines, says: "To prevent entirely the condensation of steam worked expansively, the cylinder must not only be protected by a non-conductor; it must be maintained, by independent external means,

---

* *History of the Growth of the Steam-engine*, R. H. Thurston, p. 87. N. Y., D. Appleton & Sons, 1878.

† Ibid., p. 99. A curious thought of Watt is expressed in the last of his stated "principles," viz.:—"Sixthly. I intend, in some case, to apply a degree of cold not capable of reducing the steam to water, but of contracting it considerably, so that the engines shall be worked by the alternate contraction and expansion of the steam."

gave shape to his ideas, as time went on, publishing them in 1845,* and, later, in 1863–67.† He says in his first paper, just mentioned: "The utility of the jacket, or rather that of heating the cylinders of steam-engines from the outside, . . . is rendered unquestionable, both by direct experiment and by detailed observation of the phenomena characterizing the action of steam in the cylinder, and the logical discussion of these observations." "Jackets have not for their main result the maintenance of the temperature of the steam during expansion; their use consists in the prevention of refrigeration of the walls of the cylinder while in communication with the condenser"—probably the first exact statement of this effect ever printed.‡ He even anticipates Rankine and Clausius in one of their most famous discoveries, saying: "*La vapeur d'eau, à l'état de saturation et entièrement sèche, se dilatait sans addition ni soustraction de chaleur; et nous avons montré que l'expansion est alors accompagnée d'une liquéfaction partielle de vapeur. C'est à peu près ainsi que les choses doivent se passer dans les machines à vapeur ordinaires.*" He then goes on to describe very clearly the phenomenon of "cylinder-condensation;" but in his later works he seems to have paid less attention to this action, and may not have fully realized its importance; but his conception of the processes involved in such wastes, and in the preventive action of the jacket, was exact and well expressed. This is Carnot's principle.

In the once well-known experiments of M. Combes (1843) a Farcot engine was employed under a very light load and with a very high ratio of expansion—about twenty—the trials with and without steam in the jacket occupying three or four working days each. The result was a gain of 36.4% by the use of the jacket, the steam-consumption falling from 40.40 to 30.15 lbs. per horse-power per hour.§

Mr. Crampton found a compound engine to give a gain of 30% in power under similar conditions.‖

In a little book written by Mr. Gill, as early as 1844, we find a very good explanation of the theory of cylinder-wastes and the source of the economy produced by the jacket. He says: "If

* *Traité d'Exploration des Mines.*

† *Principes de la Théorie Mécanique de la Chaleur.*

‡ Memoirs of 1843, p. 245.

§ Cornut on Steam-jackets; translated by B. F. Isherwood, U.S.N. *Engineering and Mining Journal*, Jan. 26, 1878.

‖ Ibid.

low steam pressures, and a very small ratio of expansion, and these could not have been conditions very favorable to its efficiency; yet he always employed it. It came into disuse later, except on Cornish engines, where, with steam at about 40 pounds and expanding about four times, with a peculiar construction and movement of engine especially favorable to its action—the piston making its steam-stroke rapidly and halting for a time before change of motion—the usefulness of the device was easily seen, even under otherwise unfavorable conditions. But, as remarked by Mr. Isherwood, "the fact that the economic efficiency of steam-jackets depended upon the regimen of the steam was not understood at that time; nor was any attempt made to ascertain why the jacket, with one type of engine and one regimen of steam, gave results differing from those given by another type of engine using another regimen of steam."*

About 1860, as stated by the same authority, economy of fuel became recognized as a controlling consideration in the British navy, and high expansion-ratios and steam-pressures, with jacketing and superheating, came into vogue, and the value of the jacket became once more recognized. Superheating was given up in consequence of its injurious effect on the engine; but jacketing engines remained standard practice at sea. Nearly all makers invariably use the jacket.

Watt says of his attempt to dispense with the steam-jacket: "When we tried to lay aside the jacket, we had no reason to applaud our economy; for the consumption of fuel was considerably greater."†

In 1830 Dr. Haycraft determined the quantity of steam passing through a steam-engine, using a surface condenser to effect its measurement, and found it to exceed by 30% the amount shown by the indicator at the point of cut-off. He attributed this excess wholly to "priming" at the boiler, to which source it may probably have been in part correctly ascribed.

Combes, in papers presented to and published by the *Académie des Sciences*, was probably the first to introduce into the theory of the steam-engine the consideration of that phenomenon discovered by Watt, to check the wasteful effects of which the latter invented the steam-jacket.‡ That author gradually

---

* Correspondence.

† Stuart's *Anecdotes of the Steam-engine*, London, 1829, vol. i., p. 314.

‡ *Comptes-rendus*, 1843.

As a result of his first series of experiments he was able to say : "The influence of the steam-jacket is now clearly explained: It consists in preventing the steam from partially condensing and thus lessening the pressure during expansion by that act itself. As the heat taken from the jacket is, as has been seen, a small fraction of the total heat expanded, the power so gained costs very little. Were any doubt now to exist on this point, the following facts would completely remove them :

"(1) When the engine is working with the jacket in use, if we suddenly shut off the steam and take it directly to the cylinder, the engine continues to work as before for some time, as if nothing had happened. The indicator diagrams are precisely the same as before ; it is only after ten or twenty minutes that the power of the engine falls off 23½%, in this case. It is thus evidently the heat in the walls of the cylinder, and not the simple drying of the steam, which gives us the economy of 23½%.

"(2) The jacket actually modifies very sensibly the temperature of the steam ; for, while it is acting, the steam exhausted into the condenser is at 64° C., at a tension of 0.075^{m} ; while in the other case the temperature falls to 58°, although its tension rises to 0.095^{m}. . . ."

Further on he says : "Since it is the elevation of temperature of the walls of the cylinder, heated by the steam in the jacket, which is the cause of the improvement, it is not to be doubted for an instant that any means of securing such temperature will be equally effective and economical." He then proposes the use of a smoke-jacket ; but he finds, on trial, that it is of little value, the heat being incapable of passing with sufficient rapidity from the gases in the jacket to the metal of the cylinder.

Hirn, in this memoir, also expressly proposed the measure of the heat consumed by the engine as the true measure of its efficiency.

In explaining "the utility of the steam-jacket, an appendage which, though it has always been found highly useful in practice, has nearly with the same uniformity been pronounced useless by writers on the steam-engine," Mr. J. G. Lawrie writes :

"Each of these causes," already described, "produces a considerable quantity of water which, when the cylinder is unjacketed, is passed, together with the heat which it contains, to the condenser ; while in a jacketed cylinder the water is produced, the cylinder be supplied with dry steam, and no heat is dissipated by radiation, there will still be a loss of heat in the cylinder occasioned by the sudden expansion of the steam when the communication with the condenser is opened." . . . "As the heat for evaporation is furnished by the hot metal of the cylinder, piston, etc., such heat must be returned to them by the condensation of steam during the succeeding stroke, such condensation and evaporation going on until an equilibrium is established." He suggests superheating as the best remedy.*

Mr. Isherwood calls the attention of the writer to the fact that as early as 1848, M. Farcot, of Paris, was building engines "of admirable designs, with steam-jackets on both cylinders, boiler-pressure about 50 lbs., and giving the indicated horse-power for from 17 to 18 lbs. of water per hour." "Farcot was a first-class engineer in every sense, scientific as well as mechanical. Few engineers as accomplished have ever lived. His designs were as remarkable for their elegance as for their propriety." †

Hirn published his "*Mémoire sur l'Utilité des Enveloppes à Vapeur*" in 1855.‡ This memorable paper gives us the first precise analysis of experiments showing the quantitative measures of the thermal action of the walls of the steam-cylinder. It presents an exact and scientific treatment of the case, and gives indisputable measures of the quantity of heat transferred to the metal, and restored to the steam when too late for transformation into its proportion of mechanical energy. In every experiment, Hirn measured the quantity of water entering the boiler and there converted into steam, and compared it with the quantity of steam found, at each step in the engine-cycle, in the cylinder. He even went so far as to determine, by the use of his calorimeter, the quality of the steam entering the engine, in order that his measures of that contained in the cylinder might not be rendered uncertain by the action known as priming or foaming at the boiler. He also, for the first time, determined the weight of water leaving the condenser, and its temperature, thus securing the elements for the method of computation now known as that of Farey and Donkin. He proposed no theory, believing, as he stated expressly, that, at the time, any formulation of a theory was impossible without further knowledge.

* *Improvements of the Steam-engine*, Weale's paper, Jan., 1844.
† Correspondence.
‡ *Bulletin de la Société Industrielle de Mulhouse*, t. xxvii. pp. 105–206.

der," referring, however, solely to the condensation of steam doing work.* He further says: "Accordingly, in all cases in which steam is expanded to more than three or four times its initial volume, it has in practice been found advantageous to envelop the cylinder in a steam-jacket." †

Rankine states that "in a few examples in which the small cylinder alone is jacketed, the liquefaction is found to be prevented." ‡ Hirn confirmed this view, at least as regards effective jacketing, by his experiments with a glass steam-cylinder, in which the vapor became misty when unjacketed, but remained constantly clear when the jacket was in use.

Professor Rankine, discussing Isherwood's famous experiments on the United States steamship *Michigan*, compares the products of area of surface into point of cut-off into time of action of the entering steam, with the quantities admitted, without definite result, but gives a correct explanation of the process of condensation. He, however, still attributes the initiation of the process to the quantities of steam condensed in consequence of heat transformation during the expansion of the preceding stroke, a quantity which he shows to be "exceedingly small." § He states the supply of external heat needed to prevent initial condensation may be obtained "either by means of a steam-jacket, or of a flue, or by superheating the steam which is admitted."

Rankine's last conclusion in this matter was that by the use of a jacket or other expedient it is possible "to diminish the loss of efficiency arising from the conducting power of the cylinder and piston until it becomes unimportant." ‖ He refers to a form of compound engine which he asserts to be "defective through the absence of steam-jackets, which are now known to be essential to the realizing of the economy properly due to high rates of expansion," and states that the history of the marine engine had shown "that the abandonment of the steam-jacket in the practice of all marine engineers had made it useless, if not wasteful, to employ those high rates of expansion to which the compound engine is suited." He says that Mr.

* *Steam Engine*, § 286, p. 395.
† Ibid., p. 396.
‡ Ibid., § 286, p. 396.
§ Inst. of Engineers of Scotland, February 5, 1862.
‖ *Life of John Elder*, p. 17.

not in the cylinder, but in the jacket, and being passed, not to the condenser, but to the boiler, is economized." *

Mr. John Penn, as the outcome of long experience and many experiments, concludes that, "The plan of superheating the steam before entering the cylinder is a simple and eligible mode of attaining the desired object, and appears to be preferable to a steam-jacket." † He found a gain of fuel economy of "from 20% to 30% in marine engines," with moderate superheating, and believes it possible, in large engines, to thus secure dry steam throughout the stroke.

Mr. Cowper found that in an unjacketed engine the initial condensation amounted to from 12% to 44.5% with varying expansion, while, "when the cylinder had a steam-jacket supplied directly from the boiler, he found the actual indicator figure almost exactly corresponded with the theoretical figure." He thought it possible to practically suppress cylinder condensation.‡

The writer was engaged at about this time on designs for a pair of simple Greene engines, guaranteed by their builders, Thurston, Gardner & Co., to supply 1,000 H. P. on less than 2,000 lbs. of coal per hour, gross. The engines were built from Mr. Leavitt's specifications, and were steam-jacketed; steam at about 110 lbs. in the boiler, condensing, and had a ratio of expansion of about 10. They gave on trial a figure better than guaranteed, 1.89 lbs., and 20% better than it was then considered, without jacketing, ordinarily safe to expect.

Rankine, in 1858, among the several methods of securing superheat in working steam, mentions "superheating by the steam-jacket," and says that it "takes place when the steam-jacket communicates more heat to the expanding steam in the cylinder than is necessary merely to prevent any of it from condensing. It does not appear that this kind of superheating can be made the subject of a definite calculation."§ He recognizes clearly even at that early date that "the principal effect of the 'jacket,' or annular casing enveloping the cylinder, filled with hot steam from the boiler, which was one of the inventions of Watt, is to prevent that liquefaction of the steam in the cylin-

* *The Engineer*, London, Sept. 3, 1858, p. 169.

† *Proc. Inst. M. E.*, 1859, p. 197.

‡ *Proc. Inst. M. E.*, 1859, p. 204.

§ *Steam Engine*, § 295, p. 429.

der," referring, however, solely to the condensation of steam doing work.* He further says: "Accordingly, in all cases in which steam is expanded to more than three or four times its initial volume, it has in practice been found advantageous to envelop the cylinder in a steam-jacket." †

Rankine states that "in a few examples in which the small cylinder alone is jacketed, the liquefaction is found to be prevented." ‡ Hirn confirmed this view, at least as regards effective jacketing, by his experiments with a glass steam-cylinder, in which the vapor became misty when unjacketed, but remained constantly clear when the jacket was in use.

Professor Rankine, discussing Isherwood's famous experiments on the United States steamship *Michigan*, compares the products of area of surface into point of cut-off into time of action of the entering steam, with the quantities admitted, without definite result, but gives a correct explanation of the process of condensation. He, however, still attributes the initiation of the process to the quantities of steam condensed in consequence of heat transformation during the expansion of the preceding stroke, a quantity which he shows to be "exceedingly small." § He states the supply of external heat needed to prevent initial condensation may be obtained "either by means of a steam-jacket, or of a flue, or by superheating the steam which is admitted."

Rankine's last conclusion in this matter was that by the use of a jacket or other expedient it is possible "to diminish the loss of efficiency arising from the conducting power of the cylinder and piston until it becomes unimportant." ‖ He refers to a form of compound engine which he asserts to be "defective through the absence of steam-jackets, which are now known to be essential to the realizing of the economy properly due to high rates of expansion," and states that the history of the marine engine had shown "that the abandonment of the steam-jacket in the practice of all marine engineers had made it useless, if not wasteful, to employ those high rates of expansion to which the compound engine is suited." He says that Mr.

---

* *Steam Engine*, § 286, p. 395.

† Ibid., p. 396.

‡ Ibid., § 286, p. 396.

§ Inst. of Engineers of Scotland, February 5, 1862.

‖ *Life of John Elder*, p. 17.

Elder knew that the true remedy was "resuming the use of the steam-jacket."

Hirn states that thirty years before the date of his work * Combes had noted a gain of about 20% by the use of the jacket on the engines of that time. His own investigations began in 1854, and were promptly directed toward the solution of this interesting problem. He at once recognized the fact that the office of the jacket was not simply to check external radiation from the working cylinder, since its enlarged surface must inevitably exaggerate that waste. He found that the jacket produced a decided modification of the expansion line on the indicator diagram, proving the fact of transmission of heat into the mass of working steam, and a change in the law of change of pressure with variation of volume; producing thus a higher mean pressure and an increased quantity of work performed after cut-off. †

M. Hirn compares the performance of two simple engines, which, as he states, were in all respects precisely alike in dimensions, speed, cut-off, pressure, etc., and only differed in the fact that one was fitted with a jacket, while the other was not.‡ The result was the production of the horse-power on an expenditure of, respectively, $10^{kg}.5728$, and $8^{kg}.0617$ of fuel per hour; or a gain of 24% for equal work done. The gain was found, not in the actual diminution of the quantity of fuel and of steam consumed, but in an augmentation of the amount of work done by the engine.

He indicates that superheating is a more efficient method of reducing cylinder-wastes than is the use of the steam-jacket; which is not only itself a waste, but is usually inapplicable at all to the surfaces of the piston.

Hirn at once saw that he must make a detailed study of the action of the interior surface of the cylinder, the piston, and its rod, both when with and when without a jacket. In his succeeding work, in which he was aided by Messrs. Leloutre and Hallauer, he made his experimental investigation of the matter and published his results in his work of 1865–1870, indicating very exactly and clearly what were the actions and effects in detail of the phenomena, which had been, till then, apparently

* *Théorie Mécanique de la Chaleur*, 1876, vol. ii., p. 17.

† *Trans. de la Soc. Indust. de Mulhouse*, April 25, 1855.

‡ *Thermodynamique*, vol. ii., p. 39.

(2) Compound engine, two cylinders, without jackets.........	Loss,	26	per cent.
Same engine, with jackets.............................	"	20	"
(3) Simple engine, without steam-jacket.....................	"	18	"
Same engine, without steam-jacket, but with steam superheated 86° C....................................	"	12	"

M. Hirn thus reports the economy arising from the use of superheated steam as compared with saturated steam in the same engine:

(4) A compound engine working with superheated steam at a temperature of 210° C., an economy of 20%.

(5) An engine with one cylinder, working with superheated steam at 225° C., an economy of 31%.

(6) The same engine, with steam at 245° C., an economy of 47%.

Herr Heim reports to the German Society of Engineers the results of experiments to determine the economy to be derived by the addition of steam-jackets to various forms of steam-engine. He finds that a six horse-power portable engine, unjacketed, demanded an excess of 35% over the theoretical quantity of steam which should have been required to do the work; an eighteen-inch Wheelock engine required the same excess over the calculated quantity. Both were non-condensing. Condensing-engines experience a still greater loss due to internal "cylinder condensation." Engines expanding ten times demand 74% excess; when cutting off at one-fifth, 62%, and expanding three times, 55% more than the calculated amount when they are unjacketed. By adding a jacket, he concludes that the loss can be reduced to 64, 54, and 48%. The effect of increase of piston speed is similar to that of adding a jacket. An engine at three feet, and at seven feet piston speed per second, gave a record of loss amounting to 96 and to 70%. The addition of the condenser causes increase of this loss. A twenty-inch non-condensing engine, working at five atmospheres pressure, was provided with a condenser, and, while the power was increased 140%, the waste was increased from 42 to 62%. A hoisting-engine, working intermittently, exhibited a loss of 142%.

M. Hallauer considers that the jacket may aid efficiently in securing in the simple engine that high efficiency ordinarily found to be attainable only by compounding. He says:*

"Il est possible de construire une machine verticale à balan-

* *Moteurs à Vapeur*, Trans. Soc. Ind. de Mulhouse, Jan. 30, 1878.

action, the greater part of this heat is devoted to the increase of the work of expansion; when the jacket is shut off, the major part of this heat, on the contrary, is expended in the period of condensation, a pure loss. The remarkable fact is that this reversal of the proportion of heat utilized and wasted depends only upon the very small quantity of heat supplied by the jacket itself."

In practice the commercial efficiency of steam-engines is generally measured by the consumption of coal per I.H.P per hour, but this is not a good standard for comparison of different experiments, as the calorific power of coal varies according to its quality. A more scientific standard is formed by taking the weight of dry saturated steam used per indicated horse-power, or per effective horse-power, per hour. The following results of experiments by M. Hallauer are all reduced to those standards:

ENGINES WITH TWO CYLINDERS.

		Ratio of Expansion.	Consumption in Kilos per I.H.P.	Consumption in Kilos per E.H.P.
(1) Beam Engine,	Jacketed	13	7.9	9.1
Do.	Do.	7	8.2	9.3
(2) Horizontal Engine,	Do.	6	8.3	9.2
(3) Vertical Engine,	Do.	5	8.5	9.5

ENGINES WITH ONE CYLINDER.

(4) Corliss' Horizontal Engine.	Jacketed	6	7.9	8.6
(5) Hirn's Beam Engine, Superheated Steam at temp. 200° C.	Non-jacketed.	7	7.2	8.0

These results, according to Hallauer, show that superheated steam is more economical than saturated; in the second place that simple engines *can* be designed which are as economical in the steam used per I.H.P. per hour as well designed compound engines.

M. Hirn thus reports the effects of the different methods of reducing the loss due the action of the sides of the cylinders:

(1) Simple engine, without steam-jacket...................Loss, 33 per cent.
Same engine with steam-jacket.......................... " 21 "

surcharged with a large accession of moisture, was actually dried and somewhat superheated.*

Mr. Clark, from a study of available data to date (1877), concludes:

"For the development of the highest efficiencies of steam, as used in the steam-engine, the steam-jacket, or other means for protecting the steam from the cooling and condensing action of the cylinder, must be employed. The superheating of steam prior to its introduction into the cylinder is probably the most efficient means that may be employed for this purpose. The application to the cylinder of hot gases—hotter than the steam—is probably the next best means; and the next course the steam-jacket."†

The following table is given by Clark as representative of fair average practice up to the date of writing (1877), and indicates his judgment to be that the jacket, in these typical cases, may be expected ordinarily to afford a gain of about 15% in the simple and 10 or 12% in the compound engine.‡

Engine.	r.	Steam, pounds per I. H. P. per hour.
Simple ; condensing ; jacketed	6	19–20
" non-condensing ; jacketed ; steam 70 lbs.	4	24
" condensing ; unjacketed	3½–4½	24
" " steam superheated	4	24
" non-condensing ; unjacketed ; steam 90 lbs.		
" cylinder well protected and heated	3	21
Compound ; condensing ; jacketed.		
Receiver	6	18–20
Woolf	10	20–21
Compound ; condensing ; unjacketed.		
Receiver	..	...
Woolf	7	23

Mr. Clark concludes, after a discussion of Emery's experiments on the U. S. R. M. steamers, that: §

"For the single cylinder the use of the steam-jacket makes a great increase of efficiency in reducing the consumption of steam per horse-power, as against the absence of steam from the jacket. There is also a gain of efficiency by the same means with the

* *Railway Machinery*, 1856; *Manual for Mechanical Engineers*, 1877, p. 888.

† *Manual*, p. 888.

‡ *Manual*, pp. 889–890.

§ *The Steam Engine*, vol. i., p. 555.

cier, à un seul cylindre et quatre tiroirs, à enveloppe de vapeur, aussi économique au moins que le système Woolf pour l'emploi de la vapeur saturée."

He considers that the same engine supplied with steam superheated to 220° C. (428° Fahr.) would unquestionably give the same amount of work with a gain of 5 to 10%; since, as he asserts, without the jacket superheating gives it an advantage over the Woolf type of 14.5% nearly, in indicated power. He considers that the application of the steam-jacket to the Hirn simple engine, with saturated steam, gives at once a gain of at least 10%, placing it directly on a level with the Woolf compound, the ratio of expansion ranging between 5 and 7, and may even excel the latter, if condensing. The speed of this engine varied between twenty-five and seventy-five revolutions per minute.

M. Labouchère asserts: "L'utilité des enveloppes à vapeur à détente est parfaitement constaté aujourd'hui, et économies de 20 et 25 pour 100 résultent, le plus souvent, de leur emploi dans les machines où la détente de la vapeur est portée à trois ou quatre fois le volume primitif." *

Mr. L. E. Fletcher, in one of his annual reports, remarks: "I have been induced to confine my remarks to the use of steam-jackets and superheating, since I consider their general adoption by members the most important step they can take at present in the economic use of steam." †

Mr. D. K. Clark, the veteran engineer and pioneer in the systematic investigation of these wastes, says:

"By the application of a jacket of steam from the boiler to the cylinder, a material increase in the efficiency of the steam has been, in most circumstances, effected. But it is incontestable that the steam-jacket, though it diminishes, does not wholly prevent initial condensation of the steam admitted."

That by effective external protection and heating of the cylinder—such as sometimes occurs, for example, in inside cylinders of locomotives which are immersed in the smoke-box—cylinder condensation can sometimes be wholly prevented, is indicated by experiments reported by Mr. Clark on the "Great Britain." He found that, in that engine, the steam, instead of becoming

* Labouchère, *Machine à Vapeur*, §2, p. 22.

† Read to Manchester Association for Prevention of Steam-boiler Explosions, Jan., 1862.

pounding is carried, and would apply the jacket to all the cylinders, but thinks its value greatest on the low-pressure and least on the high-pressure cylinder.

He finds the gain by its use, under the conditions of ordinary practice, as a general average, to be about 20% on small and 8 or 9% on large engines, varying through intermediate values with intermediate sizes, it being understood that the jacket has an effective circulation, and that both heads and sides are jacketed. Effective clothing of the engine-cylinder with non-conductors may slightly reduce the necessity of jacketing.

Mr. Isherwood finds, as a rule, in his experience that about one-half the condensed steam in the jacket usually comes of heat-transfers to and from the metal of the cylinder. He thinks it possible entirely to prevent this loss by simply adopting the principle, first enunciated by James Watt, of "keeping the cylinder as hot as the steam which enters it." *

According to Ledieu: "A non-conducting covering is obviously indispensable from every point of view, and its employment cannot be too fully recommended. But it is not so with steam-jackets, the utility of which is sometimes contested. In fact, experiment sometimes indicates an economy by their use of 10 to 12%, while in other cases the benefit is inappreciable. . . . Steam-jackets are now usually formed of *sheets* of iron, which envelope the cylinder, leaving an interspace. They have also begun at Creusot to employ a second cylinder of *cast* iron around the real steam-cylinder. In all cases the steam which is taken into the valve chest flows through the jacket, entering the latter by a small pipe furnished with a cock. Often, too, the steam is carried into the cylinder head and cylinder bottom. There is also always at the lowest part of the jacket a cock which serves to draw off the water of condensation, produced by the passage of heat into the cylinder. This cock or valve should be so set that it will allow the water formed to flow away, but none of the steam."

"In old engines steam-jackets may be found cast with the steam-cylinder, and into which the steam from the boiler enters on its way to the valve chest. This arrangement has been long since abandoned. For, in the first place, there are serious difficulties in casting the jacket and cylinder in one piece; in the

* Correspondence.

compound cylinders; and in both cases the gain increases as the ratio of expansion is increased. But the gain is much less for the compound cylinders than for the single cylinder—an unexpected comparative result, which is due to the fact that the range of pressure and temperature is less in each of the compounded cylinders, between which it is distributed, and which are completely isolated from each other, than in the single cylinder."

Clark also concludes, after a comparison of jacketed and unjacketed cylinders, both on long and on short-stroke simple engines,* that the evidence so gathered "proves what has long been acknowledged—the economical advantage of superheating the steam; and, more remarkably, the striking disadvantage of short-stroke versus long-stroke cylinders." . . . "The relatively large absorbing surfaces of the covers and the piston of short-stroke engines are disturbing influences which affect the operation of the steam in the cylinder to a greater extent, proportionally, than in long-stroke cylinders." He also adds, later: †

"Large second cylinders proportionally to first cylinders, in the ratio of 4 or $4\frac{1}{2}$ to 1, may be employed with economy when the cylinders are thoroughly steam-jacketed; but they are unfavorable to economy when the cylinders are only partly, or not at all, steam-jacketed."

Mr. Isherwood considers the steam-jacket to be "of real economic value; but in degree very variable." ‡ He concludes from his experiments and observation that "when steam is used without expansion, the value of the jacket is a minimum, and is absolutely small for cylinders of medium and large size; though considerable for small engines. When considerable expansion is practised the efficiency is greater with greater ratios of expansion, with increasing initial pressure of steam, and decreased speed of rotation, with large quantities of water in the entering steam, and the smaller the relative area of steam-ports. It is less as size of engine increases, and as superheating is made more effective, becoming zero when all initial condensation is thus checked."

With steam in the jacket direct from the boiler,§ he considers the value of the jacket to increase with the extent to which com-

* *Steam Engine*, vol. i., p. 577.

† Ibid., 581.

‡ Correspondence.

§ This is, of course, not always practicable, with high pressures, for the large cylinders.

	With jacket.	Without.
Fuel per I. H. P. per hour..............	2.6 lbs.	3.8 lbs.
Water " " "	22.5 "	32.7 "

This gain of 30% was obtained, as he states, with the same steam pressure, the same vacuum, and the same speed; but he developed less power without the jacket. This may probably have somewhat exaggerated the apparent gain. *

The same writer, later, says that the results of his experience, generally, on compound engines, with and without jacket, indicate "a saving to 23 to 25%" by its use.† Messrs. Farey and Donkin give for the case of a compound engine, with steam at 45 lbs. by gauge, and a ratio of expansion of 9, a water-rate of 24¼ lbs. with and 39½ lbs. without the jacket; and a correspondent of the same journal found the fuel account to be raised "14 to 20 cwt. per day by the shutting off of the jacket." ‡

Still other testimony is given by other correspondents to the effect that a marine compound engine, built by Messrs. R. Napier and Sons, dropped off from 63 to 58 revolutions per minute when the jackets were shut off; also, "when once an energetic engineer finds that a steam-jacket gains him five or six revolutions per minute, you may feel assured that he will keep them on; and, last but not least, the owner gains 12 or 13% by the mere fact of the engineer doing his duty." § The same writer asserts: "Trials conducted under my charge showed as much as 15% gain in marine engine practice." ‖

Turnbull thinks the steam-jacket more effective in the compound than in the simple engine, and attributes to this cause a part of the economical advantage observed in the employment of the former. He states that, "as the variation of temperature is much greater in the single cylinder than in either of the cylinders of a compound engine, the heat from the steam-jacket must pass through the metal with great rapidity to replace that wasted by condensation, and this it cannot do so effectively as when the temperatures are not so widely varied; and this is one of the great advantages possessed by the compound or double-cylinder engine." ¶

---

* *Lond. Engr.*, May 15, 1874, p. 352.

† Ibid., June 5, 1874, p. 409.

‡ Ibid., August 29, 1874, p. 389.

§ Ibid., April 10, 1874, p. 257.

‖ Ibid., May 8, 1874, p. 339.

¶ *Compound Steam Engine*, Van Nostrand's Science Series, No. 8, p. 34.

second place the steam is only distributed by the valve after it has been cooled to some extent by its passage around the cylinder. Such cooling is incompatible with the best utilization of the fluid." *

In comment on the above it may be suggested that the length and proportions of the cylinder will probably be found to modify the result. Little, if any, trouble is met with in engines of short stroke in consequence of variation of expansion and contraction, unless through gross carelessness in management.

Ledieu would always use the steam-jacket, even on the high-pressure cylinder of the compound engine, but would prescribe that it be assured a correct construction and efficient action. He would use it in combination with both compounding and superheating, and thinks that it is easy now to understand how, by the combination of these expedients, "economies of 40%" may sometimes be attained. He would be especially careful, in the multiple-cylinder engine, to properly adjust the ratios of expansion for each cylinder; otherwise the jackets may fail to yield a satisfactory gain in economy.†

Ledieu asserts that the recent developments respecting the action of the steam-jacket afford a complete explanation of "economies of fuel of 15 to 25% of the anterior expense which the experiments of later years accord, incontestably, to-day, to the use of steam-jackets applied to high-pressure engines, at large ratios of expansion and under good conditions of operation." ‡

He goes on to say that the jacket may cease to be useful when it is incapable of supplying heat enough to prevent the cylinder-wastes becoming equally great as without it, as may be the fact when it fails to secure a dry cylinder at the close of the exhaust period. Indeed, it may then become even detrimental, since this waste may then become even greater during the exhaust than when no jacket is attached to produce accelerated evaporation of existing moisture and rejection of such excess of heat into the condenser.

Mr. B. W. Farey, in 1874, gave data which he considered "conclusive" as to the value of the steam-jacket on such an engine as he then referred to, a compound engine of small power, thus:

* *La Machine à Vapeur*, Ledieu, vol. ii., pp. 72–73.

† Ibid., p. 416.

‡ *Machines à Vapeur*, 1882, p. 405.

same conditions being otherwise maintained as nearly as possible, M. Cornut found that shutting off the jackets produced a very sensible reduction of initial pressure in the small, and of work in the large cylinder, increased the loss of pressure between the two cylinders, and reduced economy.*

Similarly testing a pair of Corliss engines, he obtained the indicated horse-power on an expenditure of 25.5 with the jackets in use and of 31.56 without, a gain of 19.1% by the use of the jacket. He concludes that, with ratios of expansion of 6 to 10, a good jacket should give an economy of 15 to 20%.

Professor Cotterill says: "Experience shows that a steam-jacket is advantageous; and the reason why it is so has been pointed out," . . . "but the amount to be gained will vary according to circumstances. In many cases it may be that the advantage is small."† "Great caution is necessary in drawing conclusions from any special set of experiments on the influence of jacketing." ‡

Seaton only claims for the jacket that it affords a means of "warming up" the engine before starting. He refers to the compound engine.§

Sennett asserts that "it has been abundantly proved, both by experiment and by actual experience, that steam-jackets are not only advisable, but absolutely necessary, in order that high rates of expansion may be efficiently carried out and the greatest possible economy of heat attained;"‖ "the economy derived from its use is so decided as to be beyond the region of doubt."¶ He thinks its value practically greater on the sides than on the heads.**

A correspondent of one of the technical journals remarks: "A large proportion of the steam-jackets in use are steam-jackets only in name, owing to the accumulation of water or of air, or both, in them, arising from the careless manner in which they have been made, or laziness on the part of the attendant,"†† and a number of cases have been reported in which they have been actually shut off, and allowed thus to fill, by the attendants to "keep the engine room cooler." ‡‡ A jacket thus

* *Eng'g and Mining Journal*, Jan. 26, 1878.

† *The Steam Engine*, etc., p. 266.

‡ *The Steam Engine*, etc , p. 267.

§ *Manual of Marine Engineering*, p. 122.

‖ *The Marine Steam Engine*, p. 185. ¶ P. 187. ** P. 188.

†† *London Engineer*, Sept. 28, 1877, p. 245.

‡‡ Ibid., April 10, 1874.

It is said by Mr. Salter that the steam-jacket "was originally applied to the body of the cylinder only; then to the end and cover, and finally some engineers have admitted steam to the piston. This is, of course, expensive; but it no doubt tends to economy, appreciably where the surfaces are large."*

Professor Trowbridge says: "It may be stated, as a general conclusion, that the use of a steam-jacket with a single cylinder under ordinary circumstances results in an important saving of fuel, especially for high degrees of expansion; that in the compound engine, with the large cylinder jacketed, there is a saving in economy over a single cylinder, jacketed, even with the same pressure and degrees of expansion in both."†

Mr. G. B. Dixwell thinks that "Mr. Emery's experiments go to show that the steam-jacket does but very little toward suppressing cylinder condensation in the large cylinders of compound engines, or in the single cylinders of other engines."‡ On the other hand, the Messrs. Perkins account for an apparent discrepancy of over 35% in the performance of their steamer *Anthracite*, at boiler pressures approximating 400 lbs. per square inch, in a triple-expansion engine, or between 17.8 and 21.6 lbs. of water per horse-power per hour, mainly by "the steam-jackets being improperly worked." §

Mr. Emery, in his own discussion of the value of the steam-jacket, as deduced from his work on the revenue steamers, concludes: ‖

The "saving by the use of a steam-jacket on a single-cylinder engine worked at its most economical point of cut-off" was 11.78%. "With more expansion .... the jacket produces a greater saving." The saving due to a jacket on the large cylinder of a compound engine was "shown to be 11.72%. He finds that, in the cases reported on, "the steam-jacket on the smaller or high-pressure cylinder of the compound engine, working with the ordinary degree of expansion, was, contrary to the views of Rankine on the subject,¶ of comparatively little value."

Experimenting with a compound engine with and without steam in its jackets, the pressure being about 100 lbs., the

---

* *Economy in the Use of Steam*, p. 36.

† *Johnson's Cyclopedia*, vol. iv., p. 524.

‡ Monograph *On Cylinder Condensation*, p. 27.

§ Sir Frederick Bramwell's Report.

‖ *Journal Franklin Institute*, 1875.

¶ *Steam Engine*, § 286.

designing and constructing large marine engines, does not feel inclined to expect much from their use. He says: "It is not, therefore, only to the use of steam higher in pressure, or in such complications as steam-jackets, that we must look for economy, but in the employment of an engine with a degree of efficiency proportionate to the increase of pressure where the range of expansion and temperature in each cylinder during one stroke is limited by the division of the work between two or more cylinders."*

M. Schneider, of Creusot, made numerous experiments, extending over a period of six months, upon a Corliss engine, at the Creusot works, the results of which were reported by M. Delafond in the following year.† In these experiments careful examination was made of the disputed useful effect of the steam-jacket, with what M. Delafond considers satisfactory results. The jacket covered the cylindrical portion of the engine only.

The results of these elaborate and carefully conducted investigations, so far as they relate to the steam-jacket, are the following:

"The jacket reduces the expenditure the more at equal ratios of expansion, as the pressure is higher; its effect, important at 7.75 atmospheres, with condensation, becomes very slight at 2.5.

"The economy due to the jacket is the less at the same pressure as the effective power is the greater—*i.e.*, as the expansion is less.

"It is found advantageous to employ in the jacket steam of higher temperature than that in the engine cylinder."

In this latter case, the gain was between 5 and 6% only, thus:

	With Jacket.	Without.
Pressure, initial	$4^k.15$	$3^k.79$
" at cut-off	2.27	1.95
Admission	0.57	0.57
I.H.P.	162.8	137.8
Water per H.P. per hour	$10^k.00$	$10^k.55$

The boiler and jacket pressures were about 7 atmospheres (7^k).

The gain by the jacket in these experiments was usually not far from 15 or 20% under ordinary conditions of operation.

* "Utilizing Steam of High Pressures." *Proc. Eng'rs' Club of Philadelphia*, June 21, 1884.

†"Essais effectués sur une machine Corliss." *Annales des Mines*, September, October, 1884.

treated becomes worse than useless. Instead of warming the cylinder at the critical periods, it may act as a refrigerator by enclosing a large mass of comparatively cold water.

Mr. E. D. Leavitt has expressed the opinion that, in his practice, steam-jackets produce an increase of efficiency of from 15 to 20%.*

Mr. Longridge reports substantially as good work, in the case of a very economical compound engine at Heyworth, without as with the jackets in operation. In fact, he obtained a trifle better work without, the water-rate being given by him as 16.9 and 16.8, respectively, and at ratios of expansion varying from 8 to 11. He thinks that if a jacket is used at all it should be on the high-pressure cylinder.†

The practical objection to the use of the steam-jacket, in many cases, even when of admitted efficiency, is illustrated by a remark in *Engineering*, referring to the results of an engine trial: "A steam-jacket would have prevented the large condensation during admission and thus reduced the exhaust and expansion waste. When designing the engine, it was determined not to use the steam-jacket, some doubt being felt as to probable trouble and loss from leakage."‡ The designer usually looks upon this as a really serious matter and a strong objection to the use of the jacket. Nevertheless, it is a difficulty which it is possible, unquestionably, to meet satisfactorily.

"The economy which follows the application of the steam-jacket is very variable, according to the type and condition of installation of the engine. Experience has demonstrated that it is especially useful in 'Woolf engines,' having two cylinders, in which it considerably increases the work of expansion. We believe that the steam-jacket will be found the more efficacious as the stroke of piston is the greater in proportion to its diameter, and as the expansion is carried further. Its use is indispensable in condensing engines. If superheated steam is employed, it should be led to the jacket through a special supply pipe. The economy varies, as already stated, between wide limits, and fluctuates between 10 and 25%."§

Mr. Horace See, who has had a very extended experience in

* *Trans. Am. Soc. Mech. Engrs.*, 1880, vol. i. Discussion of Mr. Hoadley's paper, p. 36.

† *London Eng'g*, March 10, 1882, pp. 220–266.

‡ *London Eng'g*, November 24, 1882, p. 495.

§ *Revue Industrielle*, January 9, 1884.

and start. His general conclusion is that, as locomotives are customarily handled, the jacket has no appreciable value, all things considered, on simple, and may be a decided disadvantage with compound, engines. This fact and its cause are shown by the circumstance that a gain observed while the engine was tested under cover, was lost when under steam on the road.

In the discussion of this report Mr. Halpin stated that, with well-drained jackets, he had attained an economy of 17% by their use, and the greater the condensation in the jacket the greater that economy.

Mr. Halpin considers that the value of the jacket increases with the expansion ratio, and with the completeness and effectiveness of the jacketing; and that it decreases with increase of engine speed.*

Good drainage is always insisted upon, and this is always carefully attended to in Cornish engines, which are commonly arranged so as to drain the water of condensation back to the boiler. When the cylinder is of large diameter and short stroke it is considered especially important to jacket the heads.† The general belief of engineers is unquestionably that experience to date shows the jacket to be desirable on slow engines at considerable ratios of expansion; ‡ and as stated by Mr. Greig, "a slow engine would not do without it, and a fast engine would do badly without it, but not as badly as a slow one." § This belief and the facts are not undisputed for exceptional instances, however, as seen elsewhere.

Mr. T. R. Crampton, as early as 1857, designed six locomotives for 140 lbs. steam and a ratio of expansion of 5, and applied steam-jackets, having previously satisfied himself of their importance by experiment on simple engines.‖

Mr. Benj. Walker found steam-jackets on mill engines under but 44 lbs. steam-pressure to give, in experiments, each extending over some weeks, a gain as compared with the results obtained with the jackets shut off, and considered that 10 to 15% saving should be expected from their use.¶

Mr. Wylie stated, referring to steam-jacketing, long ago, that the

* *Proc. Brit. Inst. M. E.*, Aug., 1886

† Ibid., Mr. Davey's testimony.

‡ Ibid., Ryan.

§ Ibid.

‖ *Proc. Brit. Inst. M. E.*, Aug., 1886.

¶ *Proc. Brit. Inst. M. E.*, Aug., 1886.

M. Borodin, in his extended and valuable investigations of jacket-action on locomotives, adopted Hirn's system of test and computation as far as practicable.* All his experiments on the common type of engine exhibit economy due to the use of the jacket, amounting to 12% with a ratio of expansion of 3.33, and of 16% at ratios of expansion approximating 5.†

The effect of the steam-jacket was clearly seen in these cases to decrease initial condensation; to decrease reëvaporation, usually, during expansion, to increase the mean pressure. In one case the engine gained 14%, through a decrease of 8% in initial condensation; in other cases, gains of 4, of 11, of 7, of 16, of 12, of 17, and of 21% through diminution of cylinder condensation at admission to the amount of, respectively, 2, 9, 8, 15, 16, 13, and 20%. The general result was fully confirmed by trials of a standard engine, precisely similar, except that it had no jacket. Initial condensation, with the reversing lever in the first notch, amounted to 40% reëvaporation during expansion, to nearly the same proportion; with the lever in the second notch, the figures become 19% in both condensation and reëvaporation, the ratio of expansion being considerably less than in the first case. This difference in wastes accounts for the fact that no economy of steam followed the change from the lower to the higher ratio of expansion in this case.

The attempt to determine the value of the jacket on compound locomotives did not succeed, since, as M. Borodin states, the jackets were defective; but they do show, very clearly, the necessity of proper construction and effective air and water drainage of the jacket. In the simple engine, "following," nearly full stroke, the jacket had no value and even appeared sometimes to act injuriously. With the compound engine, this somewhat singular result became vastly more apparent, probably for the reason just given, and the loss by the jacket ranged from 0 to 16% and without any apparent relation to the degree of expansion adopted. M. Borodin, however, feels justified in concluding that the useful effect of the jacket should be expected to be less with compound than with simple engines, and calls attention to an hitherto unnoted source of waste—that due to heating and cooling of cylinders and jackets at each stop

* "Steam-jacketing and Compounding Locomotives." *Proc. Inst. Mech. Eng.*, 1886.

† Ibid., p. 315.

to which it limits condensation to the quantity which can be re-evaporated during expansion, leaving no evaporation to be effected during the exhaust stroke, whether of a single-cylinder engine or the last cylinder of a multiple stage expansion engine."*

In the tests of pumping engines reported by Mr. Mair, the value of the jacket is well illustrated.† In simple engines at low speed, at the average work, with jackets in use the steam used was about 22 lbs. per horse-power per hour; without jackets the reported figure is 26.5 lbs., and an increase of cylinder condensation is computed from 25 up to 37%. These engines averaged, however, but about twenty revolutions per minute. The same writer gives tables of similar results with compound pumping engines, in which the unjacketed engine demanded from 19.2 to 26.6 lbs, while the jacketed engine used but 14.8 to 17.4, a difference of 20 to 35%, presumably largely due to the operation of the jackets.‡

Holmes, referring to an analysis of the work of Emery on jacketed and unjacketed engines, remarks: "We see that, under all circumstances, whether the engine was worked simple or compound, a considerable saving was effected by the use of the jacket, when corresponding rates of expansion are compared. For instance, when worked compound, about 3 lbs. of water were saved per horse-power per hour, being at the rate of about 13%, while in the case of the simple engine, worked at the higher rates of expansion, as much as 22.5% was saved."§

Mr. Vaughan Pendred had tried a triple-expansion engine with the jacket of its high-pressure cylinder "carefully blown through and left perfectly dry. Again he had tried the engine with the jacket full of water, and again he had tried it with the steam cut off entirely; but he never could find that there was the slightest difference in the amount of steam which was condensed in the high-pressure cylinder." . . . "Mr. Mudd, of the Central Engineering Works, had said to him: 'I always find that the high-pressure cylinder is wet. I cannot account for it in any way except that I believe all marine boilers send a

* High-pressure Steam and Steam-engine Efficiency; *Proc. Soc. of Eng's.*; Dec. 3, 1888.

† *Minutes of Proc. of Brit. Inst. C. E.*, vols. lxx., lxxix.

‡ *Enc. Brit.*: Art., St. Engine, § 96, p. 490.

§ *The Steam Engine*, p. 454.

"theoretical advantages of the system have long since been admitted."* But "keen competition seems evidently to have been one of the reasons why this fitting, in many instances, has been discontinued, in order to meet the demand for a cheap engine." He considers that in triple-expansion engines the benefits of the jacket "are naturally not as great as in a single-cylinder engine with a high ratio of expansion;" but, however carefully it may be designed, "the jacketing of at least the intermediate and low-pressure cylinders is essential to maximum efficiency." General experience to date would seem to indicate, however, that cylinder-waste takes place to the greatest extent in the high-pressure cylinder, while the steam works quite dry in the low-pressure engine.

Major English found that even jacketing the steam-pipe in engines tested by him sometimes increased their efficiency 5% and over, so sensitive is the expansive steam-engine to variations of quality of steam.†

According to Professor Ewing: "In high-speed engines its beneficial effect is necessarily small, and, in certain cases, the benefit may be even more than neutralized by the drawbacks which have been alluded to." . . . "In general, however, the steam-jacket forms a valuable means of reducing the wasteful action of the cylinder walls, especially when the ratio of expansion is considerable. Experiments made with and without a jacket on the same engine have shown that jacketing may increase the efficiency 20 or 25%."‡

"Many experiments have shown that in large, and especially with slow-running engines, the influence of a steam-jacket is, in general, good; and this is to be ascribed to the fact that it reduces, though it does not entirely remove, the evils of initial condensation. To be effective, however, the jacket must be well drained and kept full of live steam, instead of being, as many are, traps for condensed water or for air."§

Beaumont also asserts: . . . "It would appear that a steam-jacket can have but little if any value in triple or quadruple expansion engines, and probably not in compound engines; and in any engine its value is probably confined to the extent

* "On Triple-expansion Marine Engines." *Trans. Brit. Inst. M. E.*, 1886.
† *Trans. Inst. Mech. Engrs.*, 1887.
‡ *Encyclopædia Britannica*, Art., St. Engine, § 92.
§ *Encyclopædia Britannica*, St. Engine, § 92.

after a while shut it off. In fact, the subdivision of expansion into successive steps was a much more efficient method of economizing steam than the steam-jacket had ever been, and had rendered the latter useless.' "

"In one of the papers* to which I have alluded I pointed out that the use of a steam-jacket, as ordinarily designed, is a violation of a fundamental law of maximum efficiency of heat engines, which requires that they should receive all their heat at the maximum and give it out at the minimum temperature, and not, as in the case of an engine with a steam-jacket, at temperatures between these, and at times when the heat imparted lessens the efficiency, which it evidently must do at and near the end of the stroke. The steam-jacket may thus be looked upon as a necessary evil, justified only by the physical properties of the steam, and of the materials hitherto used in the construction of engines, for while it increases the work done by the expanding steam this increase is by no means so great as it would be if the heat employed in the jacket steam had been employed to generate more steam. The advantage to be derived from the use of a steam-jacket therefore varies according to the circumstances under which it is employed, and in some cases—as, for instance, when low rates of expansion are used—the jacket may not only be useless but wasteful. On the other hand, when high rates of expansion are used, by preventing the temperature of the cylinder from falling below the boiling-point corresponding to the initial pressure of the steam, the economy resulting from the action of the jacket is considerable, the work developed by the engine being increased by an amount varying from 15 to 25% of what would be developed without the jacket. The necessity for a careful study of the conditions of efficiency of a steam-jacket, of care in its application, and of experiments to test its effects, are thus evident."

Professor Dyer concludes that the gain by the use of a jacket may be, in actual work, anywhere from 0 to 20 or 30%; that it should be employed when the ratio of expansion is large; when the variations of pressure, expansion, and range of temperature are great, its value is doubtful; that superheating is a better method of reducing wastes; that the higher the engine-speed the less the value of the jacket; from a theoretical point of view the jacket would be considered desirable on the small cylinder,

*Trans. Inst. Eng. and Ship. Scot., vol. xxix., p. 61.

very large quantity of water into the cylinder, and what you consider to be condensed steam is really priming.'" . . . "He found that, as a rule, triple engines worked extremely dry steam in their low-pressure cylinders."*

Professor Dyer, in his paper on recent steam-engine trials, and referring to the discussion of the report of Mr. Willans, says: †

"Mr. E. A. Cowper dwelt on the importance of steam-jackets and pointed out the difference between the action of the condensing steam and reëvaporating water, and alluded to engines he had designed more than forty years ago in illustration of his opinions. But many things have happened in these forty years, and, as Mr. Cowper well knows, successive cylinder expansion engines have come into very general use, and altered the circumstances very considerably. As Dr. Kirk remarked: 'No doubt in the earlier non-compound engines, when the steam was worked through a large range of temperature, jackets were a very valuable addition; but as far as he had observed, with the ranges of temperature in the best compound and in the modern triple-expansion engine, he could not trace any advantage in the jacket. The ideal function of a steam-jacket was a neutral one: simply to prevent condensation. Unfortunately, it also acted as an evaporator. When so acting it was in fact a boiler, in which a higher-pressure steam was employed to generate steam of a lower pressure. Without going into the ultimate value of its action in the single-acting Cornish engine, it was clear that to expend boiler steam to generate lower-pressure steam in the low-pressure cylinder (or even in the intermediate) was not an economical way of using it, the more so as the steam generated in the low-pressure cylinder had little opportunity of doing any work, but went immediately into the condenser. Better it should go in as water. Jacketing the high-pressure cylinder seemed to add nothing perceptible to the heat economy, but contributed sensibly to the wear and tear. The very large volume of water that came from a steam-jacket, although the range of temperature in its cylinder was small, led him to think that the steam thus condensed would be better employed if put into the cylinder itself. He had found intelligent marine sea-going engineers use the steam-jacket to heat up the engine, and then

* *Proc. Brit. Soc. of Eng's*, Dec. 3, 1888.

† *Trans. Inst. Engineers, of Scotland*, January 22, 1889, p. 54.

The three independent engines combined in the compound machine were of the following dimensions :

	Cylinder. Diam.	Stroke.
No. 1	5 inch.	10 inch.
" 2	8 "	10 "
" 3	12 "	15 "
Air-pump on No. 3	9 "	4½ "
Feed-pump	1½ "	2 "

All were jacketed, sides and heads ; steam was carried at 200 lbs. per square inch, and boiler pressure was maintained in all the jackets.

The results were the following, with and without the jackets in use :

	With Jackets.	Without.
Coal, per horse-power per hour	1.33 to 1.50	1.62 to 1.81
Water, " "	12.68 " 14.10	15.90 " 17.30

The effect of radiation was determined and found somewhat considerable. Deducting this waste, the figures stand :

	With Jackets.	Without.
Coal	1.21 to 1.30	1.54 to 1.77
Water	11.90 " 12.30	15.10 " 16.60

This is one of the most satisfactory approximations to the ideal engine and to minimum wastes that has ever yet been recorded.

In this remarkably economical engine, it is seen that the loss by shutting off the jackets was from 25 to over 35% in fuel consumption ; or from 25 to 30% in water expenditure.

Of the total heat received, exclusive of radiation, 19.4% was converted into work with jackets in action, and but 15 without, a difference of over 23% of the first quantity, or 29 of the latter. The ideal engine, under similar thermo-dynamic conditions, would have utilized 23%.

The effect of the jacket on the high-pressure cylinder, where the difference of temperature between jacket-steam and initial was small, was found to be slight as affecting cylinder condensation. In No. 2, the effect, with a difference of temperatures in this respect of 80° Fahr., that condensation was reduced from 30 to 5% ; while in No. 3, with a difference of 180° Fahr, such condensation was sensibly zero, and the "saturation expansion-curve," assumed by Rankine to be attainable by this use of the jacket, was perhaps for the first time produced.

M. Dwelshauvers-Dery does not think that we may expect but practically it is found that it is better to omit it from that cylinder, as it observably exaggerates wear; that the use of the jacket is contrary to the thermo-dynamic principles of efficiency.* He states that some quadruple-expansion engines without jackets have proved satisfactory.

Studying the action of the jacket where it is used under ordinarily efficient conditions in general, and analyzing the final efficiency into its several factors, † M. Dwelshauvers-Dery finds that, in the case of the experimental engine reported on by Professor Reynolds (1889), a gain of 17% was produced by the cylinder jackets, by the net gain due their action in promoting efficiency of working, aside from thermo-dynamic efficiency. Similarly studying the famous experiments of M. Hirn at Logelbach, with and without superheating, he finds that superheating produced a gain of 25% by promoting thermo-dynamic efficiency, raising its value from 0.324 to 0.433; while the saturated steam was made more efficient as a working substance by jacketing, and to the extent of 17, although the purely thermo-dynamic efficiency was actually reduced 1.6%.‡

A triple-expansion, condensing, three-cylinder engine, having jackets on cylinders and intermediate receivers, as tested by Professor Schröter, in 1889, was reported to consume but 6.9 kilogrammes (15.2 lbs.) of steam per indicated horse-power and per hour, when driving about 200 H. P., and including in that weight all steam passing through the jackets, which averaged about 17% of the whole. The boiler steam pressure was about 150 lbs. (10.5 kilogrammes per square centimeter). Clearance was reduced to 4 and 5%, the total ratio of expansion averaging about 24, ranging from 22 to 26.§ No superheating was practised. The efficiency of the machine was 88.5%, the friction of the unloaded engine demanding 11.5 of the total maximum power.

One of the most satisfactory of recent determinations of the value of the steam-jacket on compounded engines is that of Professor Osborne Reynolds, of Owens College, Manchester, employing the triple-expansion engines of the Whitworth Laboratory.‖

* Correspondence.

† *Trans. Amer. Soc. Mech. Engineers*, 1882; *Journal Franklin Institute*, 1882; *Thurston on the Several Efficiencies of the Steam-engine.*

‡ *Engineering*, Feb. 21, 1890, p. 201.

§ *Zeitschrift des Vereins Deutsche Ingeniure*, January 4, 1890.

‖ *Proceed. Brit. Inst. C. E.*, December 10, 1889.

The three independent engines combined in the compound machine were of the following dimensions :

	Cylinder. Diam.	Stroke.
No. 1	5 inch.	10 inch.
" 2	8 "	10 "
" 3	12 "	15 "
Air-pump on No. 3	9 "	4½ "
Feed-pump	1½ "	2 "

All were jacketed, sides and heads ; steam was carried at 200 lbs. per square inch, and boiler pressure was maintained in all the jackets.

The results were the following, with and without the jackets in use:

	With Jackets.	Without.
Coal, per horse-power per hour	1.33 to 1.50	1.62 to 1.81
Water, " "	12.68 " 14.10	15.90 " 17.30

The effect of radiation was determined and found somewhat considerable. Deducting this waste, the figures stand:

	With Jackets.	Without.
Coal	1.21 to 1.30	1.54 to 1.77
Water	11.90 " 12.30	15.10 " 16.60

This is one of the most satisfactory approximations to the ideal engine and to minimum wastes that has ever yet been recorded.

In this remarkably economical engine, it is seen that the loss by shutting off the jackets was from 25 to over 35% in fuel consumption; or from 25 to 30% in water expenditure.

Of the total heat received, exclusive of radiation, 19.4% was converted into work with jackets in action, and but 15 without, a difference of over 23% of the first quantity, or 29 of the latter. The ideal engine, under similar thermo-dynamic conditions, would have utilized 23%.

The effect of the jacket on the high-pressure cylinder, where the difference of temperature between jacket-steam and initial was small, was found to be slight as affecting cylinder condensation. In No. 2, the effect, with a difference of temperatures in this respect of 80° Fahr., that condensation was reduced from 30 to 5%; while in No. 3, with a difference of 180° Fahr, such condensation was sensibly zero, and the "saturation expansion-curve," assumed by Rankine to be attainable by this use of the jacket, was perhaps for the first time produced.

M. Dwelshauvers-Dery does not think that we may expect that the good results of jacketing and superheating may be "superposed"—*i.e.*, that effective superheating may be expected to add to the efficiency attained by a good system of jacketing, or *vice versa*. This was the view also of M. Hirn, who, November 7, 1889, only a few weeks before his death, wrote M. Dwelshauvers: "The use of superheated steam absolutely dispenses with the need of a jacket. Conforming with the ideas of Leloutre, I have applied to one of our engines, not a complete jacket, but jackets surrounding the ends of the cylinder, leaving about one-half at the middle uncovered. I passed through these jackets steam, saturated, but not superheated. I never found the least advantage in their action." He goes on to say, referring to his engines at Logelbach, that he had there shown the greater utility of superheating, properly carried out, and that "if in England and elsewhere superheating has been abandoned, it should be presumed that proper precautions had not been observed." *

M. Dwelshauvers "has more faith in superheated steam than in the jacket, and would adopt with it a suitable compression," but thinks it possible that M. Hirn may have somewhat exaggerated its relative ultimate value. He had himself, however, described the results of the use of superheating steam when he had been able to secure so effective an action of the superheater as to retain the steam in a sensibly dry and saturated condition to the end of the stroke.†

He deduces from the experiments of Reynolds the following gain by the use of the jacket, in three comparisons: ‡ 18.79%, 18.50%, and 17.16%, or an average very closely approximating 18%. But he finds, as the writer earlier suggested, that the maximum speed is not necessarily that of maximum efficiency in these trials, although the highest speed is more economical by an eighth than the lowest. An intermediate speed is more economical than either.§

M. Hirn thinks that it would be unwise to use superheated steam in the jacketed engine, since, as he suggests, if the jacket contained saturated steam it would then act as a refrigerator;

* Correspondence.

† *Bull. de la Soc. Ind. de Mulhouse*, 1888.

‡ *Proc. Brit. Inst. C. E.*, 1890.

§ See President's address, Am. Soc. Mech. Engrs., R. H. Thurston, *Trans.*, 1880, vol. i., p. 14: "It seems possible to reach a point in steam-jacketed cylinders at which lower speed may tend to secure efficient working."

while, if supplied with superheated steam, there would be some liability of its being inactive. M. Dwelshauvers is inclined to accept the complication of compounding and jacketing for pressures approximating 200 lbs. per square inch, but would try the simple engine, unjacketed, with a good superheating apparatus with lower and ordinary pressures.

M. Cornut tells the following instructive incident: "A manufacturer of Roubaix employs two compound engines. The steam which supplies their jackets is taken directly from the boiler, and is afterward used for heating some special apparatus, the arrangement being such that the jackets can be filled with either saturated or superheated steam. One morning, the engine being much loaded and making with difficulty the twenty revolutions a minute which was its required rate, the engineer thought he would try the use of superheated steam in the jacket instead of the saturated steam. Effecting the change, he found, to his surprise, the engine make 21.5 to 22 revolutions per minute—*i.e.*, an increase of nearly 10%."

Professor Geo. I. Alden, using a small experimental expansion engine of but 10 I. H. P., has attained an economy measured by the expenditure of but 17½ lbs. of steam per horse-power per hour, and without the jacket the engine required about 20½ lbs., the saving by its use amounting to about 15%.

M. Mallett, after a long experience, says: "The utility of the steam-jacket is incontestable; and it is so much the greater as the difference of temperature is greater between the sides of the cylinder and the inflowing steam. Accordingly, the jacket will be more advantageous for condensing than non-condensing engines, for larger than for smaller measures of expansion, for engines with one cylinder than for compound engines." *

M. Mallett, the French engineer and constructor, concludes; "(1) The usefulness of the steam-jacket is incontestable, being greater as the differences in temperature increase, so that jackets are more advantageous for condensing than for non-condensing engines, for great expansion than for slight, for single than for compound engines. But when expansion is considerable the jacket is not sufficient. (2) Engines of two cylinders have a decided superiority over those with one; so that, for moderate expansions, steam-jackets may be dispensed with." †

* Cornut.

† *On Compound Engines*, sec. iii.

The late Henry R. Worthington, who had an enormous experience with engines of low ratio of expansion, perhaps averaging inside 3, was accustomed to reckon the gain by the jacket in such cases at about 6%.* In engines of the Worthington type, experience shows clearly and indubitably that the jacket affects condensation in the cylinder so greatly as to have a sensible and important effect in sustaining the pressures on the piston and enabling the stroke to be completed when it would otherwise fail to make a full stroke. It is for this reason, if for no other, made invariably an appendage of such engines.

Professor Unwin considers that "in all cases and on all cylinders the jacket is useful, provided, of course, ordinary, not superheated, steam is used;" but "the advantages may diminish to an amount not worth the interest on extra cost." In most cases, so far as experiment shows, there is some saving.†

Mr. Edwin Reynolds, the designer of an immense number of engines of many kinds, considers that "the jacket is of real economic value, the amount of that value varying with conditions under which the engine is worked; the slower the speed and less the work done in a given time, the greater the value of the jacket becomes.‡ In some of his experiments on a "quarter-crank" compound engine, at 12 revolutions per minute, he obtained an economy of 33% with jackets, as compared with the same engine without them. Driving the engine up to 56 revolutions, the gain fell to about 9%. The tests of the Milwaukee Pumping engines gave a decided saving due to the jackets. The experience of Mr. Reynolds has indicated a marked advantage where the larger cylinder is jacketed with boiler steam, over jacketing with steam of lower pressure.

The experience of the Holly Manufacturing Co. on their compound pumping engine, as designed by the late Mr. Gaskill, leads to these conclusions on the part of their superintendent: "In regard to the benefits derived from steam-jackets on our steam-cylinders, I am somewhat of a sceptic. From data taken on our own engines, and tests made, I am yet to be convinced that there is any practical value in the steam-jacket." . . . "You might practically say that there is no difference." He finds the engine to run more quietly with the jacket than with-

* Private Correspondence.

† Correspondence.

‡ Private Correspondence, December 31, 1889.

out, as, in the latter case, there is more condensation in the larger cylinder and consequent "water-hammer" at times at the end of the stroke.* He therefore uses the jacket.

The experiments of the Westinghouse Company, on their "Duplex No. 1," an engine having both cylinders partly jacketed, "showed no appreciable effect of the jacket, the saving of steam in the cylinder being fully equalled by the condensation in the jacket."†

The tests were considered conclusive as to that type, which, however, was a special design which has only been used experimentally.

A somewhat similar experience is reported to the writer by Mr. F. H. Ball. A saving was measurable, but not enough to be considered important. These compound engines, are, therefore, not, in this respect, improved by the addition of the jacket.

Mr. E. D. Leavitt, who has had an exceptionally extended experience, and unusual opportunities of observation, states: "The steam-jacket is of really economic value in my own experience, this fact being clearly proven by the Lynn pumping engine, and in that instance, the value is from 10 to 12%. By consulting the reports it will be seen that after the new cylinders were put in (the old ones having been cracked and the jackets off) the yearly duty went up from about 90,000,000 to over 100,000,000. No change was made except in the jackets. On engines of 100 H.P. and upward, where fuel is of any account, steam-jackets should be employed on simple, compound, triple, and quadruple-expansion engines, and on all the cylinders." "I have tried compounding a jacket and find it a good plan, whether the piston speed be high or low."‡

But Mr. Leavitt, some years ago, in testing the Hecla hoisting engine, obtained the lowest water rate, 16 lbs., with jackets shut off; the case was regarded, however, as exceptional and singular.

The experience of Mr. John W. Hill, which has included an unusual number of engine trials of all types and under all ordinary conditions, leads to the following conclusions: §

(1) The jacket is certainly of benefit when used under proper conditions, and may be expected to improve the economy from 8 to 10%—perhaps more in some cases.

* Private Correspondence, December 30, 1889.

† Private Correspondence, January 2, 1890.

‡ Private Correspondence.

§ Correspondence.

(2) The jacket is adapted to low-speed, high-expansion engines. With compound engines, which operate at piston-speeds of 100 to 240 feet per minute, the jacket is effective even when the expansion ratio is as low as 4; and in slow-speed engines, with expansion ratios of 12 to 20, the jacket is indispensable for high economy. With simple engines, and expanding 4 times or more at low piston speeds, no gain would be anticipated from the use of the jacket, and the economy might even be higher without it.

(3) High steam pressure, with wide range of expansion and temperature, would make the jacket more desirable than the use of low steam and small differences of temperature, the purpose of the jacket being to reduce the losses due to the differences of temperature of the cylinder walls and the expanding steam.

(4) The higher the ratio of expansion, the greater the need of steam-jackets; while, with low ratios of expansion, and high or moderately high speeds of piston, the value of the jacket is doubtful and, from his experience, he would not recommend it.

(5) With superheated steam, the jacket is comparatively unimportant, and the less so as the superheating is the more effective in checking cylinder-wastes.

(6) With compound engines, with slight or no expansion in the high-pressure cylinder, the jacket should be omitted there, but applied to the low-pressure cylinder; in triple-expansion engines, the jacket should be applied to the intermediate and low-pressure cylinders, even with high speed.

Professor Schröter deduces from his work on the triple-expansion engines at Augsburg * very instructive results relating to the efficiency of the jackets, and from this work and from the results of his tests of the jacket efficiency on a small engine of the Sulzer type in his own laboratory, he concludes: †

(1) The value of the jacket may vary within very wide limits, or even become negative.

(2) The shorter the cut-off, the greater the gain by the use of a jacket.

(3) The use of higher pressure in the jacket than in the cylinder produces an advantage. The greater this difference the better.

(4) The high-pressure cylinder may be left unjacketed without great loss, but the others should always be jacketed.

* *Zeits. des Ver. Deutsche Ing.*, xxxiv., s. 7.

† Correspondence.

APPENDIX.

DATA ABSTRACTED FROM REPORT OF A COMMITTEE OF THE BRITISH INSTITUTION OF MECHANICAL ENGINEERS.

Type of Engine.	Size.	p.	r.	I. H. P.	Rev.	Gain %	Date and Authority.
Simple. Non-Con.				120		8-10	Coms. de M. A.Vap. Thomas, 1846.
"	2″ x 8″	115			208	23	J. F. Inst.. 1859.
"	7 5/16″ x 26¼″	45	2.7	4.9-6.4	45-48	21.4	B. Donkin, 1874.
"			1.8-21	78.9		19.5	Cornut, 1874.
"			18-21	136		13.8	" "
"			10			19.1	" "
"	34″ x 39″	68.5	3.5-4.4	170-212	46.7-53.2	10.3	Emery, 1875.
"	16″.54 x 23″.62	62.5	3.4	55.7-59	99-102	9.3	Brodie, 1881-3.
"	16″.54 x 23″.62	57-59	3.3	54-55	100-101	7.25	" "
"	16″.54 x 24″.62	55	33	44.6-48	94-101	16.3	" "
"	16″.54 x 23″.62	50	3.4	40-41	94-95	15.9	" "
"	16″.54 x 23″.62	76-78	5.1	47-50	93	9.6	" "
"	16″.54 x 23″.62	75-78	5.1	46-54	92-94	22.4	" "
"	21″.65 x 43″.31	110	6.2-71	143-148	61-62	19.7	Delafond, 1883.
"	21″.65 x 43″.31	110	6.2	148-178	62	22	" "
"	21″.65 x 43″.31	110	5	182-194	61-62	16.7	" "
"	21″.65 x 43″.31	110	4.4	217-237	63-64	16.7	" "
"	21″.65 x 43″.31	78	5.4	121-137	61-62	14.25	" "
"	21″.65 x 43″.31	78	4.8-5.1	136-137	61	11.46	" "
"	21″.65 x 43″.31	78	3.7-3.8	178-180	60-62	11.36	" "
"	21″.65 x 43″.31	78	2.9-3.1	204-209	61	9.23	" "
"	21″.65 x 43″.31	50	3.7-3.9	108	61	7.21	" "
"	21″.65 x 43″.31	50	2.5-2.8	142-147	61	7.32	" "
"	21″.65 x 43″.31	50	1.7-1.7	169-173	60-16	4.74	" "
	21″.65 x 43″.31	50	1	145-147	61	1.19	" "
Simple Condens.		51	6	16		30.32	Thomas, 1842.
"	15″.35 x 31″.5	50-58	20	6-7		27.5-41.7	Combes, 1843.
"	5″.25 x 10″	38-56	5.3	0 93-1.13	60-61	31.2	Isherwood, 1863.
"	90″ x 117″	12	1.7	352-361	7.9-8.1	2.7	" "
"	70″ x 110″	36	4	153-161	12	12.9	Donkin, 1870.
"	7 5/16″ x 26½″	45	1.8	7.7-9.3	37-40	19.4	" "
"	20 1/16″ x 31¾	80	10	56-83	55	23.7	Hirn, 1873.
"	26″ x 30″	17	3.4	39-42	37	32.4	Rennie, 1873.
"	26″ x 30″	17	3.4	40	40	31.3	" "
"	25″ x 24″	78-79	5.2	89-116	47-54	11.8	Emery, 1874.
"	34″ x 30″	13-15	1.5-2	87-98	40-43	16.3	" "
"	34″ x 30″	36-45	2.2-6.1	123-255	43-58	6.1	" "
"	34″ x 30″	61-72	4.2-7.8	185-248	52-69	11.5	" "
"	34″ x 30″	64-65	4.5	269-282	61-62	9.5	" "
"	24″ x 48″	67	5.7	128-158	49-50	21.5	Merwin, 1878.
"	24″ x 48″	68	12.4	81-100	50	30.3	" "
"	30″ x 42″	30-60	3.1-4.7	170-191	53	15.5	Fletcher, 1877.
"	39″.37 x 70″.87	60-62	13-15	166-174	27-28	17.1	Farat, 1879.
"	32″ x 66″	41-42	3.8-4.3	123-124	20	16.6	Mair, 1882.
"	21″.65 x 43″.21	110	11-14	109-141	59-60	26.6	Delafond, 1883.
"	21″.65 x 43″.21	110	10-11	129-160	59-62	25.1	" "
"	21″.65 x 43″.21	110	8-10	155-161	60	23.0	" "
"	21″.65 x 43″.21	110	6.4	186-212	58	20.0	" "
"	21″.65 x 43″.21	89	9.3-12	112-126	60	16.7	" "
"	21″.65 x 43″.21	89	8.7-11.3	124-134	60	17.7	" "
"	21″.65 x 43″.21	89	6.8-73	150-176	60	18.8	" "
"	21″.65 x 43″.21	89	5.9	175-193	58-60	12.0	" "
"	21″.65 x 43″.21	89	4.8-5.9	193-194	60	14.3	" "
"	21″.65 x 43″.21	64	10.8	85-92	60	9.4	" "
"	21″.65 x 43″.21	64	8.2	115-117	60	7.8	" "
"	21″.65 x 43″.21	64	5.4	150	59	4.0	" "
"	21″.65 x 43″.21	64	3.9-4.4	172-175	58-59	3.9	" "
"	21″.65 x 43″.21	64	3.6	186-194	59	1.5	" "
"	21″.65 x 43″.21	50	11-12	69-76	60	6.6	" "
"	21″.65 x 43″.21	50	7.6-7.8	94-96	59	4.5	" "
"	21″.65 x 43″.21	50	5.6-5.8	120	60	3.2	" "
"	21″.65 x 43″.21	50	4-4.2	140-152	60	0.47	" "
"	21″.65 x 43″.21	50	3.2	165-179	60	0.56	" "
"	21″.65 x 43″.21	36	4.6-4.7	106-111	60	3.3	" "
"	21″.65 x 43″.21	36	2.2	160-162	61	2.9	" "

Type of Engine.	Size.	p.	r.	I. H. P.	Rev.	Gain %	Date and Authority.
Simple Condens.	21″.65 x 43″.21	36	1.7	180	61	0.61	Delafond, 1883.
	21″.65 x 43″.21	36	1.0	182–199	60	8.0	" "
Comp. Condens.	16¾″ 29¾″ x 49¼″ 66¼″	56	4.3		24	23.5	Hirn, 1855.
"	7 5/16″ 14″ x 26¼″ 36″	41	5.5	17–20	36	28.6	Donkin, 1859.
"	13 3/16″ 24″ x 39⅝″ 54″	41	11–14	28–46	33	31.	" "
"	13 3/16″ 24″ x 39⅝″ 54″	41	12	37	33	13.7*	" 1868.
"	7 5/16″ 14″ x 26¼″ 36″	45	10.3	10–17	37–44	38.6	" 1870.
"	20 3/16″ 31½″ x 51⅛″ 70⅞″	60	13	119	25	4.5	Lhoest, 1873.
"	16″ 25″ x 24″	80	6.7–7	77–99	48–53	11.7	Emery, 1874.
"	6″ 10″ x 12″	42	4.45–10–78	6.7–10.5	98	25.5†	Donkin, 1881.
"	6″ 10″ x 12″	42	5.38–16	10–12		29‡	" "
"	6″ 10″ x 12″	42	5–10	7–13		33.9§	" "
"	19″ 25″ 25″ x 24″	80	7	157–168	55	15.4	" 1887.
"	6″ 10″ x 12″	24–34	3.2–3.6	7–10	96	23.7	Kennedy, 1889.
"	6″ 10″ x 11″	45–43	3.2–4.5	9–10	97–101	24.4	" "
"	6″ 10″ x 21″	63	8.7–8.9	9	98–99	13.9	" "
"	6″ 10″ x 12″	62	6	10–12	97–99	23.9	" "
"	6″ 10″ x 12″	65	9–11	7–8	75–99	35.2	" "
"	6″ 10″ x 12″	73	3–3.6	13	96	4.4	" "
"	6″ 10″ x 12″	75	8.5–9	27–36	96	16.8	" "
"	6″ 10″ x 12″	75	6.4–6.6	10–11	99	24.2	" "
"	6″ 10″ x 12″	80	8.3–8.4	10–13	97	38.2	" "
"	6″ 10″ x 12″	80	7.6–8.6	10–13	96–97	11.8	" "
"	6″ 10″ x 12″	80	7.6–8.6	10-13	96–97	21.3	" "

DISCUSSION.

Prof. J. E. Denton.—I gather from this paper that it is the outcome of a desire, expressed during the discussion of the author's former paper presented at the New York meeting, to know upon what particular experiments was founded the statement that steam-jacketing could be relied upon to secure about 20% less steam consumption than is common to engines of good design and first class economy when operated without steam-jackets. The author, in submitting the various facts and opinions set forth in the present paper, intends the latter to be an answer to the questions put forth in the discussion referred to, namely:

* Steam in jackets only 4 lbs. pressure.

† Steam in H. P. jacket only.

‡ Steam first in H. P., then in L. P. jacket. § Steam in both or all off.

Where is the experimental proof of a saving of 20% due to steam-jacketing? and, judging from his quotation from his previous paper and the foot-note given in page 463 of the present paper, it would appear that in his opinion the contents of this paper still warrant a belief in the value of steam-jacketing, in "simple mill engines," amounting to 20% of the steam consumed in such engines unjacketed. As I am unable to acquiesce in this view from a study of the paper, and as the author states that he has other data in his hands, which, together with a "philosophy" of the cause of the saving due to a steam-jacket, he may, under proper inducements, present in a later paper, I submit the following criticisms, regarding some of the principal data contained in the paper, with a view to stimulating him to present us with another paper, which shall aim at discussing and settling the many conflicting facts about the subject of jacketing, as they are viewed from the standpoint of those whose convictions are opposed to the belief that steam-jackets, in every-day practice, are a sufficiently important element of economy to warrant the expense of their construction and maintenance on the ground of *economy* alone.

The opinions of Watt were founded upon the action of the single-acting pumping engine, which found its most economical form in the Cornish pumping engine. Admitting that the steam-jacket acted beneficially with this engine, the peculiar conditions of the latter make its record inapplicable to prove anything of importance regarding the action of jackets on modern double-acting engines. The early experiments quoted, up to 1860, are too crude to warrant confidence in them as positive proof regarding the influence of jackets, as the conditions, with and without jackets respectively, are not shown to be sufficiently identical for this purpose. The best of these early experiments is probably that of Hirn, in 1855. If the original paper of this experiment be studied at the present day, the crudities apparent in the methods and circumstances attending the measurements are quickly seen to be such as to greatly discount the character of the results for accuracy. For instance, the indicator cards were so discordant amongst themselves that they were discarded as a basis of power. The latter was adjusted through the medium of a water-wheel, previously tested with a Prony brake. Yet Mr. Hirn asserts, in the paper, that he regards the Prony brake as an unmanageable device for use to absorb the power of the engine directly. The data are on the whole obscure, there being no positive evidence that the ratios of expansion and speed were not different, with and without the jackets respectively. On page 471 of the paper we reach the period of 1860. The quotation regarding the Greene engines, with which the author was associated, has no force unless some rate of evaporation be assumed for the boilers. If the latter were of maximum economy, the steam consumption would be simply that attainable every day on such engines without steam-jackets. Referring to Professor Rankine's opinions, it is quite possible that the latter had available to him, up to the time of his death, only the compound engine performances of Elder's jacketed engines, which used steam superheated upward of 140° above the boiling point due the admission pressure. In the writings of Elder and Rankine, distributed among the various technical literature of about the date of 1860, there is clear evidence that cylinder condensation was entirely prevented by superheated jackets in those engines with which Rankine was permitted to experiment. He probably based his opinions largely on this experience, as during the ten years of his life succeeding this date little opportunity existed for any accurate knowledge of steam consumptions on marine engines. Indeed, it is only within the last few years that there have been any reliable tests of marine engines in actual service. The idea suggested, in the paragraph on page 472, that Rankine always attributed initial condensation —that is, condensation during admission—to the existence of liquefaction action due adiabatic expansion, is a faint echo of a criticism which the author, in copy of certain other writers, has made upon the soundness of Rankine's understanding of the phenomena of cylinder condensation. I believe it is quite possible to regard Rankine's idea that cylinder condensation during admission is initially attributable to the adiabatic liquefaction, as the most comprehensive view of the subject of cylinder condensation, and that, could a jacket prevent this liquefaction, the latter would be as important to economy as Rankine believed it and found it to be in Elder's compound superheating engine of slow speed of rotation. But in the high speed of rotation, compound screw-engines, which had their development after Rankine's death, the steam-jacket does not figure as an important adjunct to economy, and if Rankine could have lived to study their performance, he could easily find reason to agree to this fact, without laying himself open to the charge of not understanding that the principal loss by cylinder condensation occurred during admission, a charge which is quite ridiculous in view of his careful analysis of the Michigan experiment, coupled with his unparalleled accuracy and comprehensive methods of study and expression. It is to me a matter of great satisfaction that one of the few physicists competent to discuss the matter is shortly to present a brief paper, pointing out that Rankine's views on cylinder condensation, as expressed in his treatise on the steam-engine, or in his memoir of John Elder, are, in view of the still unsettled condition of our understanding of the laws of cylinder condensation (which, in spite of the laudable labors of Hirn, Dwelshauvers-Dery, Clark, Donkin, Isherwood, and others, have made no progress in the last thirty years), the most comprehensive—that is, take cognizance of more physical possibilities—than any other statement regarding the matter. Referring to the influence of steam-jackets on modern compound marine screw-engines, the experiments of Chief Engineer G. V. Sloat, of the Old Dominion Line, are a good illustration. He selected a set of engines of about 1,500 H. P., cylinders 34 × 54 × 5 ft. stroke, making 60 revolutions with steam at 90 lbs. above the atmosphere, and live-steam jackets on the barrels of both cylinders. The jackets were made by the aid of liners or bushings, and it was well known that the joints of the latter were very liable to leakage. But, in his case, it was made certain that there was no such leakage. Also the drainage of the jackets was thoroughly insured by receiving the condensed steam in a reservoir some two feet below the bottom of the cylinders, whence it wasted to the hot-well at a rate which prevented the accumulation of water above a few inches of depth in the reservoir, its height being shown by a water-glass, and a particular point being made of maintaining the level of the water nearly at a given point. Each jacket was supplied with its steam by a pipe 1½ inches in diameter. The ship was run many voyages with and without the jackets alternately. There was no saving in coal consumption; in fact, the little difference in the latter which could be distinguished was unfavorable to the jackets. Neither the indicator cards nor the revolutions evidenced any difference in the mean effective pressure. The steam-jackets have been abandoned in many similar ships because of the trouble of maintaining the joints tight which connect the jacket with the cylinder, and this was finally the case with this particular ship, but during the experiments there was no such imperfection to be charged against the accuracy of the results. Passing to page 473 of the paper, the second experiment quoted, as due to Mr. Hirn, brings us to the date of 1873, when this authority is quoted as proving that two similar simple engines showed a saving due the jacket of 24%. The same experiments are quoted in the report of the British Committee of the Mechanical Engineers, page 718, as having the same ratio of expansion, which is given at 10. I find, in Mr. Hirn's treatise on thermodynamics, that these engines did not have equal ratios of expansion, but that the unjacketed engines expanded the steam 14 times, as against 11 times in the case of the jacketed engines. Fearing some typographical error in the book, I have referred to the original paper in the "Bulletin de Mulhouse," and there find the same discrepancy in the ratios of expansion, a discrepancy which may in itself account for certainly a large part of the difference of steam consumption attributed to the influence of the jacket, and furthermore, the actual consumption per horse-power of the jacketed engine was no less than that given in Barrus' tables for a good non-condensing unjacketed engine under the same conditions.

In generalizing regarding the action of a jacket, Hirn, apparently on the basis of his experiments with these two engines, concludes that the jacket increases the area of the indicator card about 25% for the same total steam consumption. He appears to find that the exponent of volume of a hyperbolic expansion curve is changed from about .95 to .85 by the influence of the jacket. Modern experiments, however, afford abundant evidence that such variations of exponent are obtained without the use of jackets. For example, J. G. Mair's tests of a single condensing engine without jackets give quite as low an exponent as found by Hirn with jackets, and Mair's compound-engine tests afford exponents varying from .83 to greater than unity, and the lowest value is for an engine without jackets, using steam at 90 lbs. absolute at nearly 700 feet of piston speed.

Referring to Hirn's experiments with compound engines, as given, for instance, in the article on steam-engines of the *Encyclopædia Britannica*, the conditions with and without jackets differ, in that the ratio of expansion is about 7.8 without jackets, and about 9.8 with jackets. The results show about 10% less steam consumption with jackets. The engines were of the Woolf type, run at 34 revolutions. While I do not doubt that under the circumstances the saving due to the jackets may approximate 10%, have we any right to conclude that the conditions were sufficiently similar to establish the real gain due the jackets? The only other jacket data given in the *Encyclopædia* article are the results of Mair's experiments with a simple condensing engine, using steam at about 60 lbs. absolute and making 20 revolutions per minute. These results show that at 3.8 expansions, with jacket, the steam consumption per indicated horse-power is 26½ lbs., while at 4.3 expansions, with the jacket, the consumption is 22 lbs. Waiving the difference in the ratio of expansion, there should be a slight reduction of the figure without jackets, due to a difference of back pressure, which the original records make evident, but taking the figures as they are given, the saving due to the jacket is only 16%. Neither of the sets of data given in the article warrant the conclusion that even at the slow speed of revolution at which the engines were operated was there 20 to 25% of saving by the use of jackets, and yet the writer of the *Encyclopædia* article generalizes with regard to the subject of jacketing by stating that this amount of saving is to be relied upon, and I take it from the remarks of the author of the paper under discussion, made in closing the discussion at the New York meeting, that he was largely guided by the *Britannica* article in claiming 20% saving for jackets on simple mill engines. Mair's tests are worthy of every respect at the hands of experts as regards accuracy, but the conditions under which his engines were operated, principally as regards speed of rotation, do not represent American mill practice, either in simple or compound engines, as a speed of upward of 50 revolutions is now practically universal. Referring to page 490 of the paper under discussion, where Mair's tests are again quoted as proving that the use of a jacket was necessary to secure the low consumption of 14.8 lbs., as against 19 to 26 lbs. without the use of the jacket, it should be stated that the cases without the jacket refer to Woolf compound engines, expanding steam 8 and 9 times, while the jacketed engine was of the Receiver type, expanding 13½ times. Moreover, the unjacketed case giving 26 lbs. is considered by Mair as somewhat unreliable in view of imperfect drainage of jackets. Mair's case of 14.8 lbs. of steam per indicated horse-power, obtained at 24 revolutions per minute, is, so far as I know, unparalleled by any record of unjacketed engines at so slow a speed.

Referring to the marine testimony, page 482 of the paper under discussion: The fact that many steam-jackets on marine compound engines have been made by the use of a liner or bushing, which leaked seriously at its junctures with the cylinder, accounts for an increase of speed when the jackets were supplied with live steam, since there would be a gain of mean effective pressure due to the admission of steam to the cylinders after the valve had closed the admission ports. It was the observance of an increase of three revolutions per minute by the admission of steam to the jackets on a compound engine of a transatlantic steamship that led Mr. J. Baird to jacket the cylinders of the steamships *Hudson* and *Knickerbocker*. The engines of these ships were single condensing cylinders about 4 feet diameter by 6 feet stroke, making about 60 revolutions per minute and using steam at about 90 lbs. pressure, cut of at $\frac{1}{12}$ of the stroke, a fact which made them a novelty at the time of their advent, about 1873. Live-steam jackets were cast to the cylinders throughout their length. Most careful comparisons were made of the performance, with and without jackets respectively, during many voyages between New York and New Orleans. No difference in the revolutions due to the presence of live steam in the jackets could be distinguished, and no economy in coal consumption was found. Mr. Baird tells me that the trials were conclusive, as he was so sanguine about an advantage being derived from the jackets that he finally placed the trials in charge of a special engineer, who made it a study to give the jackets every aid as regards drainage, freedom from air, etc. It should be stated, however, that the steam used in these trials reached the cylinder superheated 65° Fahr., and that the steam blown from an indicator cock was apparently slightly superheated. The water consumption of these engines, tested within the last few years, without steam in the jackets, was about 20 lbs. per indicated horse-power, and the cylinder condensation at cut-off was less than 10%. During these trials the cut-off was about $\frac{1}{12}$, but as there is about 13% clearance due to the use of poppet valves on so short a cylinder, the real expansion is only about 5 times.

In order to reach useful conclusions regarding the value of steam-jackets, I believe distinctions somewhat as follows must be made:

First. Opinions of practical men, resulting from general convictions, not based upon a careful measurement or comparison of performance, must be almost entirely abandoned. For example, among the quotations of the author is one of this class which claims that the jackets of certain high expansion compound pumping engines of slow rotational speed have shown an economy of 15 to 20%. The writer, upon investigating the origin of this opinion, found it to rest upon the performance of a certain pumping engine which, having broken its jacket, and thereby having been obliged to operate several years without the use of steam in the jackets, was finally provided with new cylinders with jackets. The improvement in duty with the new jacketed cylinders was the basis of the opinion quoted by the author of the paper under discussion regarding the saving of jackets. But the fact was that before the engine ceased to use the cylinders of the broken jacket, the fracture due to the breakage allowed a serious escape of live steam each stroke of the engine, so that the improvement in duty was the sum of the saving due to stopping the leak and whatever amount, yet unknown, was properly due to the restoration of the jacket.

Second. When tests are made, with and without jackets, they must either be of sufficiently long duration and under a sufficiently steady load to make the probable error in the water consumption less than the probable difference between the economy with and without jackets; or, if of short duration, the heat must be measured and balanced at each end of the engine. For example, the very striking series of tests published by Delafond, and referred to on page 486 of the paper under discussion, must be classified as experiments of a very low order of accuracy. In the first place, a load of upward of one hundred horse-power was absorbed by an Appold Prony brake, which, when it is stated, as in this case, that water had to be applied to the brake pulley to maintain its temperature sufficiently low, is an utterly incompetent means of securing a uniform load for such a purpose as distinguishing between the economy of an engine with and without jackets respectively. In the second place, the steam was measured by the difference of level in a boiler only, and for periods of time ranging from an hour and three-quarters to thirty-six minutes. Moreover, the boiler was idle until the instant when the engine tested was cut off from another source of steam supply and suddenly connected with the test boiler. The latter must therefore have been in unsteady action for some time at the commencement of the test. Again, at the end of the allotted time the test engine was cut loose from the test boiler and the depreciation of water level in the latter restored by pumping water rapidly into it. As its evaporative functions must have been suddenly annulled, the effect of such a sudden introduction of water must certainly have affected the general temperature of the contents of the boiler so as to render the measurement of steam consumption liable to a serious error. No data are given in the original article to enable one to judge as to the probable error of the determinations, but it is shown that attempts to draw a balance of heat through a measurement of the heat rejected gives the wildest sort of results. It is a surprise to me that such zealous and rigorous workers as have charge of the jacket research of the British Mechanical Engineers should consider the Delafond experiments comparative in value with most of their other published data as given in their first report, from which the author of the present paper has compiled the table on page 500.

Third. The conditions of loading must be the same under test as when the engine is worked commercially. For example, Mr. Borodin's tests showed undoubted advantage due to jacketing with an artificially light load on single locomotives, but the same locomotives, run over the road under regular conditions of service, failed to indicate that the use of steam in the jacket exerted any definite or tangible influence upon the steam economy.

Fourth. Experiments must be made with engines of the size used in practice. The experiments of Kennedy on the small compound engine at University College, as given in the first report of the British Research Committee, and the experiments of Reynolds with his triple-expansion engine at Owens College, certainly show striking effects due to jacketing; but these are small engines, and the same reason which makes cylinder condensation exert a greater proportional influence in small engines than in large ones may cause the jacket to produce an influence in small cylinders which is not realizable in engines of the size in use in industrial establishments. It is possible that the zeal and opportunities of the workers with small engines, in experimental laboratories, may lead us to new and interesting views of the action of steam-jackets; but, meanwhile, we certainly lack data to prove that on the modern high-class steam-engines, either compound or triple expansion, steam-jackets are an important source of economy for speeds of rotation upward of 50 per minute. For compound engines of the Receiver type, using steam of about 90 lbs. above the atmosphere, and expanding it about 9 times, we have the record of the engine tested by Longridge (London *Engineering*, 1882), in which a consumption of about 16.9 lbs. of steam per indicated horse-power was obtained, either with live steam about the barrels of both cylinders and the receiver, or with no steam in any of the jackets, or with steam in the jacket of either cylinder or the receiver separately. The tests were duplicated, and the greatest variation of consumption amidst all these conditions was less than 1%. The heat balance was within 2% in every case. I do not understand why this apparently excellent investigation should be omitted from the report of the British Research Committee. For engines of the Receiver type, expanding steam upward of 16 times, with about 120 lbs. boiler pressure above atmosphere, and making about 50 or more revolutions per minute, the steam consumption in several reliable instances in America has been found to be less than 14 lbs. per horse-power, either when both cylinders and receiver are jacketed with live steam, or when the jackets serve as steam-chests for their respective cylinders, or when no steam is used in the jackets around the barrels of the cylinders and no steam around the receiver, but with the heads of each cylinder serving as steam chests. (This was the case with the Pawtucket pumping engine.) For triple-expansion engines using steam at 140 lbs. boiler pressure above the atmosphere, and expanding about 18 times, we as yet have no data without jackets. An excellent test of such a type of engine is that referred to, page 499 of the paper under discussion, as having been made by Professor Shröter at Augsburg. A similar engine has been recently tested by Mr. J. T. Henthorn at Providence, R. I. In both cases a steam consumption of about $12\frac{3}{4}$ lbs. per indicated horse-power was found, with jackets on all cylinders and receiver, using live steam. In referring to Shröter's test, page 493, the author quotes its consumption at 15.2 lbs., and its ratio of expansion as 24. Both of these figures are in error. The former occurs in the opening paragraph of Shröter's paper, when he speaks of the consumption of the engines of the steamship *Meteor*, tested by Professor Kennedy. The ratio of expansion, taking clearances into account, was only about 18. In conclusion, referring to the idea that 20% is to be saved by jacketing first-class steam-engines of large size and high rotational speed, it should be remembered that the best single condensing engines, as built by Corliss and others, expanding steam about 6 times at 90 lbs. boiler pressure, and in fair but not perfect order as regards tightness, can be relied upon to yield an indicated horse-power, unjacketed, with a consumption of 19½ lbs. of steam. The advent of the compound engine in the American mill districts reduces this consumption 20% by expanding steam 16 times, and the use of a second engine of twice the size of cylinder to act as the low pressure portion of the plant. If any one knows how to secure 20% of saving over the single unjacketed engine by simply jacketing it, he has a rich field before him in instituting such a saving among the many mills of New England still using the single-cylinder engine, but desiring to compete in economy with those who have adopted the compound engine, simply to save this same 20%. A "philosophy" regarding the action of the steam-jackets which will lead to an understanding of its application by any one skilled in the art, so as to save 20% of the consumption of first-class non-jacketed steam-engines, will be very acceptable.

Mr. Scott A. Smith.—About three years since I gave very searching thought to the question of the value of steam-jacketing in order to satisfy my own mind and to incorporate a simple statement of the reason for its value in a paper on the general subject of the steam-engine. The statement was practically this: that without the steam-jacket, in any given case, the water of condensation in a non-jacketed steam-cylinder goes out with the exhaust, and the heat contained in the water is lost; whereas, by the use of a steam-jacket, a practically equivalent quantity forms in the jacket space instead of in the cylinder, and returns by its own gravity, or is pumped by the power of the engine, back to the boiler, thus saving its contained heat. The paper was, at the time mentioned, published in the *Scientific American Supplement.*

I find for the first time the same idea expressed in Dr. Thurston's paper as having been given him by J. G. Lawrie, on page 470, commencing in the last paragraph. Mr. Corliss, by carefully conducted experiments, found the value of the steam jacket to vary from 7½ to 10%, according to the variations of the point of cut-off, within the ordinary limitations in the economical use of his simple condensing engines. His faith became so strong in the value of steam-jacketing that he used it even on non condensing engines of as small size as 50 I. H. P., and it is speaking within bounds to say that it is quite a universal thing in that establishment. I believe that the very philosophic mind of James Watt saw clearly the value of the steam-jacket, and that to the lesser yet still philosophic mind of Mr. Corliss its advantage was fully apparent. It is to be regretted if, through the limitations of thought and of language, the value of steam-jacketing cannot be fitly, fully, and clearly expressed in words.

Prof. R. H. Thurston. *—I think there is quite enough evidence in now, and do not see why the discussion may not be at once closed. The facts seem to me pretty well brought out, and each interested reader must draw his own conclusions. Individual opinion is of no importance, and I am as little inclined to urge mine on others as to forego the privilege of doing my own thinking in such matters. The discussion calls up one or two new points, however, and I think I may now add a word on the financial aspects of the case.

Taking up the points considered in order: The first speaker starts a little out of line. My original statement of my opinion —a matter of no importance, more than any other individual's, however—was *not* that "steam-jacketing could be relied upon to secure about 20% less steam consumption than is common to engines of good design and first-class economy," as he puts it, but that the saving "in ordinary cases" may be taken as averaging about that amount, an essentially different statement, and the interpretation of that phrase seemed to me obvious enough without further dissertation. Nothing was said by me, or suggested, about "first-class economy;" nor was it suggested that I, or any other man, believed that 20% could "always" be thus gained. Nor have I stated that I believed that "steam-jackets, in every-day practice, are a sufficiently important element of economy to warrant the expense of their construction and maintenance." As stated in the paper, it seemed to me well settled that "the effect of a steam-jacket depends upon the conditions of operation of the engine largely, and may be productive of marked advantage, or, under unfavorable conditions, of no important useful effect."

* Author's Closure, under the Rules.

The remarks about the crudity of early experimentation are, I think, perfectly correct. And inaccurate work was unquestionably carried to the height of absurdity at even comparatively late dates. It seems to me that they reached their climax in such reports as those of Wicksteed, of fifty years ago, in which he claimed a duty exceeding that of the best engines of to-day—and in excess, even, of those of a perfect, ideal engine working under the same temperature conditions—for engines of 50 and 80 inch cylinders, working in a $2\frac{1}{2}$-hour trial, at but 23 and 27 H. P.; and, also, in such reports as that of Rankine on the Thetis, in which a consumption of 1.018 pound of coal per horse-power per hour was seriously stated to have been obtained. I presume the fact of the slight experimental knowledge, in those days, of the wastes of the engine and their theory, may be taken as sufficient reason for and justification of such curiosities of engineering. Both Wicksteed and Rankine were among the ablest men of their time.

In regard to Hirn's work, however, knowing the man as I did through years of correspondence, which brought out strongly that scrupulous exactness of work which distinguished him, I am inclined to think that it would be the height of absurdity to suppose that inaccuracies in the work referred to could have been so great as to destroy the large margin reported in favor of the jacket, or any important fraction of it. The Prony brake on a water-wheel is a more manageable apparatus than on an engine, especially at low speeds; and the wheel itself is about as good a measure, a meter, of power and energy as anything that I know. At Holyoke, Mass., the whole business of the water-power company is based on water-wheels used as meters of both water and power.

In regard to the Greene engine referred to, the boilers used were those which were and are still standard. The case has little value, however, as positive evidence.

I think that the remarks about Rankine are mainly right, and may fairly be taken to represent the facts of his history. I think it evident, to-day, however, that his idea of the initiation of cylinder condensation by adiabatic condensation was erroneous, but was not aware that I had written "in copy of certain other writers," as my friend oddly expresses it. I shall be interested in looking them up. I well remember my disappointment when, in 1860, I bought my first copy of Rankine's book, and found, after I had gone through its theory of the steam-engine, that the results computed were, as it seemed to me, absurd when compared with those of practice then familiar to me, and that the book was, for my then purpose, useless. It was some time before the discrepancies were all explained and it became evident that he had developed a correct theory of the thermodynamics of the machine, and one which only required to be supplemented by the physical theory of heat-transfer to make it complete and satisfactory. I was in correspondence with Rankine for years, up to the time of his death, and think that the Michigan experiments, more than those previously made by Clark and Hirn, had begun to reveal to him the real action of the real engine; but his treatment in his book was never changed—and has not been to this day. It still assumes to compare ideal cases with real performance.

I fully agree with the speaker that that great man, had he lived, would have been found fully conversant with the later developments of research, and most probably, I would add, among the first to incorporate them into his philosophy of the engine. It is not at all to his discredit that he did not "know it all" a generation ago. Some of us may, without discredit, admit that we do not know it all, quite, yet.

The facts here given us in reference to later research and practical experience are interesting. It is unfortunate that we could not have many more such. They all help us secure more definite ideas of the real physical phenomena involved. I am especially interested in finding such confirmation of my deduction: "High-speed engines derive less advantage from its application than slow-moving machines; and compound or multiple-cylinder engines are less dependent upon it for economy than are simple engines;" and, generally, that the nearer the wastes of the unjacketed engine fall to those due to the jacket itself, the less the advantage of that appurtenance.

I had, of course, observed the fact that it often happens that the ratio of expansion, in comparisons reported, is greater in the unjacketed engine than in the jacketed machine, and had accounted for it to myself by the facts that the constant load and speed of the engine caused it to work thus, as a consequence, primarily, of the larger proportion of work done in the case of the jacketed engine during its expansion period and accidental conditions. Culling out such discrepancies and apparent defects, however, I think there still remains a large amount of unquestionable evidence and much interesting and instructive fact. The result of such study of these experiments of earlier and later days often throws a doubt upon the exactness of the reported figures; but usually the probability of error would tell as often against as for the use of the jacket; but, as it happens, the evidence runs very largely the other way under the conditions which I have stated to be favorable to its employment.

The article contributed by Professor Ewing to the *Encyclopædia Britannica* does not state, or intimate, that he was led to give the figure, 25% saving, by the study of Mair's trials only; nor was I "largely guided by the *Britannica* article in claiming 20% as the saving by jackets on simple mill engines." That statement takes altogether too much for granted. The 20% stated by me was simply the figure which had been given gradual definition in my own mind in the course of thirty years of practice, experience, and reading and observation, and would naturally refer to the general results of experience during that time, not to to-day and to engines just set at work. Professor Ewing's statement of 20 to 25% was noted, later, as corroborative, though more radical than mine. What he *does* say is: "Experiments made with and without a jacket on the same engine have shown that jacketing may increase the efficiency 20 or 25%."

I agree, in the main, with the final summary of this disputant, not excepting the suggestions that either a "philosophy" or a practice which will save 20% on "first-class steam-engines of large size and high rotational speed," having no jackets, "will be very acceptable," and that the engineer who knows how to secure such saving among the many mills of New England "has a rich field before him." My own impression is—and I express it very diffidently under the circumstances and in such presence—that, as I have already remarked, "this accessory may, under different circumstances, prove a decided advantage, an unimportant appurtenance of the engine, or, even, under exceptional conditions, deleterious to efficiency."

Now as to the finance of the case: Take even the least favorable results of which the facts are exactly known to us. I imagine that the work on the Pawtucket engine was done as carefully, and the results reported as accurately, as those from any trials on our list. It would be interesting to ascertain whether the small effect of the jackets in this case, both on duty and in reducing the still somewhat heavy cylinder condensation, is due to defect of construction or of operation, or to a really high resultant efficiency of dry steam. Take them as showing that, with thoroughly dry steam, in a compound engine even, no more than 3% gain may be expected from the use of the jacket. Suppose such an engine working, as it ought, at least 6,000 hours in the year, the duty to average 120,000,000, the coal for I. H. P. per hour thus to be 1.63 lbs., and the total per annum, in round figures, 750 tons for 150 H. P. The saving of 3% on this weight, at $5 per ton, for such coal and in New England, would be about 12% on $1,000, an excessive estimate of cost added by the jacket; and the incidental advantages of reduced risks from water in the engine, of cracking by sudden changes of temperature on starting and stopping, etc., may be thrown in. I imagine, on figuring up the costs and returns, it will be found an advantage to use the jacket, assuming it properly designed and attached, in very many cases where ordinarily it has not been thought of. But on high-speed engines of large size, and especially on high-speed compounds of considerable size, I imagine it would usually be of questionable value. The above would mean a splendid investment.

But a gain of even 1% in New England in some cases may, I think it possible, prove worth accepting. Ten per cent. is a gain which can rarely, I think, be wisely rejected; for the necessary cost of the jacket is small, and its maintenance need not give rise to any expense if properly constructed and operated. I imagine that steam jacketing will have a place in steam-engineering for some time to come, under what are coming to be recognized as its proper conditions of action. The experience of Mr. Corliss, as he gave it to me at times during an almost life-long acquaintance, may perhaps be that of many other builders. He resisted the introduction of the steam-jacket, as he did that of the forged shaft and other innovations, for a long time; but he came to it finally, and improved his work continually. I am prepared to see at an early date, however, high speed of rotation, compounding, and superheating moderately, so common and usual in practice that the jacket may, in time, drift out of use again. It should be noted that, in cases above referred to, where the expansion without jacket is greater than with, the fact adds to the gain to be observed with jacketing, as the expansion is, in such cases, probably greater than is economical.

I hope that in this matter I shall not be classed with those referred to in the sub-title of Whistler's famous book, in which the author speaks of instances "in which the serious ones of this earth, carefully exasperated, have been prettily spurred on to unseemliness and indiscretion, while overcome by an undue sense of right." I am satisfied to accept facts, whatever their complexion or source.

CCCCXXXVI.*

TOPICAL DISCUSSIONS AND INTERCHANGE OF DATA.

XXIID MEETING, RICHMOND, NOVEMBER, 1890.

No. 436-85.

Is there any reason why corrosion should be more active in one place rather than another, inside a steam-drum properly piped to connect several boilers in a battery ?

Mr. Jas. McBride.—The accompanying illustrations (Figs.

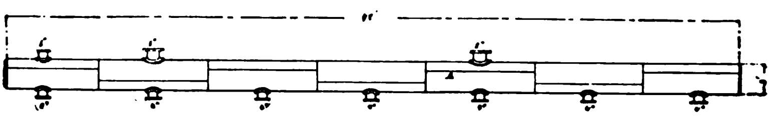


Fig. 177.

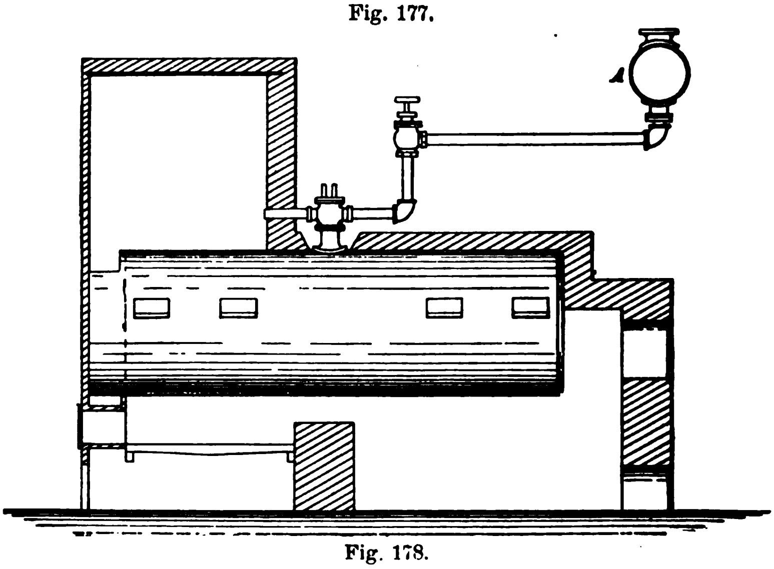


Fig. 178.

177, 178) show the setting and connecting drum of a return tubular boiler such as we use in our chemical works.

I have noticed a very great corrosion in the drums in the last

* Presented at the Richmond meeting of the American Society of Mechanical Engineers (1890), and forming part of Volume XII. of the *Transactions*.

few years. I watched it carefully until it reached a point which I considered dangerous, and I had new drums made to replace the old ones. I cut out a number of specimens from the old drums, which are exhibited at the meeting, and I would like the members of the Society to examine them. I found that the corrosion was not confined to any particular place. Immediately above the points where the steam entered from the several boilers I found bad corrosion, spots corroded almost through, and in one case entirely through. At the point *A*, on the side of the drum, is a row of rivets, near which the plate is badly corroded. Other parts of the drum—sides, bottom, and top—are badly corroded, as shown by the specimens. The drum is made of iron plate. A piece of 4-inch pipe, taken out somewhere along in the line of steam-pipe, shows the same corrosive action, so that you will see that it is not confined to the top, or bottom, or sides. The water used in these boilers, the Ridgewood water, is said to be the best in America. I used this water exclusively, and a percentage of it is the water which is returned from the coils of my vacuum pans. I do not know exactly what percentage—probably 20% —maybe not so much as that. These boilers carry 60 to 75 pounds of steam. I do not think the boilers are overworked. We are evaporating, as near as I can remember, about 7 or 8 pounds of water per pound of fuel. There are seven boilers connected with this drum.

Mr. Allan Stirling.—I would like to have Mr. McBride repeat that statement. How many pounds of water to pounds of coal?

Mr. McBride.—Between 7 and 8 from and at 212°.

Mr. Stirling.—Per pound of combustible or per pound of coal?

Mr. McBride.—Well, per pound of coal, I think. I do not recall whether these are the figures or not. I am not sure.

The Chairman—The boilers appear to be 21 in all, in batteries of 7, each battery connected to its own separate steam-drum. The corrosion he speaks of is in many instances directly opposite the orifice in the top of the shell through which the steam passes into the drum. In some cases it is on one side and in some cases on the other side of the drum, and in some cases it is at the plates where the rivets are. The material is boiler iron and the action is one that is very peculiar, and one of which perhaps some other gentlemen here may have some experience that will throw some light upon it.

Mr. Geo. R. Babbitt.—I would like to inquire if the 20% of distilled water contained any grease?

Mr. McBride.—I think not. I do not think I use more than about a quart of oil in each engine per day. The cylinders are 30 inches in diameter, 5 feet stroke, running 53 revolutions a minute. There is not the slightest indication of any deposit of oil on the tubes or shell of the boiler, as usually occurs where large quantities of cylinder oil are used.

Mr. Babbitt.—There is no fat or tallow used at all?

Mr. McBride.—No, sir; none. I use the best cylinder oil I can get. There is no foreign substance that I know of, of any kind.

The Chairman.—It might be interesting to know if any corrosion takes place below the water-line.

Mr. McBride.—Not a particle. The sheets are as clean and bright when cleaned as the day they came from the mill. It begins at the point at which the steam leaves the boiler. We find the connecting pipe corroded and eaten up in places. The corrosion seems to commence at the boiler and continues right on to the end of the system of steam-piping.

Mr. Babbitt.—Not in the boiler-plates at all?

Mr. McBride.—Not in the boiler-plates at all; only in the drum.

Mr. Babbitt.—It occurred to me that possibly it might be some fatty or animal oils which caused the corrosion. Such oils will corrode iron. That has been my experience.

Mr. Carleton W. Nason.—I would like to ask Mr. McBride whether or not, in such extensive corrosion as that, there has ever been any analysis made of the water. Dealing in logwood as he does, there is a suggestion, at least, of the possibility of there being a leak of tannic acid into the water. That, of course, would be shown by analysis. As shown in the sketch, and on the piece of iron, the greatest corrosion has occurred directly opposite the inlet point where the particles of water would be carried against the drum. The amount of corrosion there is so excessive that it would seem necessary to look to some special cause other than what would be found in the water, there being nothing but Ridgewood water, and one would look for the same experience in other water.

Mr. McBride.—Yes, sir. I had our chemist analyze the water. I suspected something of the kind myself. I have had it analyzed at different times, but found no foreign substance in it whatever which we could think would produce that result. We would take a specimen of the water as it is drawn from the hydrant and take a specimen of the feed-water and analyze them, and we found no practical difference.

Mr. Nason.—Did I understand you to say that there is something like 25% of warm water going back into the boiler?

Mr. McBride.—Yes, sir; but that has been analyzed separately and we found nothing in it.

Mr. Paul H. Grimm.—I had a similar experience in the destruction of steam-pipes and fittings, and I have attributed those things at the time to entrained water in the steam passing through the pipes under the influence of wire-drawing, the particles of water being projected against the elbows. I have known large elbows to be cut through entirely. Now it has occurred to me, because the steam-drum is only 24 inches in diameter and the direct nozzle looks up to the upper side of the drum, that there may be entrained water enough in the steam to be projected against the upper side, and thus cause the destruction shown by impact of the entrained water against the inner surface of the steam-drum. I do not know how Mr. McBride takes off his steam from that drum. The water circulating around in the drum being agitated by the different currents of steam going in there, may possibly have a destructive influence. I know that it would, directly opposite where it comes in; at least, such has been my experience.

The Chairman.—The suggestion of the gentleman would be a very good one if it were not the fact, as stated here, that the same trouble occurs on the side of the drum.

Mr. Louis G. Engel.—I can corroborate what the gentleman has said. I have had the same experience. We have had elbows half an inch thick of cast-iron cut through inside of two months, and this is constantly occurring, due to the impact of the particles of water in vapor. Our practice is just the same as that of Mr. McBride, and we are half a mile from him. We use the distilled water in the same way from the pans, and probably a much larger quantity in proportion, and we have never had any pitting whatever. I think it might be due to what the other gentleman has said, that the water is thrown into a fountain by the entering steam. The water which collects in the bottom of the pipe is thrown up against the sides. In fact, I have taken some measures to prevent that very thing. Our boilers are piped as shown in the illustration (Fig. 179), which represents one battery of two Babcock & Wilcox boilers. Steam is taken out of the drums of the boilers *A A* and *B B*, into a settling drum, *C*, from which we take the steam out into a main by the pipe *D*. Now there is considerable water collected in this drum, *C*. I know it is there, because we have trapped it out. And I believe, when water is allowed to congregate there, it would be thrown up in a whirl by the entering steam, and our lack of evil effects is due to the size of the drum *C*. We have put nozzles in at *E E* of such a height that the entering steam would not agitate the water which we drain off. I do not see but that local abrasion would be a natural result of having a

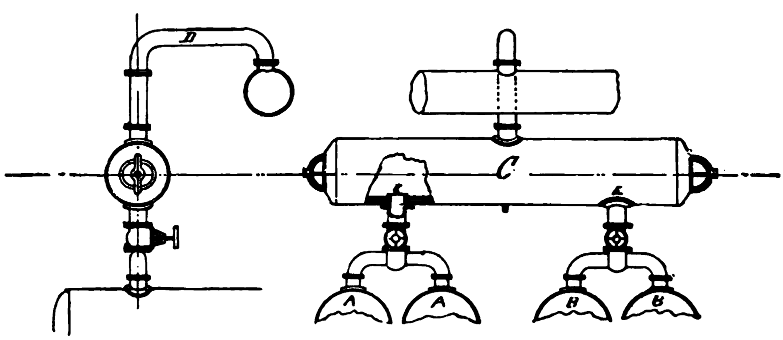


FIG. 179.

series of pipes enter the bottom of a drum, especially if the steam contained water and had been thrown into complicated currents by elbows and changes in the diameter due to putting screw fittings on to wrought-iron pipe.

The Chairman.—It would be interesting to know how the cast-iron of the fittings has been affected, as I take it that the elbows referred to by Mr. Grimm were of cast-iron.

Mr. McBride.—They are affected very much in the same way. I find that they corrode through, and, while they last four or five times as long as the wrought-iron pipe, they do perish just the same as the other. In reference to the water and steam being thrown out in a fountain against the shell of the drum, I found that above some of the outlets from the boiler the shell is good. It does not appear to be attacked above some of the outlets at all, except in very small spots. You will notice a piece of eight-inch pipe among the samples. You will find that it is eaten all around. The heads of the rivets where the cast-iron nozzles are riveted on are entirely eaten off. The cast-iron flange goes on there, and the rivets are driven on the inside. I venture to say there are probably not half a dozen of these rivets in the drum which are not nearly eaten off.

Mr. Allan Stirling.—I would like to ask Mr. McBride if there was any copper in his condensing apparatus where this condensed steam came from; if he has made any calorimeter tests of his steam to find if the steam was dry. There may be three causes for this: First, the presence of some action due to the copper; second; the impact of water in the steam; or, third, the presence—if this steam is very dry—of mechanical electricity developed in that way.

Mr. Charles Sperry.—As a case illustrating the corrosion of cast-iron elbows, I will say that on the steamer *Bristol*, of the Fall River Line, the steam entered the lower end of the steam-chest through a cast-iron elbow of about three feet in diameter. On taking this elbow off I found that, at the centre of the outer curve of the elbow, the iron was less than one-eighth of an inch thick. The thickness gradually increased toward each end and around to the inner curve of the elbow to about three-fourths of an inch.

Mr. McBride.—In answer to the gentleman's first question, I would say that the condensed water which I take back comes from copper coils, from copper pipes, and also from a brass coil in a Berryman heater. In regard to the second question, I would say that, last summer a year ago, I made over forty calorimeter tests for the quality of the steam, and I found they ran all the way from $\frac{1}{2}$ of 1% up, I think, in one case, to nearly 9%. Those were taken under all conditions of work, with no load, with heavy load, with high water in the boilers, with low water in the boilers, with heavy fire and low fire, and under all the working conditions, and the average of the moisture—if my recollection serves me—was $4\frac{8}{10}$%.

Mr. Wm. M. Barr.—Judging from the plates now on exhibition, together with the sample of pipe now being passed around for examination, it does not seem to me that the pitting or corrosion in either could be due to impact. The sample of plate indicates chemical action and nothing else, and so also does the sample of pipe. I am somewhat incredulous about the impact theory. I believe that it is entirely due to chemical action, and that the causes which lead to this chemical action can be traced directly to the feed-water.

Mr. Jno. T. Hawkins.—I am not at all incredulous as to the impact theory in this case. If the author of this topical query has made any investigation of the system of currents which are set up in these cases, possibly the particular located corrosion may be accounted for by him, although I do not think it at all necessary that we attribute it to entrained water. It seems to me, from an examination of the specimens, I should reason somewhat in this way: We know that in the interior of the boiler there is no such thing as a clean metallic surface. We know that the interior of both the steam and water spaces is covered with a coating of oxide, and we also know that currents of steam such as would be set up in those nozzles and in the drum would have the effect of removing that oxide, perhaps in an insignificant degree, for any short time; but, as boilers are running for weeks and months and years, if this oxide is removed in an insensible degree in certain places by the action of these currents during activity it becomes reoxidized when not in use, and this process, carried on for a long period of time, would produce the results shown. Now it occurred to me, from an examination of these specimens, that I could see just how the currents might be set up in this drum such as would produce exactly the effect in this plate exhibited, in which a channel is corroded along the seam, and I will endeavor to point it out on the black-board. In the first place, Mr. McBride says that the corrosion commences at this point, and that it is in this first elbow and then in that—the second from the boiler; and, finally, he finds the worst case, I think, directly over the opening into the drum. Now it seems to me that, with the steam passing out of the boiler into that first elbow, the first impact it makes is at the top of that elbow. That would be the first point where the apparatus would be worn away by the steam. Then it is deflected and strikes the second elbow, and then the third, and so on. Now we come to the case in which the corrosion occurs along the seam of the drum. We can readily suppose that the steam, as it comes against the top of the drum, is deflected both ways, and, in passing out to the engine, establishes a rapid horizontal current somewhere in that line of rivets. We may believe that there must be a great many currents set up which would produce these, effectually wearing away the oxide while in action only to become reoxidized again when the action ceases. I would like to ask Mr. McBride if he has, in investigating this subject, figured out to himself whether the currents of the steam in these parts, as entering at the point shown and leaving at a point distant from it, would establish where the point of wear or apparent corrosion should take place.

Mr. E. F. C. Davis.—The action of steam suggested by the gentleman is very similar to the action of mine-water, and I have often been struck with the same action in mining-pumps and mining-pipes. Wherever the water is perfectly quiet, no matter how acid it is, the iron will protect itself by a scale, or what you might call almost a scab, of oxide and other foreign matters accumulating with the oxide, and that iron will remain for years. I have known a pump which was in an abandoned mine which remained under water twenty-one years, and when we had occasion to take that pump out, we found the iron was in a perfect state of preservation. I put that pump in another colliery and it worked very well, and wore out, as all pumps do. That is an illustration of how long it will stand where there is no cutting action of the current. But in pumping the same kind of water wherever there is a bend or any disposition of the water to run past any surface so fast that the coating of oxide cannot accumulate, the action of the water bares the surface of the iron, removes the coat of oxide, and the iron is eaten so very rapidly that it is almost impossible to pump mine-water under those conditions at all.

Now that seems to me to be a very good way to account for this action. If we cannot arrange to have the currents of steam to suit the points that wear out; or, on the other hand, if that is the case and this is perfectly pure steam, why is it that all our steam-pipes do not wear out all the time? Now we all know that steam-pipes running to shop-engines have lasted for a life-time, elbows and all; so that the argument does not seem to hold good all through.

The Chairman.—With regard to the pump of which you spoke, in the mine, the absence of oxidation, or, rather, the continuation, was it not due to the fact that there was no air there, and that the pump was entirely submerged?

Mr. Davis.—No, sir; wherever there is sufficiently rapid action of the water the iron wears out very fast.

The Chairman.—Was the pump in operation?

Mr. Davis.—Yes, sir; but while the pump was still there was no action whatever. The pump which was under water twenty-one years, was, of course, an extreme case.

Mr. F. Meriam Wheeler.—I am inclined to agree with Mr. Barr, that chemical action has something to do with this trouble. These samples of iron certainly show it, and makes me think very much of the looks of condenser tubes which have been injured by galvanic action.

Possibly the water Mr. McBride uses may be impregnated with acid, and cause pitting and eating away of the iron. I know that salt waters having certain chemical qualities will often make trouble with surface condensers.

Mr. Scott A. Smith.—The disintegrating action which I see has taken place on this piece (a piece of the boiler-shell including a seam) seems to me to be easy of explanation. There is going on in the iron of all steam-boilers what may be called an insensible vibratory motion which would be accented in each sheet near the seam, as there it would tend to terminate. This furrowing, or disintegration, usually takes place at the water-line, and would be aided by this action *when* there is present, in the water used, any substance having a tendency to unite chemically with the iron. This so-called chemical action may be a part of a galvanic action, *providing* that there is connected with the boiler, in some manner, either copper or brass.

Mr. Geo. R. Henderson.—That is very similar to a case at Norfolk. The lower portion of the tubes covered by the water was perfectly sound, but the upper part was eaten away in a few months. I think it would be interesting if Mr. McBride would state if there was any scale formed on the inside of the boiler.

Mr. McBride.—No, sir.

Mr. Nason.—Is there any indication of water lying in the bottom of the drum?

Mr. McBride.—No, sir; the water cannot lie in the bottom of the drum.

Mr. Nason.—There is steam there at the same time, though. Was there any corrosion on the bottom of the drum?

Mr. McBride.—Oh, yes, sir; on the bottom of the drum as well as the top, and all over.

Mr. Scott A. Smith.—I would like to ask if they have experimented by hanging zinc plates to prevent this action?

Mr. McBride.—No, sir, we haven't.

Mr. Smith.—It seems to me it must be due partly to chemical and partly to galvanic action, the action being between the iron and the copper or brass; and if zinc plates were hung in the boiler, then the action would be on the zinc instead of on the iron, thus saving the boiler.

You replied to my question about water, that the water contained no acid any more than was generally the case. Any acid in the water would facilitate such an action very materially. That is the way that you go to work to produce such an action.

Mr. Hawkins.—It is quite a common thing in the old-fashioned steam-boat boilers with a high vertical drum, or steam-chimney as it was called, with the steam taken from near the top, the steam having to circulate around the narrow space between that part of the shell proper forming the lower part of the chimney and the outside of the drum, so that it would come together as a current and impinge against the drum around the steam-pipe opening, to corrode the boiler just immediately around the flange where the pipe is attached. I have seen three or four cases in my experience where it was just as clearly marked as in any of the cases now shown. It seemed to be clearly attributable to the currents formed by the steam in passing out of the boiler. In this particular case of the corrosion under the line of the rivets, there seem to be ridges transversely to the groove, and as germane to the suggestion of the successive formation and removal of oxide, I think that this effect may be very easily accounted for on this same theory of the action of currents of steam, from the fact that I think if the specimen is examined we will find that the plate is rolled in a direction transverse to the line of rivets, and the natural formation of fibre in the wrought-iron plates from puddled iron, which we all know takes place as due to the mixture of scoria remaining in all such iron, that it would produce exactly that appearance even if the oxide were removed by the operation of the currents, as I have suggested, the ridges of scoria being less easily oxidized than the iron proper.

Mr. Davis.—While I suppose, as several gentlemen said, that the indications go to show that this is an acid action, and that acid action is given opportunity to take effect by the cutting action of the steam in removing the scale, I cannot help being reminded of several instances that I think would prove that that would not be likely to be the case. I know quite a number of cases where locomotive boilers of the Philadelphia & Reading road are run through certain portions of the coal regions where it is impossible to get water which is not impregnated with sulphuric acid. These boilers suffer from this water to such an extent that they can only use the engines periodically. That is, they have to change to other places where the water is purer. And in going through the shop and repairing these boilers it is found that the damage is done almost exclusively very much below the water-line—in fact, on the very bottom of the boilers. The acid seems to be a little heavier than the water and settles down, and steam-spaces have never, from my observation, been affected in the least by this. I do not believe that the acid evaporates with the water. The water evaporates and the acid is left behind. In the case of colliery boilers the water used is frequently highly acid, so that the boilers are expected to give out in a short time. The acid in this way cuts the boilers away below and at the water-line, very badly; but it has no appreciable effect upon the pipes or upper part of the boiler. We used wrought-iron steam-pipes for years to carry the steam to the mines, when the boilers used had to be renewed. That does not look much as if it were acid in Mr. McBride's case, because it does not act in the same way.

The Chairman.—I think the members would like to hear from Mr. McElroy.

Mr. Samuel McElroy.—I do not think I can add anything to the discussion. I have followed it with a certain theory in mind which does not explain the effects here stated, and this is, that in our experience with cast-iron pipes there is sometimes a very rapid tuberculation, which results from chemical action on the carbon. That is the conclusion at which we arrived after a good deal of study. It was supposed that it was due to the quality of the water, but it seems quite clear that it is the quality of the iron in tho pipes. I can hardly trace much connection between that theory and the particular phenomena described to-night. Still, there may be certain portions of those plates containing more carbon, or some other ingredient of the kind, which would lead to a more rapid corrosion, boiler-plates being especially subject to ordinary tuberculation.

Mr. Webster.—It seems to me that chemical action has something to do with the corrosion. I have a set of eight boilers, the setting of which is identical and the piping identical with the sketch on the board except that the drum is 36 inches in diameter, and the four-inch pipes, instead of connecting in at the bottom, connect in at the side of the drum. The steam is taken out at one end through a ten-inch pipe. I would say that these boilers have been running with the same pipe and the same drum for fifteen years. Previously the drum had been used on a battery of Harrison boilers for about four years. Just about the time of the change in boilers the system of the works was also changed, so that all the water from the coils of vacuum pans was run into a separate tank to be used for other purposes, and did not go back to the boilers at all.

The Chairman.—I would inquire if you have any theory to account for the fact that the top of the shell of the boiler is not affected in any way.

Mr. Webster.—I think that is due to the difference between the motion in the pipe and that in the boiler—that the currents are very much more severe in the pipe than in the boiler.

CCCLXXXI.*

REPORT OF COMMITTEE ON A STANDARD METHOD OF CONDUCTING DUTY TRIALS OF PUMPING ENGINES. (REVISED FORM.)

To the American Society of Mechanical Engineers:

The committee of five, who were appointed at the Nashville meeting of the Society, held in May, 1888, to determine upon a standard method of conducting duty trials of steam pumping engines, have endeavored to carry out the work intrusted to them, and they beg leave to report upon the same as follows:

1. The need of a uniform system of determining the performance of pumping engines is widely recognized, and it has already been commented upon at such length in the Paper and Discussion which led to the appointment of the committee, that but little requires to be added here. The main objects to be secured by the proposed standard method seem to be, first, to establish, for the benefit of members of the Society, and of others who care to use it, a mode of determining the performance of pumping engines, which may guide them in making tests themselves, and which, in contracts between builder and purchaser, may be specified as the mode to be followed in determining whether or not the guaranteed duty of the engine is realized; and, second, to furnish a common basis on which to compare the economy of different engines.

2. The committee has taken it for granted that the scope of its work extends over the whole field of duty trials, and that it is not confined simply to devising a suitable method for carrying on the operations of making the test.

It requires only a brief examination of the subject to see that much of the present variety of results which are obtained, and which it is desirable to overcome, is due to the varied nature of the coal unit upon which the duty is now based. In the eastern section of our country the unit of "100 lbs. of coal" may refer either to Cumberland bituminous coal or to anthracite coal, and in the middle or western section to Pittsburg, Illi-

* The following report, presented first at the Cincinnati Convention, May, 1890, has been revised by the committee in the light of criticism made in discussion of it, and is now published as the final views of that committee, with the discussion appended to it.

nois, or Ohio bituminous coal. In some localities the fuel may be petroleum, natural gas, coal screenings, tan-bark, or sawdust. There is a wide difference in the evaporative efficiency of these various fuels, even when of good quality, and differences of quality in the same grade of coal are the cause of still greater diversity. These variations in kind and quality of fuel, which are significant in no small degree, make the old standard unfit either for a commercial or a scientific basis for duty ratings. It is proposed, therefore, at the outset, that the existing unit, "100 lbs. of coal," be abolished, and that, in its place, a new basis, "1,000,000 heat units," be established. The new unit is the precise equivalent of 100 lbs. of coal in cases where each pound of coal imparts 10,000 heat units to the water in the boiler, or where the evaporation is $\frac{10.000}{965.7} = 10.355$ lbs. of water from and at 212° per pound of fuel. This evaporative result is readily obtained from all grades of Cumberland bituminous coal, used in horizontal return tubular boilers, and, in many cases, from the best grades of anthracite coal. The proposed new unit is thus, in reality, though not in name, in close accord with the existing unit, and, furthermore, it retains its numerical simplicity.

3. Considering that the two processes by which steam is generated and used are wholly distinct and independent of each other, there is a natural line of separation between the work of the boiler and that of the engine, and it is impossible to determine their individual economy if treated as a whole. There is reason here for separating the performance of the engine, in the proposed standard, from that of the boilers, and this mode of independent treatment is the one which the committee recommends. An additional reason for this course is found in the somewhat extended practice, among those who purchase pumping engines, of obtaining the boilers from independent builders.

In order that a contract may be brought into accord with this provision, where the complete plant is furnished by one party, it would be framed with two clauses relating to the performance, one referring to the duty of the engine proper, and the other referring to the evaporative duty of the boilers.

4. Starting with a heat-unit basis of computing duty, it is proposed to make the computation from the quantity of heat supplied to the complete plant; using not only that supplied to the

engine cylinders, but that supplied to all the accessory parts of the engine, such as the steam-jackets, the donkey feed-pump, the independent air-pump, if this be driven with steam, and any other apparatus using steam which is necessary to the operation of the engine. It is recommended that the scope of the test be made so broad that, for the sake of completeness, the quantity of steam which passes through the cylinders of the engine be determined independently of that used for other purposes, and likewise, that the quantity of steam used by each accessory part of the engine be also determined. In contract tests, if a steam-pump be used for the boiler feed-pump, the quantity of heat supplied for operating this apparatus is to be included in the total quantity, not only in cases where both boiler and engine are supplied by one party, but also where the boiler is furnished by a separate contractor. In this connection it should be added that if the engine contractor does not furnish the boiler feed-pump, he should be permitted to specify, if he desires, the kind of feeding apparatus which shall be used during the test.

The heat-unit method requires that the actual total heat of the steam shall be known, and for this purpose allowance will necessarily be made for any moisture or superheat contained by the steam furnished to the engine.

5. In determining upon a suitable method of measuring the amount of work done, which involves a measure of the quantity of water discharged into the force main, the committee have endeavored to find one which may be employed universally, and which may, in a reasonable manner, serve the ends of the builder, purchaser, and all interested parties. Plunger displacement and weir measurement have heretofore been the common means of ascertaining the quantity of water discharged. The use of a Venturi tube, so called, inserted in the force main, has been advocated, as also the employment of nozzles, or other similar means of indirect determination. [See Appendix.]

The objection to the plunger-displacement method is that, on account of "slip," which, as the committee understand, covers all the losses due to leakage of plunger and valves, to the return of water through the valves during the interval of closing, and to imperfect filling of the pump, the quantity calculated is greater than the actual discharge, and the method does not afford, in contract tests, the protection to the purchaser which his interest demands. The objection to the indirect modes of measurement, as by the use of weir, tube, or nozzles, is that the person making the test must educate himself in the manipulation of the apparatus chosen, so as to be himself assured of the reliability of the data; and, furthermore, whether thus assured or not, he must compute the desired quantities by the use of coefficients, the accuracy of which he cannot himself verify, and which may not exactly apply to the precise conditions relating to the individual case in hand. There is no method thus far used or advocated which is not open to some kind of objection, and the committee are obliged to recommend a course which, though subject to criticism, appears to reduce the objectionable features to a minimum.

In view of the fact that the number of foot pounds of work done by the pump is the vital thing to be considered, it is proposed that the plunger displacement system of measurement be employed; and, further, that the purchaser's interest be protected by the determination of the amount of slip in the pump, so far as slip is produced by leakage of the plunger, especially in pumps with inside plungers, and leakage of valves, if this occurs through faulty design. It may be said that, if it were not for leakage, the pump itself, in well-constructed engines, would undoubtedly furnish the most reliable meter which could be had, of the quantity of water discharged. A satisfactory determination of the approximate extent to which leakage occurs does not present serious difficulty.

In deciding upon this mode of measurement the committee does not for a moment underrate the importance and desirability of measurement by weir, tube, or nozzle, whenever either of these can be employed to advantage. It is strongly recommended that these additional measurements be undertaken in all cases where it is practicable to do so, that the results of the test may be supplemented by the additional data thus obtained. [See Appendix.]

6. In determining the quantity of work done by the pump, the committee recommends that the work of overcoming the friction of the water in passing through the passages and valves in the pump should not be included in the desired total; but that the work expended in friction of both the force and suction mains be included in that on which the duty is computed. It is held that the efficiency of the engine should not be made dependent upon any condition which is foreign to itself, and that the builder of the engine should be held responsible only for the work done

from the time when the water enters the pump to the time when it leaves it. The purchaser, it should be observed, should guard his interest in the matter by having the mains furnished of such capacity as to reduce their friction to a minimum.

To carry out these provisions, the data to be determined, apart from that relating to the plunger displacement, are the indication of a pressure gauge attached to the force main, that of a vacuum gauge attached to the suction main, and the vertical distance between the centres of the two gauges.

It is recommended that no air be allowed to enter the pump cylinders during the progress of the test, thereby removing all possibility of imperfect filling. If it is necessary, in special cases, for air to be "snifted in," this should be regarded as a defect in the action of the pump, which should be noted by the expert in his report, and such allowance should be made for the imperfect filling, due to the presence of air, as may be determined upon by an examination of the indicator diagrams taken from the pump cylinders, or from other data which may be secured.

7. The necessary data having been obtained in accordance with these recommendations, the duty of an engine, and other quantities relating to its performance, may be computed by the use of the following formulæ:

1. $\text{Duty} = \dfrac{\text{Foot pounds of work done}}{\text{Total number of heat units consumed}} \times 1{,}000{,}000$

$$= \frac{A(P \pm p + s) \times L \times N}{H} \times 1{,}000{,}000 \text{ (foot pounds).}$$

2. $\text{Percentage of leakage} = \dfrac{C \times 144}{A \times L \times N} \times 100 \text{ (per cent.).}$

3. Capacity = number of gallons of water discharged in 24 hours

$$= \frac{A \times L \times N \times 7.4805 \times 24}{D \times 144}$$

$$= \frac{A \times L \times N \times 1.24675}{D} \text{ (gallons).}$$

4. Percentage of total frictions

$$= \left[\frac{I.H.P. - \dfrac{A(P \pm p + s) \times L \times N}{D \times 60 \times 33{,}000}}{I.H.P.}\right] \times 100$$

$$= \left[1 - \frac{A(P \pm p + s) \times L \times N}{A_s \times M.E.P. \times L_s \times N_s}\right] \times 100 \text{ (per cent.);}$$

or, in the usual case, where the length of the stroke and number of strokes of the plunger are the same as that of the steam-piston, this last formula becomes:

$$\text{Percentage of total frictions} = \left[1 - \frac{A\,(P \pm p + s)}{A_s \times M.E.P.}\right] \times 100 \text{ (per cent.)}.$$

In these formulæ, the letters refer to the following quantities:

A = Area, in square inches, of pump plunger or piston, corrected for area of piston-rod. (When one rod is used at one end only, the correction is one-half the area of the rod. If there is more than one rod, the correction is multiplied accordingly.)

P = Pressure, in pounds per square inch, indicated by the gauge on the force main.

p = Pressure, in pounds per square inch, corresponding to indication of the vacuum gauge on suction main (or pressure gauge, if the suction pipe is under a head). The indication of the vacuum gauge, in inches of mercury, may be converted into pounds by dividing it by 2.035.

s = Pressure, in pounds per square inch, corresponding to distance between the centres of the two gauges. The computation for this pressure is made by multiplying the distance, expressed in feet, by the weight of one cubic foot of water at the temperature of the pump well, and dividing the product by 144; or by multiplying the distance in feet by the appropriate quantity, found in the following table. The quantities in this table are computed from the weights of one cubic foot of water at the various temperatures, as given by D. K. Clark in his *Rules and Tables*, which also correspond to Charles T. Porter's figures, in his work on the *Richards' Steam-Engine Indicator*.

Temperature of Water in Pump Well. Deg. Fahr.	Weight of 1 cu. ft. of Water divided by 144.	Temperature of Water in Pump Well. Deg. Fahr.	Weight of 1 cu. ft. of Water divided by 144.
32	.4335	75	.4325
35	.4335	80	.4322
40	.4335	85	.4319
45	.4334	90	.4315
50	.4333	95	.4311
55	.4332	100	.4307
60	.4331	105	.4303
65	.4329	110	.4298
70	.4327		

L = Average length of stroke of pump plunger, in feet.

N = Total number of single strokes of pump plunger made during the trial.

A_s = Area of steam-cylinder, in square inches, corrected for area of piston-rod. The quantity, $A_s \times M.E.P.$ in an engine having more than one cylinder, is the sum of the various quantities relating to the respective cylinders.

L_s = Average length of stroke of steam-piston, in feet.

N_s = Total number of single strokes of steam-piston during trial.

$M.E.P.$ = Average mean effective pressure, in pounds per square inch, measured from the indicator diagrams taken from the steam-cylinder.

$I.H.P.$ = Indicated horse-power developed by the steam-cylinder.

C = Total number of cubic feet of water which leaked by the pump plunger during the trial, estimated from the results of the leakage test.

D = Duration of trial, in hours.

H = Total number of heat units [$B.T.U.$] consumed by engine = weight of water supplied to boiler by main feed-pump × total heat of steam of boiler pressure reckoned from temperature of main feed-water + weight of water supplied by jacket-pump × total heat of steam of boiler pressure reckoned from temperature of jacket-water + weight of any other water supplied × total heat of steam reckoned from its temperature of supply. The total heat of the steam is corrected for the moisture or superheat which the steam may contain. For moisture, the correction is subtracted, and is found by multiplying the latent heat of the steam by the percentage of moisture, and dividing the product by 100. For superheat, the correction is added, and is found by multiplying the number of degrees of superheating (*i.e.*, the excess of the temperature of the steam above the normal temperature of saturated steam) by 0.48. No allowance is made for heat added to the feed-water, which is derived from any source, except the engine or some accessory of the engine. Heat added to the water by the use of a flue heater at the boiler is not to be deducted. Should heat be abstracted from the flue by means of a steam reheater connected with the intermediate receiver of the engine, this heat must be included in the total quantity supplied by the boiler.

The total and latent heats may be found by reference to the Tables of the Properties of Saturated Steam, given in Table No. 1 of the Appendix. The quantities of heat contained in one pound of water at various temperatures are appended in Table No. 2. These Tables are copied from Charles T. Porter's treatise on the *Richards' Steam-Engine Indicator*. The pressures here given are reckoned from zero. To convert the gauge pressure to that referred to as the zero basis, the barometric pressure is to be added to the corrected indications of the gauge. When the barometer indicates 29.92 inches the pressure to be added is 14.7 lbs. per square inch. For other indications of the barometer, the corresponding pressure may be found by using the multiplier 0.491.

The following examples are given to illustrate the method of computation. The figures are not obtained from tests actually made, but they correspond in round numbers with those which were so obtained :

First Example.—Compound tandem direct-acting duplex engine. Both high-pressure and low-pressure cylinders jacketed with live steam. Jet condenser used, with air-pump driven by main engine. Boiler feed-pump also driven by main engine. Jacket water returned to boiler by gravity. Main supply of feed-water drawn from hot well.

DIMENSIONS.

Diameter of each high-pressure cylinder (two).......	15	ins.
Diameter of each low-pressure cylinder (two)........	30	"
Diameter of piston-rod, each cylinder (one at each end high-pressure, two at one end low-pressure)...	3.5	"
Diameter of pump plungers (two)..................	15	"
Diameter of piston-rod, each plunger (one at one end)	3.5	"
Nominal stroke..................................	18	"

GENERAL DATA.

1. Duration of test (D)......................	12 hrs.	
2. Boiler pressure by gauge (barometric pressure 14.7 lbs.)................................	120 lbs.	
3. Temperature of water in pump well..........	80°	
4. Temperature of main supply of feed-water....	100°	
5. Temperature of jacket water........	280°	
6. Percentage of moisture in steam.............	0%	
7. Weight of water supplied to boiler by main feed-pump................................	22,400	lbs.
8. Weight of water supplied to boiler by jackets.	2,560	"

DATA RELATING TO WORK OF PUMP.

9. Area of plunger minus ½ area of rod (A).	171.9	sq. ins.
10. Average length of stroke (L and L_s)...	1.572	ft.
11. Total number of single strokes during trial (N and N_s)....................	76,000	
12. Pressure by gauge on force main (P)...	100	lbs.
13. Vacuum by gauge on suction main....	9.3	ins.
14. Pressure corresponding to vacuum given in preceding line (p)	4.57	lbs.
15. Vertical distance between gauges......	8	ft.
16. Pressure corresponding to distance given in preceding line (s)...........	3.46	lbs.
17. Volume of water which leaked through the plungers computed from results of leakage test (C).................	5,900	cu. ft.

DATA RELATING TO WORK OF STEAM-CYLINDERS.

18. Area of high-pressure piston minus area of one rod (A_{s_1}).......	167.09	sq. ins.

19. Mean effective pressure high-pressure cylinder ($M.E.P._1$)	61.31	lbs.
20. Area of low-pressure piston minus ½ area of two rods (A_{s_2})	697.24	sq. ins.
21. Mean effective pressure low-pressure cylinder ($M.E.P._2$)	13.72	lbs.
22. Number of double strokes each side per minute	26.39	
23. Indicated horse-power developed by steam-cylinders	99.61	I.H.P.
24. Feed-water consumed per indicated horse-power per hour	20.88	lbs.
25. Heat units consumed per indicated horse-power per hour	23,003	B.T.U.

= 383 B.T.U. per minute.

TOTAL HEAT OF STEAM RECKONED FROM THE VARIOUS TEMPERATURES OF FEED-WATER AND COMPUTATIONS BASED THEREON.

26. Total heat of 1 lb. of dry steam at 120 lbs. gauge pressure reckoned from 0° Fahr.	1,220.6	B.T.U.
27. Ditto, reckoned from temperature of main feed-water (100°)	1,120.5	"
28. Ditto, reckoned from temperature of jacket-water (280°)	938.5	"
29. Heat consumed by engine (H) (22,400 × 1120.5) + (2560 × 938.5)	27,501,760	"

RESULTS.

Substituting these quantities in the formulæ, we have :

$$1.\ \text{Duty} = \frac{\overset{A}{171.9} \times (\overset{P}{100} + \overset{p}{4.57} + \overset{s}{3.46}) \times \overset{L}{1.572} \times \overset{N}{76,000}}{\underset{27,501,760}{H}} \times 1,000,000$$

$$= 80,671,622 \text{ foot pounds.}$$

$$2.\ \text{Percentage of leakage} = \frac{\overset{C}{5,900 \times 144}}{\overset{A}{171.9} \times \overset{L}{1.572} \times \overset{N}{76,000}} \times 100 = 4.1\%.$$

$$3.\ \text{Capacity} = \frac{\overset{A}{171.9} \times \overset{L}{1.572} \times \overset{N}{76,000} \times 1.24675}{\underset{12}{D}}$$

$$= 2,133,735 \text{ gallons.}$$

4. Percentage of total frictions

$$= \left[1 - \frac{\overset{A}{171.9} \times (\overset{P}{100} + \overset{p}{4.57} + \overset{s}{3.46})}{(\overset{A_{s_1}}{167.09} \times \overset{M.E.P._1}{61.31}) + (\overset{A_{s_2}}{697.24} \times \overset{M.E.P._2}{13.72})}\right] \times 100$$

$$= 9.4\%.$$

Second Example.—Compound fly-wheel engine. High-pressure cylinder jacketed with live steam from the boiler. Low-pressure cylinder jacketed with steam from the intermediate receiver, the condensed water from which is returned to the boiler by means of a pump operated by the engine. Main steam-pipe fitted with a separator. The intermediate receiver provided with a reheater supplied with boiler steam. Water drained from high-pressure jacket, separator, and reheater, collected in a closed tank under boiler pressure, and from this point fed to the boiler direct by an independent steam-pump. Jet condenser used operated by an independent air-pump. Main supply of feed-water drawn from hot well and fed to the boiler by donkey steam-pump, which discharges through a feed-water heater. All the steam-pumps, together with the independent air-pump, exhaust through the heater to the atmosphere.

DIMENSIONS.

Diameter of high-pressure steam-cylinder (one)........	20 ins.
Diameter of low-pressure steam-cylinder (one).........	40 "
Diameter of plunger (one)...........................	20 "
Diameter of each piston-rod	4 "
Stroke of steam-pistons and pump plunger............	3 ft.

GENERAL DATA.

1. Duration of trial (D).....................	10	hrs.
2. Boiler pressure indicated by gauge (barometric pressure, 14.7 lbs.)...............	120	lbs.
3. Temperature of water in pump well	60°	
4. Temperature of water supplied to boiler by main feed-pump, leaving heater	215°	
5. Temperature of water supplied by low-pressure jacket pump....	225°	
6. Temperature of water supplied by high-pressure jacket, separator, and reheater pump, that derived from separator being 340°, and that from jackets 290°.........	300°	
7. Weight of water supplied to boiler by main feed-pump	18,863	lbs.
8. Weight of water supplied by low-pressure jacket pump..........................	615	"
9. Weight of water supplied by pump for high-pressure jacket, separator, and reheater tank, of which 210 lbs. is derived from separator.........................	1,025	"
10. Total weight of feed-water supplied from all sources	20,503	"
11. Percentage of moisture in steam after leaving separator.........................	1.5%	

DATA RELATING TO WORK OF PUMP.

12. Area of plunger minus ½ area of piston-rod (A)	307.88	sq. ins.
13. Average length of stroke (L and L_1)...	3	ft.

14. Total number of single strokes during trial (N and N_s)	24,000	
15. Pressure by gauge on force main (P)	95	lbs.
16. Vacuum by gauge on suction main	7.5	ins.
17. Pressure corresponding to vacuum given in preceding line (p)	3.69	lbs.
18. Vertical distance between centres of two gauges	10	ft.
19. Pressure equivalent to distance between two gauges (s)	4.33	lbs.
20. Total leakage of pump during trial, determined from results of leakage test (C)	3,078	cu. ft.
21. Number of double strokes of pump per minute	20	
22. Mean effective pressure measured from pump diagrams	105	lbs.
23. Indicated horse-power exerted in pump cylinders	117.55	I.H.P.

DATA RELATING TO WORK OF STEAM-CYLINDERS.

24. Area of high-pressure piston minus ½ area of rod (A_{s1})	307.88	sq. ins.
25. Area of low-pressure piston minus ½ area of rod (A_{s2})	1,250.36	"
26. Average length of stroke, each	3	ft.
27. Mean effective pressure measured from high-pressure diagrams ($M.E.P._1$)	59.25	lbs.
28. Mean effective pressure measured from low-pressure diagrams ($M.E.P._2$)	13.60	"
29. Number of double strokes per minute (line 21)	20	
30. Indicated horse-power developed by high-pressure cylinder	66.33	I.H.P.
31. Indicated horse power developed by low-pressure cylinder	61.82	"
32. Indicated horse-power developed by both cylinders	128.15	"
33. Feed-water consumed by plant per indicated horse-power per hour, corrected for separator water and for moisture in steam	15.60	lbs.
34. Number of heat units consumed per indicated horse-power per hour	15,652.1	B.T.U.
35. Number of heat units consumed per indicated horse-power per minute	260.9	"

TOTAL HEAT OF STEAM RECKONED FROM THE VARIOUS TEMPERATURES OF FEED-WATER AND COMPUTATIONS BASED THEREON.

36. Total heat of 1 lb. of steam at 120 lbs. gauge pressure, containing 1.5% of

moisture, reckoned from 0° Fahr. = 1,220.6 − (1.5% of 866.7)	1,207.6 B.T.U.
37. Ditto, reckoned from 215°, temperature of main feed-water = 1,207.6 − 215.9	991.7 "
38. Ditto, reckoned from 225°, temperature of low-pressure jacket water = 1,207.6 − 226.1	981.5 "
39. Ditto, reckoned from 290°, temperature of high-pressure jacket and reheater water = 1,207.6 − 292.3 ...	915.3 "
40. Heat of separator water reckoned from 340° = 353.9 − 343.8..........	10.1 "
41. Heat consumed by engine (H) = (18.863 × 991.7) + (615 × 981 5) + (315 × 915.3) + (210 × 10.1)..........	20,058,150 "

RESULTS.

Substituting these quantities in the formulæ, we have:

$$1.\ \text{Duty} = \frac{\overset{A}{307.88} \times (\overset{P}{95} + \overset{p}{3.69} + \overset{s}{4.33}) \times \overset{L}{3} \times \overset{N}{24,000}}{\underset{20,058,150}{H}} \times 1,000,000$$

$$= 118,853,044 \text{ foot pounds.}$$

$$2.\ \text{Percentage of leakage} = \frac{\overset{C}{3,078 \times 144}}{\underset{307.88 \times 3 \times 24,000}{A \quad L \quad N}} \times 100 = 2.0\%$$

$$3.\ \text{Capacity} = \frac{\overset{A}{307.88} \times \overset{L}{3} \times \overset{N}{24,000} \times 1.24675}{\underset{10}{D}}$$

$$= 2,763,716 \text{ gallons.}$$

4. Percentage of total frictions

$$= \left[1 - \frac{\overset{A}{307.88} \times (\overset{P}{95} + \overset{p}{3.69} + \overset{s}{4.33})}{(\overset{A_{s1}}{307.88} \times \overset{M.E.P._1}{59.25}) + (\overset{A_{s2}}{1250.36} \times \overset{M.E.P._2}{18.6})}\right] \times 100$$

$= 9.0\%$.

8. The method to be followed in conducting the boiler test which is recommended is the standard mode determined upon by the Society's Committee on Boiler Trials.* It is suggested that the results of the boiler test be expressed, not only in terms of the number of pounds of water evaporated per pound of coal, in

* Vol. VI., p. 267, *Transactions A. S. M. E.*, 1885.

the customary manner, but also in terms of the number of pounds of coal required to generate 1,000,000 heat units, so that a simple calculation may determine the duty of the engine, if desired, on the basis of 100 lbs. of the coal actually used.

9. In order that the contract between builder and purchaser of a pumping engine may conform to the proposed standard, the guarantee as to performance should be expressed in the following terms:

> 1. The engine shall perform a duty, based upon plunger displacement, equivalent to not less than....foot pounds of work for each one million heat units consumed.
>
> 2. The leakage of the pump shall not exceed....per cent. of the total plunger displacement, when the engine is working at its rated capacity.
>
> 3. The boiler shall supply one million heat units to the engine on a consumption of....pounds of....coal, or it shall evaporate not less than....pounds of water from and at 212 degrees per pound of the combustible portion of the coal named.
>
> 4. The mode of determining these quantities is to conform to the standard method of conducting duty trials recommended by the Committee of the American Society of Mechanical Engineers.

Should one contractor furnish the engine, and another the boiler, separate guarantees will be made, the individual requirements of which are the same as those noted.

It is desirable, where both parties concur therein, to introduce into the contract the following additional provision regarding friction, viz.:

> "The friction of the engine shall not exceed....per cent. of the indicated power developed in the steam-cylinders."

10. Having thus far noted the main principles which have been followed, and having pointed out the various steps by which the results of a test are computed, the committee now beg to submit, in the following pages, the full particulars regarding the standard method of conducting duty trials which they recommend.

The general mode of operation is to first subject the plant to a preliminary run under the working conditions, for the purpose of determining the temperature of the feed-water, or the several temperatures where there is more than one supply. It is usually impracticable to weigh the main supply of water, derived, as it generally is, from a low-placed hot well, and the test of the main quantity of feed-water used must, as a rule, be made with cold water drawn from the service main. The changed conditions in the working of the plant thus introduced, and the arrangement of apparatus which is frequently needed to measure the additional supplies of feed-water, make it desirable to obtain the working temperatures as a preliminary to the main duty trial. Hence the preliminary run is made, as noted, merely for securing the temperatures. The main test of the boiler and engine is then carried forward, and during this test the weights of the various supplies of feed-water are determined, and the remaining data needed for making the computations. Finally, as soon as practicable after these tests are completed, the rate of leakage through the pump is measured, with the engine at rest.

As to the duration of the test, it appears to the committee that, so far as the main trial is concerned, which is practically a feed-water test, it need not be prolonged more than ten hours, unless, in that time, appreciable errors should be produced by inaccuracies in the observations of the height of water in the gauge glass. The duration of the boiler trial might, with good reason, be made longer, were it not that the results of the boiler test are independent of those of the duty trial. It is desirable to reduce, if possible, the number of hours of the trial to such a point that the time expended upon the work, including that required in preparation for the beginning of the test, and that spent in bringing the test to a close, shall be such that the same expert, without undue physical exertion, may have the test under his continuous supervision from beginning to end. This is feasible where the length of the duty trial, according to the plan proposed, does not exceed ten hours.

Shortly after the committee were appointed several pumping-engine manufacturers were asked to submit their ideas as to the best form of standard to use. The committee take pleasure in acknowledging the receipt of suggestions from the Davidson Steam Pump Company, the Deane Steam Pump Company, the Holly Manufacturing Company, and the George F. Blake Manufacturing Company, the three last named having given full expression to their views upon the subject.

Respectfully submitted,

GEO. H. BARRUS,
A. F. NAGLE,
EDWIN REYNOLDS,
J. J. DE KINDER,
J. S. COON,
Committee.

STANDARD METHOD OF CONDUCTING DUTY TRIALS.

1. TEST OF FEED-WATER TEMPERATURES.

The plant is subjected to a preliminary run, under the conditions determined upon for the test, for a period of three hours, or such a time as is necessary to find the temperature of the feed-water (or the several temperatures, if there is more than one supply) for use in the calculation of the duty. During this test observations of the temperature are made every fifteen minutes. Frequent observations are also made of the speed, length of stroke, indication of water-pressure gauges, and other instruments, so as to have a record of the general conditions under which this test is made.

DIRECTIONS FOR OBTAINING FEED-WATER TEMPERATURES.

When the feed-water is all supplied by one feeding instrument, the temperature to be found is that of the water in the feed-pipe near the point where it enters the boiler. If the water is fed by an injector this temperature is to be corrected for the heat added to the water by the injector, and for this purpose the temperatures of the water entering and of that leaving the injector are both observed. If the water does not pass through a heater on its way to the boiler (that is, that form of heater which depends upon the rejected heat of the engine, such as that contained in the exhaust steam either of the main cylinders or of the auxiliary pumps) it is sufficient, for practical purposes, to take the temperature of the water at the source of supply, whether the feeding instrument is a pump or an injector.

When there are two independent sources of feed-water supply, one the main supply from the hot well, or from some other source, and the other an auxiliary supply derived from the water condensed in the jackets of the main engine and in the live-steam reheater, if one be used, they are to be treated independently. The remarks already made apply to the first, or main, supply. The temperature of the auxiliary supply, if carried by an independent pipe either direct to the boiler or to the main feed-pipe near the boiler, is to be taken at a convenient point in the independent pipe.

When a separator is used in the main steam-pipe, arranged so as to discharge the entrained water back into the boiler by gravity, no account need be made of the temperature of the water thus returned. Should it discharge either into the atmosphere to waste, to the hot well, or to the jacket tank, its temperature is to be determined at the point where the water leaves the separator before its pressure is reduced.

When a separator is used, and it drains by gravity into the jacket tank, this tank being subjected to boiler pressure, the temperatures of the separator water and jacket water are each to be taken before their entrance to the tank.

Should there be any other independent supply of water, the temperature of that is also to be taken on this preliminary test.

DIRECTIONS FOR MEASUREMENT OF FEED-WATER.

As soon as the feed-water temperatures have been obtained the engine is stopped, and the necessary apparatus arranged for determining the weight of the feed-water consumed, or of the various supplies of feed-water, if there is more than one.

In order that the main supply of feed-water may be measured, it will generally be found desirable to draw it from the cold-water service main. The best form of apparatus for weighing the water consists of two tanks, one of which rests upon a platform scale supported by staging, while the other is placed underneath. The water is drawn from the service main into the upper tank, where it is weighed, and it is then emptied into the lower tank. The lower tank serves as a reservoir, and to this the suction-pipe of the feeding apparatus is connected.

The jacket water may be measured by using a pair of small barrels, one being filled while the other is being weighed and emptied. This water, after being measured, may be thrown away, the loss being made up by the main feed-pump. To prevent evaporation from the water, and consequent loss on account of its highly heated condition, each barrel should be partially filled with cold water previous to using it for collecting the jacket water, and the weight of this water treated as tare.

When the jacket water drains back by gravity to the boiler, waste of live steam during the weighing should be prevented by providing a small vertical chamber, and conducting the water into this receptacle before its escape. A glass water-gauge is attached so as to show the height of water inside the chamber, and this serves as a guide in regulating the discharge valve. The chamber may be made of piping in the manner shown in the appended figure (108).

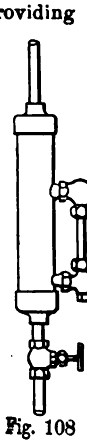


Fig. 108

When the jacket water is returned to the boiler by means of a pump, the discharge valve should be throttled during the test, so that the pump may work against its usual pressure—that is, the boiler pressure as nearly as may be, a gauge being attached to the discharge pipe for this purpose.

When a separator is used and the entrained water discharges either to waste, to the hot well, or to the jacket tank, the weight of this water is to be determined, the water being drawn into barrels in the manner pointed out for measuring the jacket water. Except in the case where the separator discharges into the jacket tank, the entrained water thus found is treated, in the calculations, in the same manner as moisture shown by the calorimeter test. When it discharges into the jacket tank, its weight is simply subtracted from the total weight of water fed, and allowance made for heat of this water lost by radiation between separator and tank.

When the jackets are drained by a trap, and the condensed water goes either to waste or to the hot well, the determination of the quantity used is not necessary to the main object of the duty trial, because the main feed-pump in such cases supplies all the feed-water. For the sake of having complete data, however, it is desirable that this water be measured, whatever the use to which it is applied.

Should live steam be used for reheating the steam in the intermediate receiver, it is desirable to separate this from the jacket steam, if it drains into the same tank, and measure it independently. This, likewise, is not essential to the main object of the duty trial, though useful for purposes of information.

The remarks as to the manner of preventing losses of live steam and of evaporation, in the measurement of jacket water, apply to the measurement of any other hot water under pressure which may be used for feed-water.

Should there be any other independent supply of water to the boiler besides those named, its quantity is to be determined independently, apparatus for all these measurements being set up during the interval between the preliminary run and the main trial, when the plant is idle.

2. THE MAIN DUTY TRIAL.

The duty trial is here assumed to apply to a complete plant, embracing a test of the performance of the boiler, as well as that of the engine. The test of the two will go on simultaneously after both are started, but the boiler test will begin a short time in advance of the commencement of the engine test, and continue a short time after the engine test is finished. The mode of procedure is as follows:

The plant having been worked for a suitable time under normal conditions, the fire is burned down to a low point and the engine brought to rest. The fire remaining on the grate is then quickly hauled, the furnace cleaned, and the refuse withdrawn from the ash pit. The boiler test is now started, and this test is made in accordance with the rules for a standard method recommended by the Committee on Boiler Tests of the American Society of Mechanical Engineers.* This method, briefly described, consists in starting the test with a new fire lighted with wood, the boiler having previously been heated to its normal working degree; operating the boiler in accordance with the conditions determined upon; weighing coal, ashes, and feed-water; observing the draught, temperatures of feed-water and escaping gases, and such other data as may be incidentally desired; determining the quantity of moisture in the coal and in the steam; and at the close of

* Vol. VI., p. 267, *Transactions A. S. M. E.*, 1885.

the test hauling the fire, and deducting from the weight of coal fired whatever unburned coal is contained in the refuse withdrawn from the furnace, the quantity of water in the boiler and the steam pressure being the same as at the time of lighting the fire at the beginning of the test.

Previous to the close of the test it is desirable that the fire should be burned down to a low point, so that the unburned coal withdrawn may be in a nearly consumed state. The temperature of the feed-water is observed at the point where the water leaves the engine heater, if this be used, or at the point where it enters the flue heater, if that apparatus be employed. Where an injector is used for supplying the water, a deduction is to be made in either case for the increased temperature of the water derived from the steam which it consumes.

As soon after the beginning of the boiler test as practicable the engine is started and preparations are made for the beginning of the engine test. The formal commencement of this test is delayed till the plant is again in normal working condition, which should not be over one hour after the time of lighting the fire. When the time for commencement arrives the feed-water is momentarily shut off, and the water in the lower tank is brought to a mark. Observations are then made of the number of tanks of water thus far supplied, the height of water in the gauge-glass of the boiler, the indication of the counter on the engine, and the time of day; after which the supply of feed-water is renewed, and the regular observations of the test, including the measurement of the auxiliary supplies of feed-water, are commenced. The engine test is to continue at least ten hours. At its expiration the feed-pump is again momentarily stopped, care having been taken to have the water slightly higher than at the start, and the water in the lower tank is brought to the mark. When the water in the gauge-glass has settled to the point which it occupied at the beginning, the time of day and the indication of the counter are observed, together with the number of tanks of water thus far supplied, and the engine test is held to be finished. The engine continues to run after this time till the fire reaches a condition for hauling, and completing the boiler test. It is then stopped, and the final observations relating to the boiler test are taken.

The observations to be made and data obtained for the purposes of the engine test, or duty trial proper, embrace the weight of feed-water supplied by the main feeding apparatus, that of the water drained from the jackets, and any other water which is ordinarily supplied to the boiler, determined in the manner pointed out. They also embrace the number of hours duration, and number of single strokes of the pump during the test; and, in direct-acting engines, the length of the stroke; together with the indications of the gauges attached to the force and suction mains, and indicator diagrams from the steam-cylinders. It is desirable that pump diagrams also be obtained.

Observations of the length of stroke, in the case of direct-acting engines, should be made every five minutes; observations of the water-pressure gauges every fifteen minutes; observations of the remaining instruments, such as steam-gauge, vacuum-gauge, thermometer in pump well, thermometer in feed-pipe, thermometer showing temperature of engine room, boiler room, and outside air, thermometer in flue, thermometer in steam-pipe, if the boiler has steam-heating surface, barometer, and other instruments which may be used, every half-hour. Indicator diagrams should be taken every half-hour.

When the duty trial embraces simply a test of the engine, apart from the boiler, the course of procedure will be the same as that described, excepting that the fires will not be hauled, and the special observations relating to the performance of the boiler will not be taken.

DIRECTIONS REGARDING ARRANGEMENT AND USE OF INSTRUMENTS, AND OTHER PROVISIONS FOR THE TEST.

The gauge attached to the force main is liable to a considerable amount of fluctuation unless the gauge-cock is nearly closed. The practice of choking the cock is objectionable. The difficulty may be satisfactorily overcome, and a nearly steady indication secured, with cock wide open, if a small reservoir having an air-chamber is interposed between the gauge and the force main, in the manner shown in the appended figure (109). By means of a gauge-glass on the side of the chamber and an air-valve, the average water-level may be adjusted to the height of the centre of the gauge, and correction for this element of variation is avoided. If not thus adjusted, the reading is to be referred to the level shown, whatever this may be.

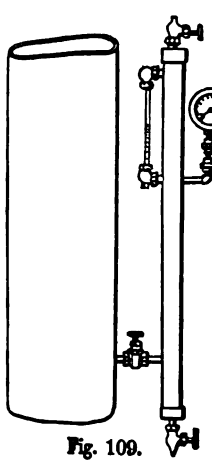


Fig. 109.

To determine the length of stroke in the case of direct-acting engines, a scale should be securely fastened to the frame which connects the steam and water-cylinders, in a position parallel to the piston-rod, and a pointer attached to the rod so as to move back and forth over the graduations on the scale. The marks on the scale, which the pointer reaches at the two ends of the stroke, are thus readily observed, and the distance moved over computed. If the length of the stroke can be determined by the use of some form of registering apparatus, such a method of measurement is preferred. The personal errors in observing the exact scale marks, which are liable to creep in, may thereby be avoided.

The form of calorimeter to be used for testing the quality of the steam is left to the decision of the person who conducts the trial. It is preferred that some form of continuous calorimeter be used, which acts directly on the moisture tested. If either the superheating calorimeter* or the wire-drawing † instrument be employed, the steam which

* Vol. VII., p. 178, 1886.

† Vol. XI., p. 791, 1890, *Transactions A. S. M. E.* (Paper on "A Universal Calorimeter," May, 1890.)

it discharges is to be measured either by numerous short trials, made by condensing it in a barrel of water previously weighed, thereby obtaining the rate by which it is discharged, or by passing it through a surface condenser of some simple construction, and measuring the whole quantity consumed. When neither of these instruments is at hand, and dependence must be placed upon the barrel calorimeter, scales should be used which are sensitive to a change in weight of a small fraction of a pound, and thermometers which may be read to tenths of a degree. The pipe which supplies the calorimeter should be thoroughly warmed and drained just previous to each test. In making the calculations the specific heat of the material of the barrel or tank should be taken into account, whether this be of metal or of wood.

If the steam is superheated, or if the boiler is provided with steam-heating surface, the temperature of the steam is to be taken by means of a high-grade thermometer resting in a cup holding oil or mercury, which is screwed into the steam-pipe so as to be surrounded by the current of steam. The temperature of the feed-water is preferably taken by means of a cup screwed into the feed-pipe in the same manner.

Indicator pipes and connections used for the water-cylinders should be of ample size, and, so far as possible, free from bends. Three-quarter-inch pipes are preferred, and the indicators should be attached one at each end of the cylinder. It should be remembered that indicator springs which are correct under steam heat are erroneous when used for cold water. When such springs are used, the actual scale should be determined, if calculations are made of the indicated work done in the water-cylinders. The scale of steam-springs should be determined by a comparison, under steam pressure, with an accurate steam-gauge at the time of the trial, and that of water-springs by cold dead-weight test.

The accuracy of all the gauges should be carefully verified by comparison with a reliable mercury column. Similar verification should be made of the thermometers, and if no standard is at hand, they should be tested in boiling water and melting ice.

To avoid errors in conducting the test, due to leakage of stop-valves either on the steam-pipes, feed-water pipes, or blow-off pipes, all these pipes not concerned in the operation of the plant under test should be disconnected.

3. LEAKAGE TEST OF PUMP.

As soon as practicable after the completion of the main trial (or at some time immediately preceding the trial), the engine is brought to rest, and the rate determined at which leakage takes place through the plunger and valves of the pump, when these are subjected to the full pressure of the force main.

The leakage of an inside plunger [the only type which requires testing] is most satisfactorily determined by making the test with the cylinder head removed. A wide board or plank may be temporarily bolted to the lower part of the end of the cylinder, so as to hold back the water in the manner of a dam, and an opening made in the temporary head thus provided for the reception of an overflow pipe. The plunger is blocked at some intermediate point in the stroke (or, if this position is not practicable, at the end of the stroke), and the water from the force main is admitted at full pressure behind it. The leakage escapes through the overflow pipe, and it is collected in barrels and measured.

Should the escape of the water into the engine room be objectionable, a spout may be constructed to carry it out of the building. Where the leakage is too great to be readily measured in barrels, or where other objections arise, resort may be had to weir or orifice measurement, the weir or orifice taking the place of the overflow pipe in the wooden head. The apparatus may be constructed, if desired, in a somewhat rude manner, and yet be sufficiently accurate for practical requirements. The test should be made, if possible, with the plunger in various positions.

In the case of a pump so planned that it is difficult to remove the cylinder head, it may be desirable to take the leakage from one of the openings which are provided for the inspection of the suction valves, the head being allowed to remain in place.

It is here assumed that there is a practical absence of valve leakage, a condition of things which ought to be attained in all well-constructed pumps. Examination for such leakage should be made first of all, and if it occurs and it is found to be due to disordered valves, it should be remedied before making the plunger test. Leakage of the discharge valves will be shown by water passing down into the empty cylinder at either end when they are under pressure. Leakage of the suction valves will be shown by the disappearance of water which covers them.

If valve leakage is found which cannot be remedied, the quantity of water thus lost should also be tested. The determination of the quantity which leaks through the suction valves, where there is no gate in the suction pipe, must be made by indirect means. One method is to measure the amount of water required to maintain a certain pressure in the pump cylinder when this is introduced through a pipe temporarily erected, no water being allowed to enter through the discharge valves of the pump.

The exact methods to be followed in any particular case, in determining leakage, must be left to the judgment and ingenuity of the person conducting the test.

4. TABLE OF DATA AND RESULTS.

In order that uniformity may be secured, it is suggested that the data and results, worked out in accordance with the standard method, be tabulated in the manner indicated in the following scheme :

DUTY TRIAL OF ENGINE.

DIMENSIONS.

1. Number of steam-cylinders ....
2. Diameter of steam-cylinders.............................. ins.
3. Diameter of piston-rods of steam-cylinders................ ins.

4. Nominal stroke of steam-pistons........................ ft.
5. Number of water-plungers..............................
6. Diameter of plungers.................................. ins.
7. Diameter of piston-rods of water-cylinders................ ins.
8. Nominal stroke of plungers............................ ft.
9. Net area of plungers.................................. sq. ins.
10. Net area of steam-pistons............................ sq. ins.
11. Average length of stroke of steam-pistons during trial..... ft.
12. Average length of stroke of plungers during trial......... ft.

(Give also complete description of plant.)

TEMPERATURES.

13. Temperature of water in pump well... .... degs
14. Temperature of water supplied to boiler by main feed-pump. degs.
15. Temperature of water supplied to boiler from various other sources.. degs.

FEED-WATER.

16. Weight of water supplied to boiler by main feed-pump.... lbs.
17. Weight of water supplied to boiler from various other sources.. lbs.
18. Total weight of feed-water supplied from all sources...... lbs.

PRESSURES.

19. Boiler pressure indicated by gauge...................... lbs.
20. Pressure indicated by gauge on force main................ lbs.
21. Vacuum indicated by gauge on suction main.............. ins.
22. Pressure corresponding to vacuum given in preceding line . lbs.
23. Vertical distance between the centres of the two gauges... ins.
24. Pressure equivalent to distance between the two gauges... lbs.

MISCELLANEOUS DATA.

25. Duration of trial.. hrs.
26. Total number of single strokes during trial................
27. Percentage of moisture in steam supplied to engine, or number of degrees of superheating........................ % or deg.
28. Total leakage of pump during trial, determined from results of leakage test.................................. lbs.
29. Mean effective pressure, measured from diagrams taken from steam-cylinders.................................. M. E. P.

PRINCIPAL RESULTS.

30. Duty .... ft. lbs.
31. Percentage of leakage.................................. %
32. Capacity .... gals.
33. Percentage of total frictions............................ %

ADDITIONAL RESULTS.*

34. Number of double strokes of steam-piston per minute .....
35. Indicated horse-power developed by the various steam-cylinders.. I. H. P.

* These are not necessary to the main object, but it is desirable to give them.

36. Feed-water consumed by the plant per hour.............. lbs.
37. Feed-water consumed by the plant per indicated horse-power per hour, corrected for moisture in steam......... lbs.
38. Number of heat units consumed per indicated horse-power per hour.. B. T. U.
39. Number of heat units consumed per indicated horse-power per minute.. B. T. U.
40. Steam accounted for by indicator at cut-off and release in the various steam-cylinders............................ lbs.
41. Proportion which steam accounted for by indicator bears to the feed-water consumption..........................

SAMPLE DIAGRAMS TAKEN FROM STEAM-CYLINDERS.

[Also, if possible, full measurements of the diagrams, embracing pressures at the initial point, cut-off, release, and compression; also back-pressure, and the proportions of the stroke completed at the various points noted.]

42. Number of double strokes of pump per minute............
43. Mean effective pressure, measured from pump diagrams... M. E. P.
44. Indicated horse-power exerted in pump cylinders.......... I. H. P.
45. Work done (or duty) per 100 lbs. of coal................... ft. lbs.

SAMPLE DIAGRAMS TAKEN FROM PUMP CYLINDERS.

DATA AND RESULTS OF BOILER TEST.

[IN ACCORDANCE WITH THE SCHEME RECOMMENDED BY THE BOILER TEST COMMITTEE OF THE SOCIETY.]

1. Date of trial..
2. Duration of trial...................................... hrs.

DIMENSIONS AND PROPORTIONS.

3. Grate surface wide long Area............ sq. ft.
4. Water-heating surface................................ sq. ft.
5. Superheating surface.................................. sq. ft.
6. Ratio of water-heating surface to grate surface............
(Give also complete description of boilers.)

AVERAGE PRESSURES.

7. Steam pressure in boiler by gauge....................... lbs.
8. Atmospheric pressure by barometer...................... lbs.
9. Force of draught in inches of water...................... ins.

AVERAGE TEMPERATURES.

10. Of steam... degs.
11. Of escaping gases..................................... degs.
12. Of feed-water... degs.

FUEL.

13. Total amount of coal consumed * .... lbs.
14. Moisture in coal .... %
15. Dry coal consumed.. lbs.
16. Total refuse (dry)... lbs.
17. Total combustible (dry weight of coal, item 15, less refuse, item 16).. lbs.
18. Dry coal consumed per hour................................. lbs.
19. Total heat of combustion of the coal obtained by calorimetric trial.. B. T. U.

RESULTS OF CALORIMETRIC TEST OF STEAM.

20. Percentage of moisture in steam .... %
21. Number of degrees superheated .... degs.

WATER.

22. Total weight of water pumped into boiler and apparently evaporated †.. lbs.
23. Water actually evaporated corrected for quality of steam.. lbs.
24. Equivalent water evaporated into dry steam from and at 212° Fahr.‡.. lbs.
25. Equivalent total heat derived from fuel, in British Thermal Units .... B. T. U.
26. Equivalent water evaporated into dry steam from and at 212° Fahr. per hour.................................. lbs.

ECONOMIC EVAPORATION.

27. Water actually evaporated per pound of dry coal from actual pressure and temperature .... lbs.
28. Equivalent water evaporated per pound of dry coal from and at 212° Fahr...................................... lbs.
29. Equivalent water evaporated per pound of combustible from and at 212° Fahr...................................... lbs.
30. Number of pounds of coal required to supply 1,000,000 British Thermal Units.................................... lbs.

RATE OF COMBUSTION.

31. Dry coal actually burned per square foot of grate surface per hour.. lbs.

---

* Including equivalent of wood used in lighting fire. One pound of wood equals 0.4 of a pound of coal, not including unburned coal withdrawn from fire at end of test.

† Corrected for inequality of water-level and of steam pressure at beginning and end of test.

‡ Factor of evaporation $= \frac{H-h}{965.7}$, H and h being respectively the total heat units in steam of the average observed pressure, and in water of the average observed temperature of feed.

RATE OF EVAPORATION.

82. Water evaporated from and at 212° Fahr. per square foot of heating surface per hour .... lbs.

[NOTE.—To determine the percentage of surface moisture in the coal a s of the coal should be dried for a period of twenty-four hours, being subjec a temperature of not more than 212°. The quantity of unconsumed coal tained in the refuse withdrawn from the furnace and ash pit at the end c test may be found by sifting either the whole of the refuse, or a sample c same, in a screen having $\frac{3}{8}$-inch meshes. This, deducted from the weight c coal fired, gives the weight of dry coal consumed, for line 15.—*Duty Trial mittee.*]

APPENDIX.

MEASUREMENT OF WATER BY MEANS OF WEIRS, TUBES, AND NOZZLES.

THE following memoranda in regard to the measurement of water by means of weirs, tubes, and nozzles are appended to the report, in order that these systems may be readily availed of whenever it is practicable to do so. It is not attempted to give full directions here, but simply the general principles which should be followed in order to obtain reliable work. The reader is referred to the accounts of various investigators themselves, who have experimented in these lines, for the detailed instructions which cannot here be introduced:

WEIRS.

The measurement of water by the use of weirs is generally based, at the present time, on the results of experiments made by Mr. James B. Francis, C.E., at Lowell, in 1852, an account of which is given in *Lowell Hydraulic Experiments*, D. Van Nostrand. These experiments led to the construction of the following formula, which is known as the "Francis Formula," viz.:

$$Q = 3.33\,(L - 0.2H) \times H^{\frac{3}{2}},$$

in which Q is the discharge of water in cubic feet per second, L the length of the weir, and H the depth on the weir, all of these measurements being in terms of the English foot. The coefficient, 3.33, was obtained from the mean of eighty-eight experiments, the greatest variation from the mean in any individual case being one per cent. The length of the weir, in all but six of the experiments, was approximately ten feet. The depth of water on the weir varied from 7 to 19 inches. The formula applies to that type of weir having perfect contraction at each end, which was the form used in sixty-five experiments.

The weir was of rectangular cross-section, with a horizontal crest and vertical ends. The upper edge was made of cast iron, and the corner presented to the current was square and sharp. The horizontal part of the crest was one-fourth of an inch wide, and the remaining part was bevelled off at an angle of 45°. The ends were of similar cross-section to the crest. The depth on the weir was taken by means of hook gauges, six feet from the weir, these gauges being placed in wooden boxes situated on the sides of the canal and communicating with the water through small openings. Vertical gratings, for overcoming eddies in the current, were provided above the gauges. The distance from the side of the canal to the end of the weir was about two feet, and the depth of the canal below the crest was, in most of the experiments, about five feet.

The Francis Formula is applicable only to cases similar to the ones described. According to Mr. Francis' statement, it cannot be applied where the depth on the weir exceeds one-third of the length, nor to very small depths: furthermore, the distance from the side of the canal to the end of the weir should not be less than three times that on the weir.

In using the formula, the depth of water on the weir should be corrected for the head due to the velocity of approach according to the formula

$$H' = (H + h^{\frac{3}{2}} - h^{\frac{3}{2}})^{\frac{2}{3}},$$

in which H' is the corrected depth, H the observed depth, and h the head due to the velocity of approach. This last may be determined from the formula

$$h = \frac{V}{64.8},$$

in which V is the velocity of approach in feet per second, which may be determined by dividing the uncorrected discharge of water, in cubic feet per second, by the area of the cross-section of the stream flowing through the canal, in square feet.

The reader is also referred to the experiments on weirs made by Hamilton Smith, Jr., described in *Smith's Hydraulics*, and to those made by Fteley and Stearns, described in *Transactions American Society of Civil Engineers*, 1883.

VENTURI TUBES.

Mr. Clemens Herschel, C.E., Holyoke, Mass., in a paper read before the American Society of Civil Engineers, December 21, 1887, and printed in their *Transactions*, November, 1887, and January, 1888, recommends the use of a Venturi tube, inserted in the force main of the pumping engine, for determining the quantity of water discharged. Such a tube, applied, for example, to a 24-inch main, has a total length of about twenty feet. At a distance of four feet from the end nearest the engine, the inside diameter of the tube is contracted to a throat having a diameter of about eight inches. A pressure gauge is attached to each of two chambers, the one surrounding and communicating with the entrance, or main pipe, the other with the throat. According to experiments which Mr. Herschel has made upon two tubes of this kind, one of which was four inches in diameter at the throat and twelve inches at its entrance, and the other about thirty-six inches in diameter at the throat and nine feet at its entrance, the quantity of water which passes through the tube is very nearly the theoretical discharge through an opening having an area equal to that of the throat, and a velocity which is that due to the difference in head shown by the two gauges. Mr. Herschel states that the coefficient for these two widely varying sizes of tubes and for a wide range of velocity through the pipe was found to be within two per cent., either way, of 98%. In other words, the quantity of water flowing through the tube per second is expressed within two per cent. by the formula

$$W = 0.98 \times A \times \sqrt{2gh},$$

in which A is the area of the throat of the tube, and h the head, in feet, corresponding to the difference in the pressure of the water entering the tube and that found at the throat.

The accuracy of this form of measurement has not been determined under conditions which apply exactly to pumping engines, but the results here alluded to give promise of its becoming a valuable aid for this purpose, when thoroughly developed.

NOZZLES.

The measurement of water by computation from its discharge through orifices, or through the nozzles of fire-hose, furnishes a means of determining the quantity of water delivered by a pumping engine which can be applied without much difficulty. Mr. John R. Freeman, C.E., of Boston, has carried on a series of investigations upon fire-nozzles, described in the *Transactions American Society of Civil Engineers*, November, 1889, which are of value in this connection. Mr. Freeman's experiments covered a wide range of pressures and sizes, and the results showed that the coefficient of discharge, for a smooth nozzle of ordinary good form, was within one-half of one per cent., either way, of 0.977; the diameter of the nozzle being accurately calipered, and the pressures being determined by means of an accurate gauge attached to a suitable piezometer at the base of the play-pipe.

In order to use this method for determining the quantity of water discharged by a pumping engine, it would be necessary to provide a pressure box, to which the water would be conducted, and attach to the box as many nozzles as would be required to carry off the water. According to Mr. Freeman's estimate, four 1¼-inch nozzles, thus connected, with a pressure of 80 lbs. per square inch, would discharge the full capacity of a two-and-a-half-million engine. To serve the same end, Mr. Freeman suggests, in the *Journal of the New England Water Works Association*, March, 1890, the use of a portable apparatus with a single opening for discharge, consisting essentially of a Siamese nozzle, so called, the water being carried to it by three or more lines of fire-hose.

To insure reliability for these measurements, it is necessary that the shut-off valve in the force main, or the several shut-off valves, should be tight, so that all the water discharged by the engine may pass through the nozzles.

TABLE No. 1.

PROPERTIES OF SATURATED STEAM.

[From Charles T. Porter's treatise on *The Richards' Steam-Engine Indic*

Pressure above zero.	Temperature.	Sensible Heat above zero Fahr.	Latent Heat.	Total Heat above zero Fahr.	We One F
Lbs. per sq. in.	Fahr. Deg.	B.T.U.	B.T.U.	B.T.U.	1
1	102.00	102.08	1042.96	1145.05	.0
2	126.26	126.44	1026.01	1152.45	.0
3	141.62	141.87	1015.25	1157.13	.0
4	153.07	153.39	1007.22	1160.62	.0
5	162.33	162.72	1000.72	1163.44	.0
6	170.12	170.57	995.24	1165.82	.0
7	176.91	177.42	990.47	1167.89	.0
8	182.91	183.48	986.24	1169.72	.0
9	188.31	188.94	982.43	1171.37	.0
10	193.24	193.91	978.95	1172.87	.0
11	197.76	198.49	975.76	1174.25	.0
12	201.96	202.73	972.80	1175.53	.0
13	205.88	206.70	970.02	1176.73	.0
14	209.56	210.42	967.42	1177.85	.0
15	213.02	213.93	964.97	1178.91	.0
16	216.29	217.25	962.65	1179.90	.0
17	219.41	220.40	960.45	1180.85	.0
18	222.37	223.41	958.34	1181.76	.0
19	225.20	226.28	956.34	1182.62	.0
20	227.91	229.03	954.41	1183.45	.0
21	230.51	231.67	952.57	1184.24	.0
22	233.01	234.21	950.79	1185.00	.0
23	235.43	236.67	949.07	1185.74	.0
24	237.75	239.02	947.42	1186.45	.0
25	240.00	241.31	945.82	1187.13	.0
26	242.17	243.52	944.27	1187.80	.0
27	244.28	245.67	942.77	1188.44	.0
28	246.32	247.74	941.32	1189.06	.0
29	248.31	249.76	939.90	1189.67	.0
30	250.24	251.73	938.92	1190.26	.0
31	252.12	253.64	937.18	1190.83	.0
32	253.95	255.51	935.88	1191.39	.0
33	255.73	257.32	934.60	1191.93	.0
34	257.47	259.10	933.36	1192.46	.0
35	259.17	260.83	932.15	1192.98	.0
36	260.83	262.52	930.96	1193.49	.0
37	262.45	264.18	929.80	1193.98	.0
38	264.04	265.80	928.67	1194.47	.0
39	265.59	267.38	927.56	1194.94	.0
40	267.12	268.93	926.47	1195.41	.0
41	268.61	270.46	925.40	1195.86	.1
42	270.07	271.95	924.35	1196.31	.1
43	271.50	273.41	923.33	1196.74	.1
44	272.91	274.85	922.32	1197.17	.1
45	274.29	276.26	921.33	1197.60	.1
46	275.65	277.65	920.36	1198.01	.1

TABLE No. 1.—*Continued.*

Pressure above zero.	Temperature.	Sensible Heat above zero Fahr.	Latent Heat.	Total Heat above zero Fahr.	Weight of One Cubic Foot.
Lbs. per sq. in.	Fahr. Deg.	B.T.U.	B.T.U.	B.T.U.	Lbs.
47	276.98	279.01	919.40	1198.42	.1158
48	278.29	280.35	918.46	1198.82	.1181
49	279.58	281.67	917.54	1199.21	.1204
50	280.85	282.96	916.63	1199.60	.1227
51	282.09	284.24	915.73	1199.98	.1251
52	283.32	285.49	914.85	1200.35	.1274
53	284.53	286.73	913.98	1200.72	.1297
54	285.72	287.95	913.13	1201.08	.1320
55	286 89	289.15	912.29	1201.44	.1343
56	288.05	290.33	911.46	1201.79	.1366
57	289.11	291.50	910.64	1202.14	.1388
58	290.31	292.65	909.83	1202.48	.1411
59	291.42	293.79	909.03	1202.82	.1434
60	292.52	294.91	908.24	1203.15	.1457
61	293.59	296.01	907.47	1203.48	.1479
62	294.66	297.10	906.70	1203.81	.1502
63	295.71	298.18	905.94	1204.13	.1524
64	296.75	299.24	905 20	1204.44	.1547
65	297.77	300.30	904.46	1204.76	.1569
66	298.78	301.33	903.73	1205.07	.1592
67	299.78	302.36	903.01	1205.37	.1614
68	300.77	303.37	902.29	1205.67	.1637
69	301.75	304.38	901.59	1205.97	.1659
70	302.71	305.37	900.89	1206.26	.1681
71	303.67	306.35	900.21	1206.56	.1703
72	304.61	307.32	899.52	1206.84	.1725
73	305.55	308.27	898.85	1207.13	.1748
74	306.47	309.22	898.18	1207.41	.1770
75	307.38	310.16	897.52	1207.69	.1792
76	308.29	311.09	896.87	1207.96	.1814
77	309.18	312.01	896.23	1208.24	.1836
78	310.06	312.92	895.59	1208.51	.1857
79	310.94	313.82	894.95	1208.77	.1879
80	311.81	314.71	894.33	1209.04	.1901
81	312.67	315.59	893.70	1209.30	.1923
82	313.52	316.46	893.09	1209.56	.1945
83	314.36	317.33	892.48	1209.82	.1967
84	315.19	318.19	891.88	1210.07	.1988
85	316.02	319.04	891.28	1210.32	.2010
86	316.83	319.88	890.69	1210.57	.2032
87	317.65	320.71	890.10	1210.82	.2053
88	318.45	321.54	889.52	1211.06	.2075
89	319.24	322.36	888.94	1211.31	.2097
90	320.03	323.17	888.37	1211.55	.2118
91	320.82	323.98	887.80	1211.79	.2139
92	321.59	324.78	887.24	1212.02	.2160
93	322.36	325.57	886.68	1212.26	.2182
94	323.12	326.35	886.13	1212.49	.2204
95	323.88	327.13	885.58	1212.72	.2224

TABLE No. 1.—*Continued.*

Pressure above zero.	Temperature.	Sensible Heat above zero Fahr.	Latent Heat.	Total Heat above zero Fahr.	Weight of One Cubic Foot.
Lbs. per sq. in.	Fahr. Deg.	B.T.U.	B.T.U.	B.T.U.	Lbs.
96	324.63	327.90	885.04	1212.95	.2245
97	325.37	328.67	884.50	1213.18	.2266
98	326.11	329.43	883.97	1213.40	.2288
99	326.84	330.18	883.44	1213.62	.2309
100	327.57	330.93	882.91	1213.84	.2330
101	328.29	331.67	882.39	1214.06	.2351
102	329.00	332.41	881.87	1214.28	.2371
103	329.71	333.14	881.35	1214.50	.2392
104	330.41	333.86	880.84	1214.71	.2413
105	331.11	334.58	880.34	1214.92	.2434
106	331.80	335.30	879.84	1215.14	.2454
107	332.49	336.00	879.34	1215.35	.2475
108	333.17	336.71	878.84	1215.55	.2496
109	333.85	337.41	878.35	1215.76	.2516
110	334.52	338.10	877.86	1215.97	.2537
111	335.19	338.79	877.37	1216.17	.2558
112	335.85	339.47	876.89	1216.37	.2578
113	336.51	340.15	876.41	1216.57	.2599
114	337.16	340.83	875.94	1216.77	.2619
115	337.81	341.50	875.47	1216.97	.2640
116	338.45	342.16	875.00	1217.17	.2661
117	339.10	342.83	874.53	1217.36	.2681
118	339.73	343.48	874.07	1217.56	.2702
119	340.36	344.14	873.61	1217.75	.2722
120	340.99	344.78	873.15	1217.94	.2742
121	341.61	345.43	872.70	1218.13	.2762
122	342.23	346.07	872.25	1218.32	.2782
123	342.85	346.70	871.80	1218.51	.2802
124	343.46	347.34	871.35	1218.69	.2822
125	344.07	347.97	870.91	1218.88	.2842
126	344.67	348.59	870.47	1219.06	.2862
127	345.27	349.21	870.03	1219.25	.2882
128	345.87	349.83	869.59	1219.43	.2902
129	346.45	350.44	869.16	1219.61	.2922
130	347.05	351.05	868.73	1219.79	.2942
131	347.64	351.66	868.30	1219.97	.2961
132	348.22	352.26	867.88	1220.15	.2981
133	348.80	352.86	867.46	1220.32	.3001
134	349.38	353.46	867.03	1220.50	.3020
135	349.95	354.05	866.62	1220.67	.3040
136	350.52	354.64	866.20	1220.85	.3060
137	351.08	355.23	866.79	1221.02	.3079
138	351.75	355.81	865.38	1221.19	.3099
139	352.21	356.39	864.97	1221.36	.3118
140	352.76	356.96	864.56	1221.53	.3138
141	353.31	357.54	864.16	1221.70	.3158
142	353.86	358.11	863.76	1221.87	.3178
143	354.41	358.67	863.36	1222.03	.3199
144	354.96	359.24	862.96	1222.20	.3219

TABLE No. 1.—*Continued.*

Pressure above zero.	Temperature.	Sensible Heat above zero Fahr.	Latent Heat.	Total Heat above zero Fahr.	Weight of One Cubic Foot.
Lbs. per sq. in.	Fahr. Deg.	B.T.U.	B.T.U.	B.T.U.	Lbs.
145	355.50	359.80	862.56	1222.36	.3239
146	356.08	360.35	862.17	1222.53	.3259
147	356.57	360.91	861.78	1222.69	.3279
148	357.10	361.46	861.39	1222.85	.3299
149	357.63	362.01	861.00	1223.01	.3319
150	358.16	362.55	860.62	1223.18	.3340
151	358.68	363.10	860.23	1223.33	.3358
152	359.20	363.64	859.85	1223.49	.3376
153	359.72	364.17	859.47	1223.65	.3394
154	360.23	364.71	859.10	1223.81	.3412
155	360.74	365.24	858.72	1223.97	.3430
156	361.26	365.77	858.35	1224.12	.3448
157	361.76	366.30	857.98	1224.28	.3466
158	362.27	366.82	857.61	1224.43	.3484
159	362.77	367.34	857.24	1224.58	.3502
160	363.27	367.86	856.87	1224.74	.3520
161	363.77	368.38	856.50	1224.89	.3539
162	364.27	368.89	856.14	1225.04	.3558
163	364.76	369.41	855.78	1225.19	.3577
164	365.25	369.92	855.42	1225.34	.3596
165	365.74	370.42	855.06	1225.49	.3614
166	366.23	370.93	854.70	1225.64	.3633
167	366.71	371.43	854.35	1225.78	.3652
168	367.19	371.93	853.99	1225.93	.3671
169	367.68	372.43	853.64	1226.08	.3690
170	368.15	372.93	853.29	1226.22	.3709
171	368.63	373.42	852.94	1226.37	.3727
172	369.10	373.91	852.59	1226.51	.3745
173	369.57	374.40	852.25	1226.66	.3763
174	370.04	374.89	851.90	1226.80	.3781
175	370.51	375.38	851.56	1226.94	.3799
176	370.97	375.86	851.22	1227.08	.3817
177	371.44	376.34	850.88	1227.23	.3835
178	371.90	376.82	850.54	1227.37	.3853
179	372.36	377.30	850.20	1227.51	.3871
180	372.82	377.78	849.86	1227.65	.3889
181	373.27	378.25	849.53	1227.78	.3907
182	373.73	378.72	849.20	1227.92	.3925
183	374.18	379.19	848.86	1228.06	.3944
184	374.63	379.66	848.53	1228.20	.3962
185	375.08	380.13	848.20	1228.33	.3980
186	375.52	380.59	847.88	1228.47	.3999
187	375.97	381.05	847.55	1228.61	.4017
188	376.41	381.51	847.22	1228.74	.4035
189	376.85	381.97	846.90	1228.87	.4053
190	377.29	382.42	846.58	1229.01	.4072
191	377.72	382.88	846.26	1229.14	.4089
192	378.16	383.33	845.94	1229.27	.4107
193	378.59	383.78	845.62	1229.41	.4125

TABLE No. 1.—*Concluded.*

Pressure above zero.	Temperature.	Sensible Heat above zero Fahr.	Latent Heat.	Total Heat above zero Fahr.	Weight of One Cubic Foot.
Lbs. per sq. in.	Fahr. Deg.	B.T.U.	B.T.U.	B.T.U.	Lbs.
194	379.02	384.28	845.30	1229.54	.4143
195	379.45	384.67	844.99	1229.67	.4160
196	379.87	385.12	844.68	1229.80	.4178
197	380.30	385.56	844.36	1229.93	.4196
198	380.72	386.00	844.05	1230.06	.4214
199	381.15	386.44	843.74	1230.19	.4231
200	381.57	386.88	843.43	1230.31	.4249
201	381.99	387.32	843.12	1230.44	.4266
202	382.41	387.76	842.81	1230.57	.4283
203	382.82	388.19	842.50	1230.70	.4300
204	383.24	388.62	842.20	1230.82	.4318
205	383.65	389.05	841.89	1230.95	.4335
206	384.06	389.48	841.59	1231.07	.4352
207	384.47	389.91	841.29	1231.20	.4369
208	384.88	390.33	840.99	1231.32	.4386
209	385.28	390.75	840.69	1231.45	.4403
210	385.67	391.17	840.39	1231.57	.4421

TABLE No. 2.

QUANTITIES OF HEAT CONTAINED IN ONE POUND OF WATER AT VARIOUS TEMPERATURES, RECKONED FROM ZERO, FAHRENHEIT.

[From Charles T. Porter's treatise on *The Richards' Steam-Engine Indicator.*]

Temperature.	Heat contained above zero.	Temperature.	Heat contained above zero.	Temperature.	Heat contained above zero.
Fahr. Deg.	B.T.U.	Fahr. Deg.	B.T.U.	Fahr. Deg.	B.T.U.
35	35.00	155	155.33	275	276.98
40	40.00	160	160.37	280	282.09
45	45.00	165	165.41	285	287.21
50	50.00	170	170.45	290	292.32
55	55.00	175	175.49	295	297.45
60	60.00	180	180.54	300	302.58
65	65.01	185	185.59	305	307.71
70	70.02	190	190.64	310	312.84
75	75.02	195	195.69	315	317.98
80	80.03	200	200.75	320	323.13
85	85.04	205	205.81	325	328.28
90	90.05	210	210.87	330	333.43
95	95.06	215	215.93	335	338.59
100	100.08	220	221.00	340	343.75
105	105.09	225	226.07	345	348.92
110	110.11	230	231.15	350	354.10
115	115.12	235	236.23	355	359.28
120	120.14	240	241.31	360	364.46
125	125.16	245	246.39	365	369.65
130	130.19	250	251.48	370	374.84
135	135.21	255	256.57	375	380.04
140	140.24	260	261.67	380	385.24
145	145.27	265	266.77	385	390.45
150	150.30	270	271.87	390	395.67

CCCCXXXVII.

DISCUSSION OF REPORT OF A COMMITTEE ON A STANDARD METHOD OF CONDUCTING DUTY TRIALS OF PUMPING ENGINES.

[NOTE.—The report of the Committee was presented at the Convention held in Cincinnati, May, 1890. Copies of it were sent to a number of representative engineers and others, with a request that they should discuss it. The following contributions were received previous to November 1, 1890.—*Secretary.*]

Prof. R. H. Thurston.—I have read the report with some care and with much pleasure. There seems little to be said, in addition to the statements made by the Committee, or in criticism of their proposed system of trials of engines and boilers. There certainly is no question of the desirability of formulating a standard system of trial which may serve as a model for general practice, and which will also give us such uniformity of practice as will permit an intelligent comparison of experimental results for scientific and business purposes. The independence of the engine and the boiler, in their economic performance, obviously makes it essential to such a scheme that the method of trial and of report should give measures of efficiency for each, and independently; and the final basis of comparison must as evidently be, ultimately, one which will serve to indicate what proportion of heat energy is utilized. The now usual basis of work done per pound of fuel is entirely uncertain and unsatisfactory, as it involves the chemical composition and the method of handling and burning the coal, as well as the performance of the boiler, in the statement of the efficiency of the engine. Whatever system is adopted should be one which should determine the quantity of energy supplied to the engine in the form of heat, and the proportion of the energy so supplied which is utilized in the performance of the duty of the machine.

The deduction of the Committee is thus evidently right; and the proper measure of energy supplied is the heat-unit. One million heat-units seems to be as satisfactory, as a standard unit in this comparison, as could be found; both because it is a figure easily handled in computation, and because it so closely measures the value of the better classes of coal burned in good boilers. The custom has lately become general of assuming the performance of an hypothetical boiler evaporating 10 lbs. of water "from and at" the boiling point, as a basis of computation of duty of the engine. This substitutes for that figure another, differing from it but 3½ per cent., and one which is, to my mind, better, and on the whole, practically more satisfactory. The use of the unit or scale-measure 1,000,000 heat-units in place of the older measure 100 lbs. of coal is in all respects advantageous. This will be still more desirable if the performance of the boiler is finally reduced in such cases to a similar basis. The suggestion of the Committee to this effect is one which I would cordially endorse.

The second suggestion of the Committee — that the duty should always be computed from the quantity of heat supplied the whole "plant"—seems sensible and logical. It should, I think, be adopted as proposed in Article 4 of the Report, without qualification. The quality of steam must evidently be determined accurately, in order that the amount of heat supplied in available form may be known, since it is only that amount for which the engine should be held responsible in its conversion of heat into work. Heat carried over with the steam in the water thus taken to the engine is not only unavailable, but its presence in that fluid will always more than proportionally reduce the performance of any engine of good economical form. No engine of good design and intended for high duty can do good work except the steam be dry.

The method of measuring the quantity of work done by the determination of the quantity of water raised and of the head against which it is pumped is a matter of supreme importance, and I am not at all certain that the Committee have justified their proposal to take the plunger-displacement, corrected for "slip," as the standard. I know of no way of determining with any degree of certainty the total amount of the slip or waste in the pumps, except by the use of one of the other plans mentioned as alternatives. I have myself used the weir in all cases, and, so far as I have been able to judge, it has proved satisfactory and is probably accurate. Certainly, using Francis' formula and method of construction of the notch, all trials could be brought to substantially the same gauge, and that a fairly exact one; in which case the results will always be comparable and approximately exact—probably sufficiently so for all purposes of the engineer and of the proprietor of the engines. My own impression is that this "indirect" mode of measurement, as the Committee designate it, is more safe and exact than the direct measurement of the plunger-displacement can be made. Even if I were desirous of measuring the slip for the purposes intended by the Committee, I should, I think, adopt the weir if practicable; and if it is to be thus employed, I should prefer to take its results as a direct measure of the water delivered rather than to travel around the circle to give figures of plunger-displacement. The precaution suggested by the Committee, the avoidance of error through leakage, cannot be too carefully considered. It is often the source of uncertainty, if not of actual and serious error.

I have been much interested in the results of investigations recently made in the use of the nozzle and of the "Venturi tube" in this work. If, as now seems probable, the use of these expedients may be resorted to with confidence in the resulting measures of efficiency, it will often be found a most convenient method of avoiding much trouble and expense. The papers of Mr. Herschel and of Mr. Freeman, referred to in the appendix to the report, are particularly valuable as giving admirable accounts of the two methods as adopted by those distinguished engineers; and I should not hesitate, I think, to use either method as might prove most convenient in any particular case, the construction and use of the weir not being practicable. The large scale on which these gentlemen have worked relieves us of any uncertainty as to the applicability of either method to our purpose.

The proposition made by the Committee, that the work of the engine should be measured outside the boundaries of the construction of which the efficiency is to be determined, is evidently right. The builder should be held responsible for wastes in his own machine; but he should not suffer because of the introduction of friction-losses by the constructor of the distributing system. The only way of securing reliability and comparability of reported duties is to draw the line at delivery into the rising main, crediting the engine with net work done at that point, as proposed.

The recommendation that no air be admitted (Art. 6) is not only to be taken as a recommendation, but as an essential requirement, if the proposal of the Committee to adopt the plunger-displacement as its measure of water delivered is acted upon, and should, I think, be stated more strongly and imperatively. It seems to me a serious question whether many seemingly improbable reports of high duty, in earlier years especially, may not have come of such error as is produced by failure of the pumps to fill and by the introduction of air. It is to be recollected that the entrance of air into the rising main not only reduces the quantity of water delivered below that given by measurement at the plunger, but it has also the effect of reducing the head. The celebrated Wicksteed trials have always seemed to me to give results which are extremely improbable, and I have been much inclined to agree with many of the able engineers of their time and since, who have attributed the, in their opinion, extravagant claims made for the old Cornish engines to this fault in their operation during trial. It is the more probable, as it is not an uncommon method of easing their movements in regular operation. The air sometimes gets in through a loosely packed stuffing-box, sometimes through leaky joints, and often through a cock fitted in the suction side for that purpose. There is no reason to suppose that any such errors of reported duties as may stand recorded were due to deliberate deception.

The adoption of the tables of steam-pressures and temperatures by the Committee, from *Porter on the Indicator*, gives rise to the question which always presents itself on such occasions: of the many sets of tables now available, which is the most accurate, and which is on the whole best adapted to this work? Mr. Porter's well-known exactness and care, and the fact that he obtained his figures originally directly from Regnault, make it seem probable that they are perfectly reliable. But the handy and often-quoted tables of D. K. Clark, those of Nystrom, those of Haswell, and others of the older dates, differ in small degree usually, but enough to make it wise to select some one as standard; while the later work of production of more elaborate tables has given rise to still others which we would all, I presume, like to be able to use with confidence. I have used those of Buel in my own books and in some of my other work. Their author was very exact in his habits of work, and gave us, for the first time, all that the engineer desires to find in such tables. The latest of these tables, those of Professor Peabody, are beautifully printed and conveniently published, and are, so far as I have been able to check them, apparently very accurate. There are, nevertheless, discrepancies throughout the whole list of authors, and it would be a great boon to the engineer to secure such a comparison of authorities, so-called, as would determine definitely the extent, reason, and effect of such discrepancies, and the relative accuracy and value of the several tables for his purposes. The Committee on Steam-boiler Trials adopted the Porter tables without this definite knowledge of comparative merit; but they felt that, all things considered, it was at the time certainly the best choice. They were accessible to all; for few, if any, engineers are without the book in which they are printed; they have been revised and re-revised; they follow the experimental work of Regnault, and contain the essential data. It may be safely presumed, I imagine, that the committee here reporting adopted the same view.

The Committee suggests that the duration of the trial should be restricted to ten hours. The usual length of trial is twenty-four hours, or more, in important cases. It is easy, however, adopting the system here proposed, to secure as accurate a result in ten hours as in any greater length of time. One who has worked, as some of us have, a week at a time, in conducting such trials, with sleep only in alternate twenty-four hours, and not much at that, will be perfectly willing to subscribe to any plan which, without lessening the reliability of the work, may restrict it to shorter periods.

The Committee has done a good work, and a most important one, and deserves, and I hope will receive, heartiest thanks for its services.

Mr. F. W. Dean.—Passing by the methods of carrying out the details of a standard test of a pumping engine, which can safely be intrusted to the Committee, I think that the makers and users of pumping engines, and scientific men generally, are to be congratulated upon the completion of this work. It will soon be wondered at that it was not undertaken years ago.

It seems hardly necessary to say that the heat-unit method of testing a pumping engine is the only logical one. Many engines are weighted down with inferior boilers, as everybody knows, and it conveys no more idea of the engine to say that it performed certain work with 100 lbs. of coal than merely to say that its cylinders are of a certain size. Every accomplished engineer of the present day strives for exact methods of measurement and expression, and nothing more is to be desired when it is said that a certain engine performed certain work when using a certain number of heat-units. This leaves the boiler to rest upon its own merits, and it will receive none the less attention, for it is still an absorber of money. Its own performance will be expressed in exact terms.

I sincerely believe that mechanical engineers should always be leading those with whom they are associated, and should never look backward because they cannot be understood. It has been objected to the heat-unit method that it cannot be understood by city water boards and unprofessional men, and that therefore the old "100 lbs. of coal" method of rating duty should be retained. This would be most deplorable, and would effectually block progress. Progress is difficult enough under any circumstances, when dealing with the average layman. The probability is that very few members of water boards understand what a foot pound of work is, and therefore what is meant by duty of pumping engines, however expressed. They have become habituated to the usual expression, and if they are of average brightness they can soon talk of duty when referred to the new unit.

Mr. Albert F. Hall.—If it is desired to accurately determine the amount of water pumped, it can only be done by weir or nozzle measurement. If, however, the pump is to be used as a meter, allowance for leakage past the piston, or plunger, is the only one which can be legitimately made. Leakage past the valves cannot be determined, because this leakage will vary with the pressures, and these are not the same when the pump is at rest and in motion. All allowances for imperfect filling due to the presence of air, as determined from indicator cards, should not be permitted.

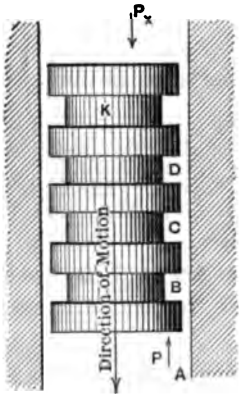


FIG. 180.

See "Versuche zur Klarstellung der Bewegung selbstthätiger Pumpenventile," von C. Bach. *Zeitschrift des Vereins deutscher Ingenieure.* Bd. XXX. and XXXI.

Allowance for leakage past the plungers may be determined as follows:

Let K (Fig. 180) be a grooved plunger moving in a cylinder in the direction shown, and let P be the pressure on the front, or forcing side, of the piston, and P_x that on the back, or draughting side. Let P_1, P_2, be the corresponding pressures in the grooves or recesses B, C, D; δ the density of the fluid, and v the velocity of the fluid from one groove to the other. Then the fluid, in passing from A to B, or from B to C, etc., will suffer a loss of pressure represented by

$$\frac{cv^2\delta}{2g} = P - P_1$$

$$= P_1 - P_2$$

$$= P_2 - P_3$$

$$= P_{x-1} - P_x$$

and consequently the total loss will be the sum of these, or

$$\frac{xv^2c\delta}{2g} = P - P_x,$$

where x is the number of ridges, in the piston shown $x = 5$, and where c represents the coefficient of resistance; consequently

$$v = \sqrt{2g\,\frac{P - P_x}{cx\delta}}.$$

If, on the contrary, the piston is made without grooves, or $x = 1$, then

$$v = \sqrt{2g\,\frac{P - P_x}{c\delta}},$$

where l is the length of the piston. This formula shows that the quantity of water leaking by the plunger in a given time is independent of the velocity with which the plunger moves, but varies directly as the square root of the difference of pressure upon its two sides, and inversely as the square root of the product of the density and number of ridges.

It is, therefore, correct to determine the loss through leakage past the plunger by ascertaining its amount when the plunger is at rest, allowance being made, however, for the difference of the pressures on the sides of the piston when the pump is in motion, and those when the pump is at rest.

For example, let P and P_x be the pressures on the forcing and draughting sides, respectively, when the piston is in motion, P_4 and P_5 the corresponding pressures when the piston is at rest, then the quantity to be used in calculating the loss will be determined by the formula,

$$Q = Q_1 \sqrt{\frac{P - P_x}{P_4 - P_5}},$$

where Q_1 = quantity determined by experiment when the piston was at rest.

(See Weisbach's *Mechanics*, Vol. III., page 1025, German edition; Braunschweig, 1855.)

Regarding this question it may be well to quote the remarks of my esteemed friend, the late Mr. Hoadley. Mr. Hoadley said:

"The loss by leakage at the pump plunger is directly proportioned to the quantity of water pumped; that is to say, it is directly proportionate, first, to the time; and, since the quantity pumped at a certain rate is also proportionate to the time, the leakage is proportionate to the quantity pumped, and that irrespective of the velocity of piston, or of the length of the stroke. Of two engines pumping a given quantity of water, under a given pressure, that engine having the larger plunger will be subject to greater leakage than that with the smaller plunger. The *percentage* of leakage in the engine increases with the decrease in the speed of the plunger. If the plunger were standing still the quantity of water leaking by the plunger, in proportion to the water pumped, would be infinite, however small the leak. If the plunger ran just fast enough to make up for the leakage, no water would be delivered, and then the leakage, instead of being infinite, would be simply 100 per cent. If the velocity be doubled, twice as much would apparently be pumped, but one-half would slip back and the leakage would be 50 per cent. And as the quantity pumped is increased the ratio of the loss due to leakage is diminished, whether the quantity pumped be made up by high piston speed and a small plunger, or by low piston speed and a large plunger. The ratio is to the quantity and not to the piston speed; that is, the ratio of the cubical displacement of the plunger."

Mr. William Kent.—"When an injector is used a deduction is to be made for the increased temperature of the water derived from the steam which it consumes." This is surely an error, and it should be corrected before final printing of the report in the *Transactions.*

The temperature of the water should be taken before it enters the injector, and if the steam is taken directly from the boiler being tested, as it should be, no correction or deduction should be made, the steam "consumed" in heating the feed-water going back into the boiler in the shape of hot water, no heat being lost except by radiation.

Mr. A. M. Wellington.—The report of the Committee on a Standard Method of Conducting Duty Trials of Pumping Engines proposes a radical change from the past practice in substituting a duty unit of "1,000,000 heat-units" for the old "100 lbs. of coal." There is the plainest necessity for this in the widely varying qualities of coal, as well as in the occasional use of wood, petroleum, natural gas, and other fuels not coal; but the language of the Committee, in describing it, is, at least, highly ambiguous and contradictory in its terms. They say (all italics mine):

"One million heat-units is a quantity of heat corresponding to that *produced* in the combustion of 100 lbs. of coal, which *gives out* 10,000 heat-units per pound, or which *produces an evaporation* of $\frac{10,000}{965.7} - 10.355$ lbs. of water from and at 212° per pound of fuel."

It will at once be seen that this is an unscientific use of language, to say the least, referring as it does to two quite distinct things, the heat "produced" and the heat put into the steam, as if they were one and the same thing; and this confusion goes so far that it is impossible to determine, except from the context, what the Committee really do mean. For just before they had been speaking of the uncertain and variable value of "100 lbs." of coal, and proposed to substitute for it "1,000,000 heat-units," which certainly implies that they propose to use as a unit the theoretical heat-units *in the coal,* and not in the steam. This is the only way, in fact, in which the new unit can be used for tests of boiler efficiency which shall be generally comparable. With this interpretation the first half of the sentence quoted above is entirely consistent, but the last half of it is wholly inconsistent.

The last half of the sentence above quoted expresses what seems to be really intended, however, and this should be made so clear as to be unmistakable. But, even then, the form given on pages 23–25 for boiler test reports does not seem to be consistent with this intention, for if the unit of comparison is to be units of heat put into the steam, and boiler tests are to mean anything when made with different coals and compared with each other, instead of the one line: "Number of pounds of coal required to supply 1,000,000 B. T. U.," there should be two lines: "Theoretical value of 1 lb. of coal in B. T. U.," and "Number of heat-units produced, to supply 1,000,000 B. T. U." It is true that the theoretical value of the coal is often not known, but it is readily determined from a chemical analysis, and is usually already of record. To be consistent, the committee should make this change in their form for boiler tests. It already requires that the moisture in the coal shall be determined.

With these exceptions the report seems to be an excellent document, calculated to do much good in unifying practice, and enabling various tests to be readily compared with each other.

Mr. John R. Freeman.—My comments at this time will refer only to that part of the Committee's report relating to the measurement of the water pumped.

Methods and apparatus for tests like this, or for general use by different engineers at widely differing times and places, in not only comparing with strict fairness the merits of different machines, but also for serving with exact justice as a basis of settlement in contracts between buyer and seller, require first of all a straightforwardness and simplicity which shall place the result above suspicion.

I think I am voicing a general sentiment to say that when, in discussing past tests of pumping engines, the fact appears that the water was metered by the strokes of the pump itself, suspicion as to the exactness and certainty of the figures is at once aroused, and often with good reason.

The method of using the pump as its own water meter is a very convenient one, and for this reason has often been adopted. Moreover, the range of error in its use in connection with the leakage test proposed by the Committee would, for a guess, probably very seldom exceed 5 per cent. or possibly 10 per cent. It is thus accurate enough for many purposes, but *is it* a method certain enough to be recommended by this society for duty trials on pumps of all sizes, speeds, makes, and conditions of repair?

The method recommended will generally tend to give results which are slightly too high and slightly in favor of the builder of the pump.

Some of the sources of error in the method proposed by the Committee are:

1st. That it neglects wholly the back-flow of the water immediately under the lifted valves at end of stroke. A pump *could* undoubtedly be so built and so adjusted that this would be zero. Or it is probable that in a pump running at high speed, particularly if its pipes were small, or if it took its supply under a head, the momentum of the rapidly moving water column might cause a "negative slip" of considerable magnitude.

A positive slip or loss by back-flow will, in all probability, occur with almost any ordinary pump. To get an idea of the per cent. of error this may introduce, we may first assume that the back-flow equals the cylinder of water standing under the full area of the valve-disk, and of height equal to lift of valve. Then for a plunger 1 foot in diameter and of 1 foot stroke, with valve area 50 per cent., and six valves in a set, a very simple computation shows the loss in suction and discharge valves together would be equal to 4 per cent. for the case assumed, even though we considered the amount of lift as only half the lift necessary to give a water way under edge of valve equal to the area through its ports. With a pump in good first-class condition I have no idea the back-flow would be so great as just computed.

If it happened that valve movements were sluggish by reason of inertia of valves, feeble springs, or a little friction of valve on its stem, then the back-flow *might be very much greater* than under the conditions assumed just above.

2d. It appears reasonable to suppose that the leakage past a plunger sliding through an unpacked ring, which is now the most common and approved type for the ordinary Duplex pumping engines, may be somewhat different at full speed from that at rest.

The movement of the surface of the plunger in a direction

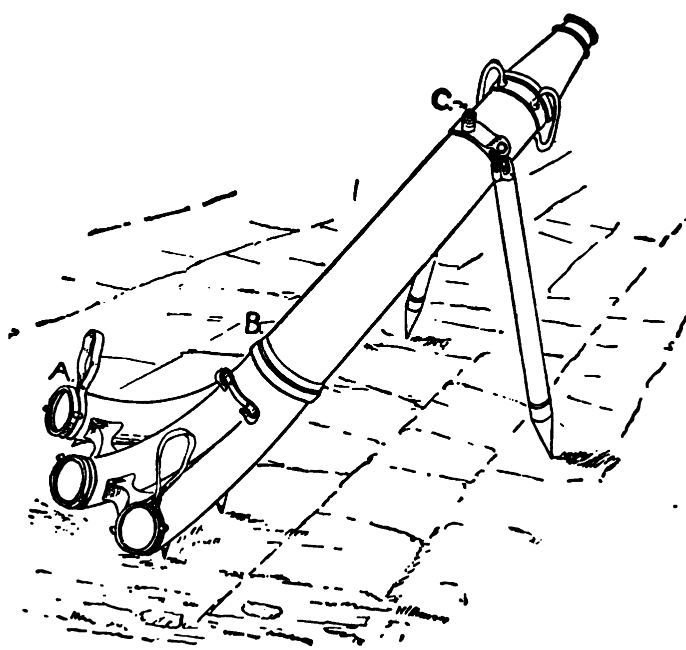


FIG. 178.

This sketch was made before the apparatus itself was built, and the apparatus is even less unwieldy than the sketch might imply. In fact, to-day, the apparatus complete, with all fittings, lies packed for transportation in a common "steamer trunk" 30 inches long by 18 inches wide by 12 inches deep.

The peculiarity of this Siamese nozzle lies in the *very great care taken to so shape the waterways where the three streams unite and bend upward that no whirls or eddies shall be produced in the current of water* past the piezometer and approaching the nozzle.

First. The bends upward are circular arcs carefully made tangent to the axis of the play-pipe, and made plane curves, avoiding even the slightest warping or corkscrew-like twist, lest the water current be thereby set rotating.

Second. The cylindrical cross-section of the waterway of each of the three branches was carefully merged into a V-shaped section where the branches united near *B*, thus preserving a thin partition of metal between each, so the currents do not unite until the straight single pipe is reached at *B*.

Third. The area of each waterway was carefully and gradually reduced as the branches came together going toward *B*, so that the water moves about 15 per cent. faster at *B* than at *A*. This gradual convergence tends to prevent eddies in the water at the concave side of the bend.

Fourth. Hollow cross-connections between the three branches at *A* were formed in the casting, so that in case one line of hose happened to be delivering more water than the others, the currents would, under the influence of the diminished area at *B*, tend to equalize, and thus give a more uniformly distributed current in pipe approaching the nozzle.

Fifth. To further reduce the liability to twisting of the currents, a thin 8-way "rifle blade," so called, extends from *B* for 16 inches toward *C*, this being set so that the blades halve the current from each of the three branches. At *C*, a short pipe serves to connect the pressure-gauge to a hollow ring covering four very carefully made piezometer orifices, whose inner edges are carefully finished flush with the inner surface of pipe, and which are also drilled exactly at right angles get loosened by the working of the pump, and pass out, thus escaping detection.

A method endorsed by this society ought not to be merely one which under *ordinary conditions* will *probably give* the truth.

It should be a method which, when followed in testing a pump, be it small or large, with high piston speed or low, fly-wheel or steam-controlled, stroke of positive or variable length, and whether pumping filtered water or sewage, should lead to *certain* results whose limits of possible error are clearly defined.

THE NOZZLE AS A WATER METER.

The Committee have kindly alluded to some experiments of my own on nozzles.

I believe these experiments to be of great interest and importance in reference to the subject now under discussion.

The following cut, reproduced from the *Journal of the New England Water Works Association*, shows an apparatus which I designed with much care and had built a few months ago especially for some tests on certain small pumping engines, but which, by reason of pressure of business, I have not yet found time to test and calibrate.

I have, however, carefully tested a similar but less perfect apparatus.

This apparatus, all complete, only weighs about 75 lbs. One man can easily move it from place to place after it has been set up; and it can be taken out from the packing-case, set up all ready for use in 10 minutes, and yet will, I think, prove equal to the gauging of a flow of 1,200 gallons of water per minute (or possibly 1,400 gallons per minute, which is at the rate of 2,000,000 gallons per 24 hours) *with an accuracy and certainty not excelled by the best arranged weir.*

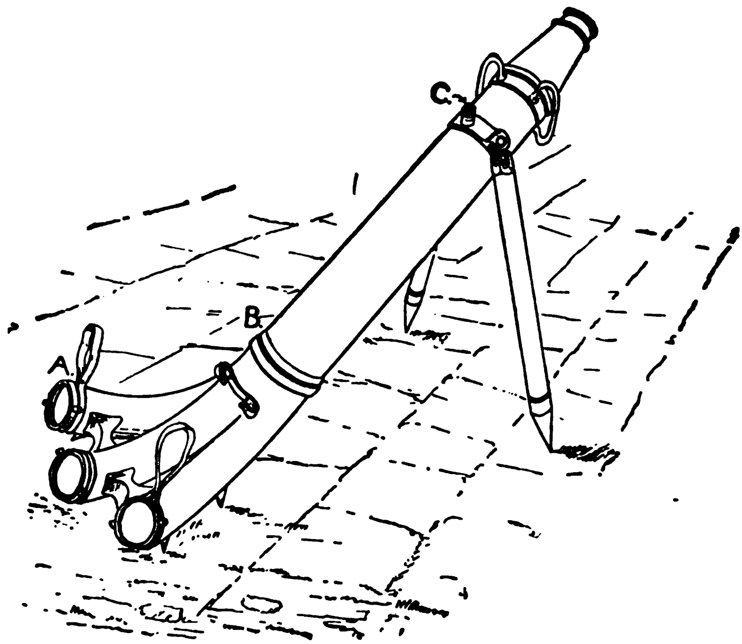


FIG. 178.

This sketch was made before the apparatus itself was built, and the apparatus is even less unwieldy than the sketch might imply. In fact, to-day, the apparatus complete, with all fittings, lies packed for transportation in a common "steamer trunk" 30 inches long by 18 inches wide by 12 inches deep.

The peculiarity of this Siamese nozzle lies in the *very great care taken to so shape the waterways where the three streams unite and bend upward that no whirls or eddies shall be produced in the current of water* past the piezometer and approaching the nozzle.

First. The bends upward are circular arcs carefully made tangent to the axis of the play-pipe, and made plane curves, avoiding even the slightest warping or corkscrew-like twist, lest the water current be thereby set rotating.

Second. The cylindrical cross-section of the waterway of each of the three branches was carefully merged into a V-shaped section where the branches united near *B*, thus preserving a thin partition of metal between each, so the currents do not unite until the straight single pipe is reached at *B*.

Third. The area of each waterway was carefully and gradually reduced as the branches came together going toward *B*, so that the water moves about 15 per cent. faster at *B* than at *A*. This gradual convergence tends to prevent eddies in the water at the concave side of the bend.

Fourth. Hollow cross-connections between the three branches at *A* were formed in the casting, so that in case one line of hose happened to be delivering more water than the others, the currents would, under the influence of the diminished area at *B*, tend to equalize, and thus give a more uniformly distributed current in pipe approaching the nozzle.

Fifth. To further reduce the liability to twisting of the currents, a thin 8-way "rifle blade," so called, extends from *B* for 16 inches toward *C*, this being set so that the blades halve the current from each of the three branches. At *C*, a short pipe serves to connect the pressure-gauge to a hollow ring covering four very carefully made piezometer orifices, whose inner edges are carefully finished flush with the inner surface of pipe, and which are also drilled exactly at right angles

A properly constructed nozzle will prove equal or slightly superior in accuracy to the weir for such cases; because,

1st. There is, with a nozzle of substantially this form, no uncertainty as to degree of contraction.

2d. Although the hook-gauge properly mounted for measuring weir depths is an instrument of wonderful precision, a mercurial-pressure gauge may equal it in resulting precision by reason of working upon a quantity of so much greater magnitude.

For example, an error of $\frac{1}{16}$ inch (due to oscillations, settlement, improper location of gauge, etc.), in the depth of six inches, is 1 per cent. of the total depth, the same as 1 lb. would be of a total pressure of 100 lbs., but this error of 1 per cent. in depth over the weir would affect the computed quantity nearly three times as much as would 1 per cent. error in determining the pressure upon the nozzle.

The discharge varies in the first case as the square root of the third power, and in the second case merely as the square root.

The nozzle, for cases where the discharge can be wasted, possesses some considerable advantage in accuracy over the Venturi tube by reason of the fact that only a single pressure is measured; second, the play-pipe leading to the nozzle can be large enough to give a slow velocity past the piezometer, and thus in great measure avoid errors such as those which might affect the indications of the piezometer at the throat of the Venturi, where the velocity is so high that any little burr or projection at the piezometric orifice would have much more influence upon the reading of the gauge; third, the magnitude of the quantity measured and furnishing a basis for the computations is greater, and may, therefore, be measured with a higher degree of relative precision.

It is further stated by the committee that the objection to the indirect modes of measurement, as by the use of weir, tube, or nozzle, is that the person making the tests must educate himself in the manipulation of the apparatus chosen, so as to be himself assured of the reliability of the data.

We may, therefore, ask which is the simpler and most easy to be assured of. To, use the nozzle in the first case, and—

1st. Make sure that outlet gates of pump are tightly closed, and notice that the water is not leaking out over the floor at the couplings of the hose; and also, before or after the test, examine the piezometer holes leading from the play-pipe outward to the pressure-gauge and see that they are flush and normal; caliper the diameter of nozzle outlet and verify the pressure-gauge; and finally, make a short and simple computation based on one of the very simplest and most firmly established of all hydraulic formulæ.

Or in the second case, to use the pump as its own water meter, and investigate the various sources of error to which I have called attention in the first part of my discussion.

The Committee further stated as an objection the necessity for the use of coefficients which cannot be verified by the observer and which may not apply exactly to the case in hand.

In reply to this it may be said that even if the coefficient for the type of nozzle recommended *were neglected altogether*, the result would be no more inaccurate than metering by plunger-displacement is very likely to be, by reason of the back-flow of water under the valve.

Excellent experiments by different observers and at different times and with nozzles differing materially in form, have shown this coefficient of discharge to affect the computed result only about 2½ per cent. at most. And for this smooth and simple nozzle so arranged as to allow of but little friction-loss between gauge connection and orifice, 1 per cent. will cover the uncertainty from the use of the coefficient.

Of course, after any piece of apparatus was once calibrated the margin of uncertainty would be smaller.

Everything about water measurement by this apparatus in plain sight, and next to the use of an absolutely tight a comparatively smooth cistern it is hard to conceive of anyth simpler and more certain, for the case in hand, than the noz

This one 3-way Siamese shown in the sketch would prob take the whole discharge of a "2,000,000-gallon" pump thro its 2½-inch nozzle; but one could easily be designed and l to take that of a 4,000,000-gallon pump, or two or three of size shown could be used.

The device is in no way hampered by patents, but is a service of the public.

The main point about which I have apprehension is th pulsations of the pump may cause inconvenience. Conc this, experiments on various types of pumps can alone de

A properly constructed nozzle will prove equal or slightly superior in accuracy to the weir for such cases; because,

1st. There is, with a nozzle of substantially this form, no uncertainty as to degree of contraction.

2d. Although the hook-gauge properly mounted for measuring weir depths is an instrument of wonderful precision, a mercurial-pressure gauge may equal it in resulting precision by reason of working upon a quantity of so much greater magnitude.

For example, an error of $\frac{1}{16}$ inch (due to oscillations, settlement, improper location of gauge, etc.), in the depth of six inches, is 1 per cent. of the total depth, the same as 1 lb. would be of a total pressure of 100 lbs., but this error of 1 per cent. in depth over the weir would affect the computed quantity nearly three times as much as would 1 per cent. error in determining the pressure upon the nozzle.

The discharge varies in the first case as the square root of the third power, and in the second case merely as the square root.

The nozzle, for cases where the discharge can be wasted, possesses some considerable advantage in accuracy over the Venturi tube by reason of the fact that only a single pressure is measured; second, the play-pipe leading to the nozzle can be large enough to give a slow velocity past the piezometer, and thus in great measure avoid errors such as those which might affect the indications of the piezometer at the throat of the Venturi, where the velocity is so high that any little burr or projection at the piezometric orifice would have much more influence upon the reading of the gauge; third, the magnitude of the quantity measured and furnishing a basis for the computations is greater, and may, therefore, be measured with a higher degree of relative precision.

It is further stated by the committee that the objection to the indirect modes of measurement, as by the use of weir, tube, or nozzle, is that the person making the tests must educate himself in the manipulation of the apparatus chosen, so as to be himself assured of the reliability of the data.

We may, therefore, ask which is the simpler and most easy to be assured of. To, use the nozzle in the first case, and—

1st. Make sure that outlet gates of pump are tightly closed, and notice that the water is not leaking out over the floor at shall be in its normal condition, regular in supply, constant in pressure, even and steady firing, and that no more labor is required than in the ordinary working of the plant, and these conditions require an absolute separation of the observation for an intelligent report upon one or both.

The points in relation to the steam used in the jackets and feed pumps are also well taken. There seems to be much diversity of opinion and practice on this subject. In several tests which I have witnessed the steam condensed in the jackets was trapped back into the boiler and no record made of it, either for the benefit of engine or boiler.

In regard to feeding the boiler some machines require a separate pump and a separate boiler to furnish steam to the feeding arrangement; in others the feed pumps are attached to the pumping-machine; but in all cases I think the work done by the feeding arrangement should not be included in the duty performed by the pumping engine, but a series of separate observations be made for this work, and a record made of it in such a manner that it can be shown as a distinct item.

In determining the amount of work done by the pumps, the plunger-displacement is open to many objections, all of which the Committee fully understand.

I think an arbitrary percentage of slip should be decided upon beforehand, and a clause inserted in the specification to that effect; and for this reason, that I believe the leakage through the pump by incomplete seating of the valves, slip round the plunger, is far greater when the pump is running than when at rest. Of course, if it is possible to obtain a weir measurement it is to be insisted upon.

In regard to testing the quality of steam, I have found the barrel calorimeter to give as fair results as any arrangement I have yet tried, if properly handled, and the instruments used in observing sensitive and correct.

The form presented for the report of the test is good, and, connected with the boiler form, will give a clear and rapid comparison with other tests made in a similar manner.

It is to be hoped that each member of your Society engaged in making tests of pumping machinery will be able cordially to endorse the standard presented by your Committee, and that it will eventually prove itself the *best* "Standard Method of Conducting Duty Trials of Pumping Engines."

Mr. Geo. H. Barrus (*Chairman of Committee*).—I have carefully looked over the remarks which have been made by Messrs. Wellington, Kent, Hall, Thurston, Freeman, and Dean, and I take this opportunity, in behalf of the Duty Trial Committee, to offer a few words in reply.

Mr. Wellington criticises the language of the report in that part of Section 2 referring to the 1,000,000 heat-unit standard. I think that no one will be misled by the language of the Committee after reading all that is given describing the manner in which the new unit is applied. The language to which he refers is not, however, strictly technical, and, in view of the criticism, suitable changes have been made in the revised copy. I do not endorse Mr. Wellington's suggestion to change line 30, in the Table of Boiler Results, as he proposes. This quantity, that is, "Number of pounds of coal required to supply 1,000,000 B. T. U.," is introduced so that by a simple calculation the results of the duty trial, according to the new unit, can be figured on the old basis of 100 lbs. of coal. It would be desirable, however, to add a line to the table, as follows: "Total heat of combustion of the coal obtained by calorimeter trial," which would show the theoretical heating value of the fuel, and accomplish the end which I think Mr. Wellington has in view. This addition has therefore been made in the revised copy.

Mr. Kent objects to taking the temperature of the feed-water discharged by an injector, and correcting for the heat added by the instrument, stating that this is evidently an error. If he will refer to the context, he will see that this is specified in a case where the water leaving the injector passes through a heater before entering the boiler. If, for example, the water left the heater at 200°, and entered the heater at 125°, being supplied to the injector at 75°, it would not be proper to work out the results of the test from the 200° reading. This reading would be corrected for the increase of temperature due to the use of the injector, which in this case would be 50°, and the actual temperature to be used, without going into the equivalent in heat-units, would be $200° - 50° = 150°$.

Mr. Hall is to be thanked for the analytical investigation as to the problem of leakage by the plunger, and the formula which he gives, showing the relation between the velocity of the plunger, pressure on the two sides, number of grooves, and density; as also for the formula for determining the allowance to b made for the differences of pressure under the running condi tions, and those which obtain during the leakage test whe the engine is at rest. The quantity of water which passes b the plunger on the leakage test should be corrected for the diffe ence of pressure referred to, especially where that difference i large, and it is desired to make the determination with gre accuracy. The formula given enables this to be done.

Professor Thurston's interesting review of the report make reference to two points which call for notice. One of these i the recommendation of the Committee to use Porter's Stea Tables for making the necessary computations. In explanatio of the course which the Committee has taken in this matter, may say that they were guided mainly by the previous action c the Boiler Test Committee. The Society, by accepting the repo of that committee, has already endorsed these tables, and w could not do otherwise than concur. The professor remark that the Committee have hardly justified their recommendatio to use the plunger-displacement method of determining th work done. He inclines to what the Committee have terme the "indirect modes" of measurement, by the use of the wei tube, or nozzle. Mr. Freeman's criticism is in the same line though of more direct and forcible character, and they ma both be taken up together. The Committee are fully aware tha the method proposed is not entirely free from objections, an they have frankly expressed this feeling in the report. They d not claim that the "slip" of the pump can be fully determine by simply measuring the leakage in the manner proposed Leakage is only one of three elements which go to make up slip the other two being imperfect filling, due to the presence of air and "back-flow" through the valves during the act of closing One of these it is proposed to overcome by allowing no air to b "snifted in" during the trial. The air which finds its wa through the stuffing-boxes and unsound pipe connections, an that originally suspended in the water, are believed to be s trifling that for practical purposes they may be disregarded The slip may thus be narrowed down to two components, namely leakage and back-flow. Back-flow does not produce such seriou results as Mr. Freeman is led to suppose. This action occur only when the plunger is at the end of the stroke. To be more exact, it does not occur until after the plunger has come to a dead stop, and has commenced the return stroke. Under these circumstances, the forward motion of water through the valves, just previous to the time when back-flow begins, has reached a very low velocity, and the valves are lifted from their seats a correspondingly small distance. The calculation whereby Mr. Freeman obtains a loss of 4 per cent. in the instance which he cites, requires a lift of the valves amounting to about three-quarters of an inch. It is doubtful if in reality the valves are lifted to anything like this extent, even during the middle portion of the stroke, when the water is passing through at the greatest velocity. Certainly the lift would be reduced to a much smaller distance at the time when the slow motion is reached at the end of the stroke. I should be quite surprised if it exceeded one-eighth of an inch at this time, and it is not improbable that it would be so small as to be almost imperceptible. One of the members of the committee, Mr. Nagle, made experiments on the lift of a double-beat pump valve, which he designed and used on the Nagle high-service pumping engine, Providence, R. I., by attaching a rod to the center of the valve, so arranged as to record its movements upon a steam-engine indicator outside, the rod acting through a freely working stuffing-box. Copies of diagrams which he obtained are given in a paper which was read at the Erie meeting, in May, 1889, entitled "Cornish or Double-beat Pump Valves."* Mr. Nagle's conclusions were that "the lift of a valve is exactly proportioned to the velocity of the plunger." It is believed, from these considerations, that the total amount of back-flow, with valves of good construction, will never exceed 1 per cent. of the quantity of water actually pumped, and this is comparatively insignificant.

Leakage, then, appears to be the most important factor in the production of slip, excepting, of course, that occurring in engines with outside packed plungers. The loss from leakage is liable to be considerable, for it must be remembered that in many engines the plunger consists of a solid plug, without packing other than a series of grooves, and a certain amount of leakage must be allowed in order that the plunger may be loose enough to work without undue friction. This is especially true when the pump is one which has been in service for a considerable period of time, and the plunger has become reduced in size by wear or abrasion of the surfaces. Though the determination of the

* Vol. X., page 527, *Transactions A. S. M. E.*, 1889

sity; as also for the formula for determining the allowance to be made for the differences of pressure under the running conditions, and those which obtain during the leakage test when the engine is at rest. The quantity of water which passes by the plunger on the leakage test should be corrected for the difference of pressure referred to, especially where that difference is large, and it is desired to make the determination with great accuracy. The formula given enables this to be done.

Professor Thurston's interesting review of the report makes reference to two points which call for notice. One of these is the recommendation of the Committee to use Porter's Steam Tables for making the necessary computations. In explanation of the course which the Committee has taken in this matter, I may say that they were guided mainly by the previous action of the Boiler Test Committee. The Society, by accepting the report of that committee, has already endorsed these tables, and we could not do otherwise than concur. The professor remarks that the Committee have hardly justified their recommendation to use the plunger-displacement method of determining the work done. He inclines to what the Committee have termed the "indirect modes" of measurement, by the use of the weir, tube, or nozzle. Mr. Freeman's criticism is in the same line, though of more direct and forcible character, and they may both be taken up together. The Committee are fully aware that the method proposed is not entirely free from objections, and they have frankly expressed this feeling in the report. They do not claim that the "slip" of the pump can be fully determined by simply measuring the leakage in the manner proposed. Leakage is only one of three elements which go to make up slip, the other two being imperfect filling, due to the presence of air, and "back-flow" through the valves during the act of closing. One of these it is proposed to overcome by allowing no air to be "snifted in" during the trial. The air which finds its way through the stuffing-boxes and unsound pipe connections, and that originally suspended in the water, are believed to be so trifling that for practical purposes they may be disregarded. The slip may thus be narrowed down to two components, namely, leakage and back-flow. Back-flow does not produce such serious results as Mr. Freeman is led to suppose. This action occurs only when the plunger is at the end of the stroke. To be more exact, it does not occur until after the plunger has come to a dead stop, and has commenced the return stroke. Under these circumstances, the forward motion of water through the valves, just previous to the time when back-flow begins, has reached a very low velocity, and the valves are lifted from their seats a correspondingly small distance. The calculation whereby Mr. Freeman obtains a loss of 4 per cent. in the instance which he cites, requires a lift of the valves amounting to about three-quarters of an inch. It is doubtful if in reality the valves are lifted to anything like this extent, even during the middle portion of the stroke, when the water is passing through at the greatest velocity. Certainly the lift would be reduced to a much smaller distance at the time when the slow motion is reached at the end of the stroke. I should be quite surprised if it exceeded one-eighth of an inch at this time, and it is not improbable that it would be so small as to be almost imperceptible. One of the members of the committee, Mr. Nagle, made experiments on the lift of a double-beat pump valve, which he designed and used on the Nagle high-service pumping engine, Providence, R. I., by attaching a rod to the center of the valve, so arranged as to record its movements upon a steam-engine indicator outside, the rod acting through a freely working stuffing-box. Copies of diagrams which he obtained are given in a paper which was read at the Erie meeting, in May, 1889, entitled "Cornish or Double-beat Pump Valves."* Mr. Nagle's conclusions were that "the lift of a valve is exactly proportioned to the velocity of the plunger." It is believed, from these considerations, that the total amount of back-flow, with valves of good construction, will never exceed 1 per cent. of the quantity of water actually pumped, and this is comparatively insignificant.

Leakage, then, appears to be the most important factor in the production of slip, excepting, of course, that occurring in engines with outside packed plungers. The loss from leakage is liable to be considerable, for it must be remembered that in many engines the plunger consists of a solid plug, without packing other than a series of grooves, and a certain amount of leakage must be allowed in order that the plunger may be loose enough to work without undue friction. This is especially true when the pump is one which has been in service for a considerable period of time, and the plunger has become reduced in size by wear or abrasion of the surfaces. Though the determination of the

* Vol. X., page 527, *Transactions A. S. M. E.*, 1889

amount of leakage does not furnish the full measurement of the slip, as thus pointed out, the Committee believe that for the practical purposes of the standard method it meets all necessary requirements.

Mr. Freeman objects to the leakage test because the leakage may be different at full speed from that at rest, and furthermore because the leakage of the valves may be greater at the time of the duty trial, through some unfortunate obstruction under the valves, than it is on the leakage test. It is true that leakage in motion and leakage at rest may be different. But this is somewhat problematical, and if there is a difference it must be of small account. It is true also that obstructions which lodge under the valves would increase leakage, but in this matter it may be said that we do not want the results of a duty trial to be made dependent upon the accidental presence of such obstructions, or even upon the presence of imperfect valves. If such defects occurred in a trial where the water was accurately measured by weir or otherwise, we should desire to make some kind of allowance for the imperfection, so as to give the engine a fair chance. In measuring leakage in the manner proposed, such obstructions or imperfect valves would be first of all removed, and the conditions made to apply in like manner to unimpeded valve action.

The general principles which governed the Committee in deciding that the plunger-displacement method of determining the work done was the best, are as follows: The duty of a pump is a quantity computed from the number of *foot pounds of work* expended in moving the water. I think that all engineers will agree with me that the indicator diagram taken from the water end of the pump furnishes a true expression of the work directly expended in pumping. The diagram tells nothing regarding slip, and little regarding the actual quantity of water delivered, but it furnishes data for calculating precisely the number of *foot pounds of work* which the plunger imparts in moving the body of water through the pump. This is not only an exact measurement of the work done, but it is a scientific one, and the truth of this position cannot be assailed. Passing a step farther, the work found by measurement of the pump card, less that due to overcoming the friction of the valves and passages in the pump, is an equally true representation of what may be called the "net work" done. This last is the measurement of work upon which the proposed standard is based. It has the advantage of being unaffected by leakage or back-flow, unless these be enormous, and the effect of air is overcome by having no air snifted in. It is accurate and scientific, and no guesswork needs to be resorted to in making the computation. It is applicable to all classes of pumps and all kinds of service. The method can be followed by any intelligent person who is at all familiar with the work of making a pumping-engine test. It has, moreover, been largely used by pump manufacturers and experts, and there is nothing which stands in the way of its universal application. The data to be determined are no different from those determined in the most elaborate trials which embrace the measurement of water by weir or other means, and the results of this determination are given in all the duty trials which have any claim to completeness. Surely these reasons are amply sufficient to justify the course which the Committee have taken in recommending that the duty be figured from the plunger-displacement, and the reading of the suction and discharge gauges after the manner detailed in the report.

It is hoped that the action of the Committee in making this decision will not be misunderstood. While making the duty result dependent upon plunger-displacement for the purposes of a standard of comparison of different pumps, and while making a determination, so far as possible, of all visible losses due to what is believed to be the main factor in the production of slip, that is, leakage, the Committee would also determine, wherever it is possible to do so, the actual quantity of water delivered, by the use of the weir or other devices for this purpose. It does not, however, seem wise to the Committee to require that such measurement be peremptorily taken, and the figures of duty be based on the results of such measurements. It may be stated as a probability amounting almost to a certainty that if the standard were to be based either on weir measurement or that by means of the Venturi tube or by nozzles, it would be only in case of comparatively few trials that the recommendations of the Committee would be carried out. It is well known that so far as weir tests are concerned they are inapplicable without considerable expense to many cases of practice. The only cases to my knowledge where a weir has been applied in the tests which have been made in the past, have been where the water was discharged into a reservoir, the we being located at the end of the pipe entering the reservoir. is needless to say, therefore, that a weir standard would only used on occasional tests. The use of the Venturi tube requir that such a tube be placed in the discharge main, of which will form a part. If this has not already been done in the co struction of the works, its use would require a certain amou of reconstruction. These would make the Venturi method limited application. The use of the Siamese nozzle, which M Freeman proposes, is something which appears to be high commendable; but this is as yet untried so far as pertains to t actual work of measuring the water discharged by a pumpi engine, and this remark is equally applicable to the use of t Venturi tube. It would be idle for any committee to reco mend either of these systems for a standard in what may called their present undeveloped state, however well they m eventually be found to subserve the purpose. Mr. Freema remarks concerning the Siamese nozzle, which he has design for making tests of this kind, are exceedingly full, and this, connection with the account of his experiments on nozzles, fu nish all that may be needed in the way of directions for thos who wish to employ the method.

The Committee believe that the proposed methods furnish standard which accomplishes all that is desired for it, so far a it can be accomplished in the present state of the art.

The Committee have solicited the discussion of several ge tlemen who are not members of the Society, one of whom, M John E. Codman of the Philadelphia Water Works, has kind submitted his views. These are incorporated with the discu sion of the members.

The following discussion was presented verbally at the Richmond meetin November, 1890, and is appended to the contributed criticism.

Prof. J. E. Denton.—I consider this report one of the fine pieces of committee work which has ever been done, not only this society but in any society, and I want to express my admir tion of it in the strongest terms. It first strikes at the princip evil in current practice regarding methods of testing the pe formance of pumps. It brings out with greater emphasis th fact that we do not test the engine by determining the coal co sumption, but we test the engine mixed up with what the boil does, and that is mixed up with what the coal may do, which is a very variable factor.

The appreciation of this is no new thing, of course. Watt expressed himself very emphatically about the necessity of measuring the steam consumption of the engine, and the fact has been recognized all through the history of steam-engines. The Committee go to very high ground in this matter. They not only recommend that the water consumption per horse-power be the basis, but they say the thermal units used by the engine shall be the basis; and we know that of late years it has been prominently brought out, and in no better place than in the little book of the chairman of this committee, that water consumption does not express the performance of an engine, but that the thermal units which it uses must be the true basis. With a non-condensing engine using 24 lbs. of water and heating the feed water to 212°, to state the water consumption per horse-power does not furnish a basis for comparison of the result with the performance of a condensing engine with the feed water at 100°. Therefore it has become necessary to state the performance of engines in heat units, and this report takes that ground.

It will never be possible, of course, to divorce the coal consumption from the steam consumption entirely. While the expert naturally wants to eliminate all ambiguity, and therefore wants this distinction made between coal and water consumption, the man buying the engine wants to have the whole plant guaranteed, so that the dollars which he pays out for coal shall give him a certain result in return; and furthermore the makers of pumping engines all have their particular boilers, and they prefer to put in engine and boilers and use the total performance of the plant as a basis of advertisement in many cases. Consequently I don't believe it will ever be possible to divorce the coal from the steam consumption. For that reason I think both should be given; but the only practical scientific basis is certainly the thermal units expended by the engine per unit of work, and this report brings that out in an admirable way.

The report gives directions for testing the performance of an engine, which show an intimate acquaintance with all the details involved in a complete test, and while mentioning every possible method which is reasonable for measuring the delivery of water it avoids committing itself to those methods which can be conceived of as the very ultimatum of perfection, but which are impracticable; as for instance determining absolutely the delivery of a pumping engine through artificial orifices.

Mr. Freeman's device, resulting from experiments of elaborateness which entitles him to the greatest credit as an experimenter, is an instrument from which much may be anticipated in the future, but it certainly is not an available substitute for the method recommended in the report, which is to make the pump its own meter. I think that is the best basis, considering practical possibilities. We know that a pump which is ready for testing is generally fairly tight. If it is not, and the fact is discovered, why, it is made tight before the test is carried out. Therefore, if you test the leakage at rest you get so near a knowledge of the condition of the pump that if the leakage is reduced to a fair minimum, as the maker will insist on doing before he lets the test go on, the plunger displacement, corrected by the leakage determination, forms an entirely satisfactory basis of measurement.

I want to say one word about Porter's steam tables. I should be sorry to see any other steam tables become standard in this country. Most of us know that Mr. Charles T. Porter has the soul of an artist. He has devoted his life to trying to do things with the utmost refinement, and in his early days it was part of his ambition to make these steam tables as accurate as computation could make them. I do not think the accuracy of the tables can be improved. Some years ago we had occasion to check the tables in comparison with Regnault's experiments, and were much impressed with the exhaustiveness of Mr. Porter's work. I hope Prof. Jacobus, who made the comparison, will exhibit some of its results as a contribution to this discussion.

Prof. D. S. Jacobus.—The report of the committee is an admirable piece of work, and the following is not intended as a criticism, but is presented in order to add a few thoughts which have been suggested in making trials of pumping engines.

In the case of many pumping engines a continuous log is preserved of the weight of coal and the water pressures in order to show the performance for long intervals of time. It would be unwise to introduce any conditions which will not allow results given by the above method to be obtained and quoted in some standard way. If, however, the report is adopted as it now stands, together with all the recommendations which follow in the discussion, it would be impossible to do so.

As the above method of testing over long intervals will necessarily be followed in many cases, it would be well to include the duty calculated per 100 lbs. of coal, as well as per 1,000,000 heat units. Again, the duty calculated from the coal is useful to parties having the supervision of pumping engines, as it represents the economy of the entire plant including the boilers.

It is evident, in regard to the methods suggested in the discussion for measuring the water delivered from the pump, that any method which prevents the pump being worked under the conditions which ordinarily exist, will be objectionable. In the case of a pump discharging water directly into the mains, to employ the Siamese nozzle, as recommended in the discussion, would necessitate shutting off the water from the mains, and thus wasting all the work done by the pump. If this is done there would be a tendency to make short tests in order to reduce the expense of the same, and such tests are often misleading.

If the duty be calculated from the piston displacement it would be well to specify that indicator cards be taken from the water-cylinders, at regular intervals during the test, in order to show that the valves operate uniformly. It is recommended in the report that this be done, but more importance should be given to it.

In the case of a pump tested about a week ago, one of the water-valves gradually became inefficient, and the power to drive the pump diminished from 145 to 125 H. P. If therefore the pump had been tested in its final condition the duty calculated from the piston displacement would have been much too great.

In the case of a duty trial which extends over several days, the indicator will show the moment when any irregularity occurs in the water-cylinders, and the test may be ended, whereas if tests are made for leakage before and after, and leakage is found to exist at the end of the test, it throws a doubt over the accuracy of all the results which have been obtained.

In connection with the discussion in regard to the relative merits of the various steam tables now in use, I take the liberty of presenting the matter contained in Table I. that leads to the following conclusions.

First.--The steam tables in general use give Regnault's values for the pressures corresponding to given temperatures, and for

TABLE I.

COMPARISION OF QUANTITIES GIVEN IN VARIOUS TABLES OF THE PROPERTIES OF SATURATED STEAM.

Temperature of Ebullition by Air Thermometer.		Absolute Pressure.						Latent Heat in B. T. U. per Pound.					Total Heat above 32° Fahr. in B. T. U. per pound.					Volume of One Pound in Cubic Feet.						
Degrees Centigrade.	Degrees Fahrenheit.	* Regnault.		† Porter.	‡ Clark.	‖ Dery.	¶¶ Peabody.	¶ Regnault.	† Porter.	‡ Clark.	‖ Dery.	¶¶ Peabody.	¶ Regnault.	† Porter.	‡ Clark.	‖ Dery.	¶¶ Peabody.	** Rankine.	†† Clausius.	‡‡ Fairbairn.	‖ Dery.	† Porter.	‡ Clark.	¶¶ Peabody.
		Atmos.	lbs. p. sq. in.	lbs. p. sq. in.	lbs. p. sq. in.	lbs. p. sq. in.	lbs. p. sq. in.																	
100	212	1	14.7	14.7	14.7	14.7	14.7	965.7	965.7	965.2	965.7	965.8	1145.6	1146.6	1146.1	1146.6	1146.6	26.36	26.36	26.29	26.38	26.34	26.36	26.60
130	266	2.671	39.26	39.26	39.13	39.26	39.26	927.2	927.3	927.0	927.3	927.9	1163.0	1163.1	1162.6	1163.1	1163.1	10.48	10.46	10.24	10.47	10.24	10.48	10.55
160	320	6.120	89.96	89.95	89.76	89.95	89.95	888.5	888.4	888.6	888.4	889.5	1179.5	1179.5	1179.0	1179.5	1179.5	4.82	4.80	4.72	4.81	4.72	4.80	4.86
190	374	12.425	182.65	182.59	182.30	182.60	182.27	848.9	849.0	850.5	849.0	849.5	1195.9	1196.0	1195.4	1196.1	1196.0	2.48	2.47	2.54	2.48	2.54	2.46	2.50

* Experiences par M. V. Regnault, vol. i, p. 624.
† The Richards Steam Engine Indicator, by C. T. Porter, pp. 66 to 74.
‡ Manual of Rules, Tables and Data for Mechanical Engineers, by D. K. Clark, pp. 387 to 390.
‖ Transactions of the American Society of Mechanical Engineers, vol. xi, pp. 101 to 105.
¶ Experiences par M. V. Regnault, vol. 1, p. 748.
** Rankine's Steam Engine, pp. 592 to 596.
†† Article by Thomas Rowe Edmunds, published in The Philosophical Magazine and Journal of Science, vol. xxx, fourth series, 1865, p. 10.
‡‡ Mills and Millwork, Fairbairn, pp. 212 to 214.
¶¶ Tables of the Properties of Saturated Steam and Other Vapors, by Cecil H. Peabody, pp. 26 to 30.

the total and latent heats of evaporation with all the accuracy necessary for practical work.

Second.—The tables differ slightly in the values of the density of the vapor, which in some cases are derived from the experiments of Fairbairn and Tate, and in others by means of the thermo-dynamic relations that exist between the temperatures, and corresponding pressures and latent heats determined by Regnault.

Third.—The greatest difference between the density of steam at ordinary pressures derived by Fairbairn's experimental formulæ and by thermo-dynamic laws is about 3%.

Fourth.—The results obtained by thermo-dynamic laws are probably the most reliable.

Fifth.—In special cases where scientific accuracy is required it would be most logical to make use of the densities derived by thermo-dynamic laws, as given in the tables of Rankine and Clausius, or as transformed by M. V. Dwelshauvers-Dery, instead of the experimental densities given in Porter's tables. For all practical purposes, however, the experimental densities are sufficiently accurate.

In Mr. Porter's table which has been included in the report on the Duty Trials of Pumping Engines, the experimental densities are employed for pressures above that of the atmosphere, and the theoretical densities for pressures lower than the atmosphere. The difference in the values of the density as given in Porter's tables and as determined by the thermo-dynamic formulæ, is about 2% for pressures in the neighborhood of 60 lbs. absolute per square inch. Above this point the difference decreases until the absolute pressure is about 125 lbs. per square inch, at which point the two values are the same. Above 125 lbs. per square inch the values differ in a reverse way from that which occurs between the pressure of the atmosphere and 125 lbs. At 210 lbs. absolute pressure, which is the highest figure given in the table, the difference is about 4%. Mr. Porter's transformations of Regnault's work are extremely accurate, and this portion of his table is more elaborate than is to be found in any other table now published. D. K. Clark's table contains slight discrepancies in the transformation of Regnault's work, which, however, are too small to affect any practical problem.

The experiments of Fairbairn and Tate cover a range of from only about 3 to 56 lbs. per square inch absolute pressure, and it is illogical to employ the empirical formula proposed Mr. Fairbairn that fits this range for much higher pressur To obtain the figures given in Mr. Porter's tables, Fairbair formula has to be employed for steam at pressures three or fc times as great as for those at which it is known to give resu that agree with experiments.

This, however, is not an argument against the accuracy of t results given in Mr. Porter's tables for pressures up to 125 l per square inch, because at this point the experimental agr with the thermo-dynamic equation. Above 125 lbs., however, t density by Fairbairn's formula varies in the opposite way fr which it did at the lower pressures, and at very high pressu the results are considerably in error.

No comparison is herein presented for pressures below tl of the atmosphere, because the values over this range are t same in all the tables; the pressures and the total and late heats being those derived from Regnault's experiments, and t volumes those given by thermo-dynamic relations.

Rankine compared the liability to error of the thermo-dynan determination of the density, as derived from the latent he and pressures observed by Regnault, and of the experimen densities determined by Fairbairn and Tate, and concluded tl the difference between the results obtained by the two metho was greater than the probable error of the experimental quan ties involved in the problem. He advanced the supposition tl there may have been a difference in the molecular condition of t steam in the two sets of experiments, as that on which Regnaul experiments were made was in motion, whereas that in Fa bairn's and Tate's experiments was at rest. The following tal was prepared by Rankine in order to compare the resu given by the two methods.*

* *Miscellaneous Scientific Papers*, by Professor Rankine, p. 423.

COMPARISON OF THE THEORY WITH THE EXPERIMENTS OF MESSRS. FAIRBAIRN AND TATE.

No. of Experiment.	Temperature. Fahrenheit.	Volume of One Pound of Steam in Cubic Feet.		Difference.	Difference ÷ Exper. Vol.
		By Theory.	By Experiment.		
1	136°.77	132.20	132.60	−0.40	$-\frac{1}{330}$
2	155.33	85.10	85.44	−0.34	$-\frac{1}{250}$
3	159.36	77.64	78.86	−1.22	$-\frac{1}{65}$
4	170.92	60.16	59.62	+0.54	$+\frac{1}{110}$
5	171.48	59.43	59.51	−0.08	$-\frac{1}{744}$
6	174.92	55.18	55.07	+0.11	$+\frac{1}{501}$
7	182.30	47.28	48.87	−1.59	$-\frac{1}{30}$
8	188.30	41.81	42.03	−0.22	$-\frac{1}{191}$
9	198.78	33.94	34.43	−0.49	$-\frac{1}{70}$
1′	242.90	15.61	15.23	+0.12	$+\frac{1}{125}$
2′	244.82	14.77	14.55	+0.22	$+\frac{1}{65}$
3′	245.22	14.67	14.30	+0.37	$+\frac{1}{39}$
4′	255.50	12.39	12.17	+0.22	$+\frac{1}{55}$
5′	263.14	10.96	10.40	+0.56	$+\frac{1}{19}$
6′	267.21	10.29	10.18	+0.11	$+\frac{1}{93}$
7′	269.20	9.977	9.703	+0.274	$+\frac{1}{35}$
8′	274.76	9.158	9.361	−0.203	$-\frac{1}{47}$
9′	273.30	9.367	8.702	+0.665	$+\frac{1}{13}$
10′	279.42	8.539	8.249	+0.290	$+\frac{1}{28}$
11′	282.58	8.145	7.964	+0.181	$+\frac{1}{44}$
12′	287.25	7.603	7.340	+0.263	$+\frac{1}{28}$
13′	292.53	7.041	6.938	+0.103	$+\frac{1}{66}$
14′	288.25	7.494	7.201	+0.293	$+\frac{1}{24}$

Mr. F. M. Wheeler.—Professor Denton speaks of the importance of considering the testing of a pumping engine on the basis of thermal units. I think that is a very important point. A friend of mine is very enthusiastic about testing steam-engines by the use of the *surface* condenser, and which he considers the simplest and most convenient mode of doing it, but he quite overlooked the all-important point of temperatures. Considerable has been said in this discussion about the "metering of water" by the plunger displacement. As a matter of fact, a large percentage of the pumping engines used to-day are direct-acting and in this type, of course, the momentum of the plungers, pistons, etc., varies with the speed—in other words, you have variable length of strokes. Now, it is a very poor meter which has a variable stroke. This variableness of stroke should therefore call for frequent indicator cards, taken from the water end of the pumping engine, and some ingenious contrivance for a continuous stroke indicator. In fact, the importance of a stroke indicator comes in there very strongly, and, as it is an instrument which can be very easily attached, it should never be omitted in making tests. I always use a pencil attached to the cross-head of the piston rods, which, travelling over a sheet of paper (moved frequently by a person especially detailed for the purpose), gives the length of stroke correctly; the more frequently the paper is moved, the better—say as often as every two or three minutes. This idea could be elaborated further, and a continuous line could be drawn by means of clockwork or otherwise suitable mechanism. Then you could secure a record of each and every stroke of the pumping engine during the whole test, and thus show the variableness of the strokes. Of course, the use of the weir is the best and safest manner of making pumping-engine tests when practicable, but the committee and those discussing this paper have said so much about the displacement (meter) system, we ought not to overlook this matter of stroke irregularity. I have seen a five-million-gallon pumping engine, having a nominal stroke of 36 inches, show a variation of nearly two inches in its stroke in as many minutes.

Another point: One of the gentlemen who discussed this paper spoke of "negative slip." This varies very much with the condition under which the pumping engine is working. I would like to speak of a case on one of the naval ships (U. S. S. *Tennessee*), where the amount of water discharged by the circulating pump was nearly 7% in excess of the actual displacement of the water piston. In other words, the pump delivered more water than its theoretical capacity. This apparent paradox was due to the very light load, and the fact that the water flowed to the pumping engine under considerable head.* This same effect is also very frequent in single-acting pumps (where the flow of water is always in one direction) under certain favorable conditions.

Mr. Wm. M. Barr.—There does not appear to be any good reason why one million thermal units should not be substituted for the present practice of basing pumping-engine performance on the number of million of foot pounds for each one hundred

* See *Franklin Institute Journal* of December, 1874.—Extract from paper by Messrs. Geo. P. Hunt, P. A. Eng. U. S. N., and Theron Skeel, C. E., entitled *The Methods of Testing Steam Engines.*

pounds of coal. The method of arriving at the number of thermal units has been quite satisfactorily shown; but I wish to say a few words in regard to plunger displacement, as I think that much more convenient and quite as accurate as the weir method, and probably more so than by the use of a large nozzle, besides which, it is not always practicable, except at great inconvenience, to employ either of these.

If a pumping engine is fitted with a crank and fly-wheel, the displacement can be very easily determined—it is simply the area of the plunger multiplied into the stroke. The usual method of fitting large pumping engines is to have a cast-iron plunger working through a solid gun-metal ring. Sometimes these rings are grooved, to prevent a free flow of water from one side of the pump to the other, and at other times they are left plain; the latter is the usual method of fitting. If the plunger and rings are new, the amount of slip is very small; I do not know how much, but in the case of an engine of, say, five millions capacity I doubt whether it would be over 2% to 5%. As these become worn, the slip will, of course, be greater, but, whatever the amount may be, it is a constant leak, and must be accounted for. If the pump be fitted with a lining and a packed piston, the leak amounts, practically, to nothing, and during a duty trial, in which the pistons would be freshly packed, I think it would be safe to assume that they were tight, and that no allowance need be made for slip.

In the case of a direct-acting pumping engine, in which it is expected that the stroke will not be constant, the measuring of the displacement is not so easy. It is the practice to fit each end of each steam cylinder in engines of this class with dash relief valves; these valves can be so carefully adjusted that the piston can be made to touch each end of the cylinder, without doing harm to the engine. This, you will see, requires constant attention on part of the engineer, and it would not be safe to allow the engine to take care of itself, with the piston coming in contact with the heads, even though that contact be very delicate, or simply enough to know that the piston touches. By making a gauge showing the contact stroke, with a pointer attached to the crosshead, the engineer can see at all times what proportion of the whole stroke is traversed by the piston. For the purpose of a duty trial this method of controlling the length of stroke is easily had, and has been very satisfactorily worked out in practice. The only objection is, that when the dash relief valves are closed, or partially so, after the duty test is over, the pumping engine will be delivering a less quantity of water than during the duty trial; and just in proportion as the length of stroke decreases, the clearance in the steam cylinders becomes greater, with its consequent reduction in economy.

I have never had any experience with negative slip, and therefore feel a little skeptical in regard to its occurring in quantities sufficiently large to affect a duty trial.

Now, in regard to the valves, the best practice is to use a large number of small valves, rather than a less number of large valves. If these valves be fitted with a spiral spring, so as to insure their prompt return, there will be very little danger of loss or leakage at the moment the piston or plunger changes direction of motion. Should there be floating matter in the water, which would get under one edge of the valve, there would, of course, be leakage at that point, and that one only. This is not a serious matter, because the probability is that at the next movement of the plunger or piston this obstruction would release itself, and it would pass on through into the delivery. Unless this obstruction becomes permanent, and the valve is thereby kept off the seat through a long interval of time, this must then be treated as any other accident; but I think the loss in the amount of water delivered through such an occurrence is, on the whole, very small, and is hardly worth taking into account.

In regard to the pumping engine supplying the boilers with feed-water during the duty trial, I think that is all right, although very few pumping engines are now fitted with boiler feed-pumps attached to the main engine, as it is much more convenient to have a separate steam-pump; and as this is the method of operating nearly all water-works plants, the duty trial ought to include that method of supplying the boilers with water. It will be understood that this is not as economical an arrangement as the other, but it is the one generally adopted.

Mr. Jno. T. Hawkins.—I would like to say one word. While endorsing the report as a most admirable paper in every way, and as the only one which brings us to the point, which is the ultimate object—where we have not been heretofore, at all events—of determining the gallons of water which we get raised to a given height for a dollar, I think it is open to one little objection. The object here, of course, or in any test of a pumping-plant, is to ascertain its commercial efficiency rather than its efficiency under experimental conditions which cannot be realized in practice, and this method very admirably differentiates the engine from the boiler, and establishes a test for each independently. But I think it makes an error in shortening the usual period taken for such tests, so far as the boiler element of it is concerned. Old engineers who have had a good deal of experience in the management of boilers know that boilers vary very much in several points; a principal one of which is the condition into which the fire will get within a given number of hours, and as a general thing, the fires will not get into that condition in ten hours, which would give us a correct index of the boiler's commercial efficiency, such as when running along for weeks and months as pumping plants are designed to do, and I think that the plan recommended makes that one mistake. It would be more in the direction of the rest of the plan if the time were made greater rather than less than twenty-four hours. I do not think that with the very best management possible the fires in any boiler can be gotten into such running condition as to represent the average commercial conditions in ten hours, although the engine element of the plant will get to it in a much less time than ten hours. Boiler fires, flues, etc., arrive at a condition after the first twelve or fifteen or twenty hours which more nearly represents practical working than they can get into before that time. I think it is a mistake to reduce the time below twenty-four hours. It ought to be extended to forty-eight hours rather than be reduced to twenty-four, so far as the boiler is concerned.

Mr. Geo. H. Barrus (Chairman of Committee).—In closing the discussion a few words should be added in reply to the questions raised by Messrs. Denton, Jacobus, Wheeler, Barr, and Hawkins.

A remark made by Professor Denton, as also a statement given in Professor Thurston's discussion, makes it appear that in computing the work done by the pumps, as proposed, a correction is applied for the loss found on the leakage trial. This is not the method adopted in the report. The quantity of leakage which is determined by the test is stated in connection with the duty result, as a matter of record and information, but no correction is applied for it. The duty given is that computed from the observed plunger displacement, and it is not affected in the calculations by the leakage, whatever quantity this may reach.

To carry out the suggestion of Professor Jacobus, line No. 45

has been added to the tabular form in the revised copy of the report, giving the work done by 100 lbs. of coal, or the duty calculated from the coal basis heretofore in use.

Professor Jacobus' reference to the continuous records, which well-managed water boards keep, of the regular work done by their pumping engines, draws attention to the question of the adaptability of the new unit to ordinary water-works practice. It is not expected that the introduction of the heat unit basis for duty trials will supplant the water-works custom of expressing the duty on the actual coal burned. The boiler and engine cannot in every-day work be separated in the manner proposed by the new method of test, and were it possible to do so, such a course would not be desirable. The new method, however, suggests a line of practice for pumping stations which will be of great service in standardizing records taken in the future, so that those obtained from one engine may be properly compared with those obtained from another. To carry out this idea the calculation of duty, in addition to being based on the quantity of coal consumed, would be based on one million heat units; and the calculation would be made from data, which, for all practical purposes, could be satisfactorily obtained under the conditions of ordinary work. Such data would be found by employing a water meter for the continuous measurement of the main supply of feed-water, tested from time to time to determine its accuracy, and using a thermometer fixed permanently in the feed-pipe for determining its temperature. With the data thus obtained the total heat imparted to the largest share, if not to the whole, of the water would be computed. If there were an auxiliary supply, like that derived from a jacket-pump, it is presumed that this would be practically constant, and if its quantity and temperature were once determined, in the manner pointed out in the report, a constant correction could be applied for the heat imparted to this water, without the necessity of obtaining it by a continuous record. The water lost by blowing off the boilers, and that used for supplying steam for heating purposes, or other work than that of driving the engine under consideration, could be estimated, and a suitable correction applied, so as to obtain a result which would refer to the engine independent of all other work. With a little study on the part of the engineer of the pumping station, or of those having charge, continuous records, on the heat-unit basis, could in this manner be-obtained, not only with little trouble, but with a degree of accuracy which would be sufficient for all practical purposes. A comparison of such records from engines in different localities, although, perhaps, not scientifically accurate, would show the relative economy of engines in practical work with greater reliability than records which are now given on the basis of actual coal consumed. This adaptation of the new method of conducting duty trials to the practical work of pumping engines, will, it seems to me, prove of great value.

Professor Jacobus' remarks upon the relative merits of the various steam tables now in use, and the tabular summary which he gives, showing the difference in the quantities given by the various authorities on the subject, form an exceedingly valuable contribution. It will be seen from this table that, so far as the main subject is concerned, that is, the determination of the number of heat units consumed by an engine, no practical difference would be produced in the results, whether one authority were used or another.

Mr. Wheeler voices the sentiments expressed in the report, that some contrivance be used in determining the length of the stroke, in direct-acting engines, by a continuous recording device. It is hoped that he will elaborate his ideas, and design some simple apparatus which can be used for this purpose.

Mr. Barr brings out many interesting points relating to direct-acting pumping engines, and adds much to the general value of the discussion.

Mr. Hawkins objects to making the duration of the test, so far as it pertains to the work of the boiler, less than twenty-four hours, on the ground that a shorter test will not show its commercial efficiency. If the object of the duty trial were simply to obtain the commercial efficiency of the plant, the duration of the trial should be at least a week, and the conditions under which the engine and boilers are run should be as nearly as possible the same as the intended working conditions. On such a trial it is probable that the engine would seldom be run continuously for a week's time, or even for twenty-four hours; but rather it would be run a certain number of hours per day, and during the time when the engine is idle the fires in the boilers would be banked. The results of the boiler test would then be affected to a greater or less extent by the conditions, as to the number of hours of running, which happened to exist. However desirable such a trial might be from a commercial point of view, it would not, as I believe, secure the object for which the greater number of duty tests are made. That object is to determine what the engine and boilers will do when working under the best conditions which in practice can be realized. For these reasons I think that the duration of the trial, whether of the engine or of the boiler, need not be extended to a longer time than is sufficient to determine the necessary data with a proper degree of accuracy. A duration of twelve hours, according to my experience in making boiler tests, furnishes an ample amount of time in which to get the data, with all the exactness needed. Another reason for making the test as short as possible, as stated in the report, is to reduce the number of hours to such a point that the whole work of the duty trial, from beginning to end, including the boiler test, can be under the eye of the same expert without undue physical exertion.

PAPERS

OF THE

PROVIDENCE MEETING

(XXIIId)

CCCCXXXVIII.

PROCEEDINGS

OF THE

PROVIDENCE MEETING

(XXIIId)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS,

June 16th to 19th, 1891.

LOCAL COMMITTEE OF ARRANGEMENTS:—Henry A. Du Villard, Gardner C. Anthony, Geo. R. Babbitt, Robert J. Gilmore, E. H. Parks, George H. Smith, Scott A. Smith, George E. Whitehead.

FIRST DAY JUNE 16TH.

THE Sessions of the Providence Meeting were held in St. John's Hall of the Masonic Building, Dorrance and Eddy Streets. The Headquarters of the Convention were in the Narragansett Hotel. adjoining the office in the main corridor. The Secretary's Register in Headquarters showed the following members in attendance during the Convention:

Alden, Geo. I.	Worcester, Mass.
Allen, Jno. F.	New York City.
Almond, Thos. R.	Brooklyn, N. Y.
Ames, W. L.	Terre Haute, Ind.
Anthony, Gardner C.	Providence, R. I.
Armington, Pardon	Providence, R. I.
Archer, E. R.	Richmond, Va.
Ashworth, Daniel	Pittsburgh, Pa.
Babbitt, Geo. R.	Providence, R. I.
Bailey, Chas. L.	Washington, D. C.
Baldwin, S. W. (*Vice-President*)	New York City.
Ball, Frank H. (*Manager*)	Elizabeth, N. J.
Bang, H. A.	New York City.
Barnaby, C. W.	Meadville, Pa.
Bardwell, A. F.	Stamford, Conn.
Barnes, Abel T.	Jamaica Plain, Mass.

Barnum, Geo. L. ....New Haven, Conn.
Barr, John H. ....Minneapolis, Minn.
Barrus, Geo. H. ....Boston, Mass.
Baugh, S. A ....Detroit, Mich.
Beach, Giles ....Gloversville, N. Y.
Beach, Chas. S. ....Bennington, Vt.
Bigelow, Frank L. ....New Haven, Conn.
Benham, E. ....Providence, R. I.
Binsse, Henry L. ....Newark, N. J.
Bird, W. W. ....Cambridgeport, Mass.
Bixby, E. M ....Boston, Mass.
Blair, H. P. ....Rochester, N. Y.
Bond, Geo. M. (*Manager*) ....Hartford, Conn.
Borden, T. J. ....Fall River, Mass.
Boyd, Jno. T. ....Erie, Pa.
Britton, J. W. ....Cleveland, O.
Brooks, Morgan ....St. Paul, Minn.
Brown, A. T. ....Syracuse, N. Y.
Brown, C. H. ....Fitchburg, Mass.
Bulkley, H. W ....New York City.
Bullock, Edwin R. ....Pawtucket, R. I.
Bushnell, F. N. ....Providence, R. I.
Butterworth, Jas. ....Philadelphia, Pa.
Caldwell, A. J ....New York City.
Canning, W. P. ....Lowell, Mass.
Capen, T. W. ....Chicago, Ill.
Cary, A. A. ....Bridgeport, Conn.
Charnock, J. M ....Boston, Mass.
Cheney, W. L ....Meriden, Conn.
Church, E. D ....Brooklyn, N. Y.
Church, W. L. ....Boston, Mass.
Churchill, Thos. L. ....Boston, Mass.
Clark, W. L ....Philadelphia, Pa.
Clarke, S. J ....New York City.
Coleman, J. A. ....Providence, R. I.
Conrad, H. V ....North Tarrytown, N. Y.
Cooper, Henry R. ....Syracuse, N. Y.
Corliss, Wm. ....Syracuse, N. Y.
Cottrell, C. B ....New York City.
Creelman, Wm. J. ....Rochester, N. Y.
Cremer, Jas. M. ....Brooklyn, N. Y.
Cullingworth, Geo. R. ....New York City.
Dallett, W. P. ....Philadelphia, Pa.
Daniels, Fredk. H. ....Worcester, Mass.
Darling, Edwin ....Pawtucket, R. I.
Dashiell, Benj. J., Jr. ....Baltimore, Md.
Davis, E. F. C. ....Richmond, Va.
Day, F. M ....Hopedale, Mass.
Dean, F. W. ....Boston, Mass.
Deane, Chas. P. ....Holyoke, Mass.
Denton, Jas E. (*Manager*) ....Hoboken, N. J.

Derbyshire, W. A. . . . . .Philadelphia, Pa.
Dinkel, Geo., Jr. . . . . .Jersey City, N. J.
Doane, W. H. . . . . .Cincinnati, O.
Doran, W. S . . . . .New York City.
Drown, F. E. . . . . .Pawtucket, R. I.
Drysdale, W. A. . . . . .Philadelphia, Pa.
Du Villard, H. A. . . . . Providence, R. I.
Eberhardt, F. L. H. . . . . .Newark, N. J.
Edwards, V. E. . . . . .Worcester, Mass.
Ehlers, Peter . . . . .Albany, N. Y.
Fawcett, Ezra. . . . . .Alliance, O.
Fickinger, P. J. . . . . .Beaver Falls, Pa.
Firestone, Jos. F . . . . .Columbus, O.
Fish, Chas. H. . . . . .Manchester, N. H.
Fletcher, Wm. H . . . . .Hoboken, N. J.
Freeman, John R. . . . . .Boston, Mass.
Gabriel, W. A. . . . . .Elgin, Ill.
Gale, Horace B. . . . . .St. Louis, Mo.
Galloupe, F. E. . . . . .Boston, Mass.
Gantt, H. L . . . . .Philadelphia, Pa.
Gilmore, R. J. . . . . .Providence, R. I.
Gobeille, J. L. . . . . .Cleveland, O.
Godrey, E. S. . . . . .Brooklyn, N. Y.
Goodale, A. M . . . . .Waltham, Mass.
Gordon, F. W. . . . . .Philadelphia, Pa.
Gorton, John C . . . . .Stamford, Conn.
Gould, W. V . . . . .Norwich, Conn.
Granger, W. S. . . . . .Providence, R. I.
Greene, I. C . . . . .Baltimore, Md.
Greenwood, P. F. . . . . .Richmond, Va.
Grimm, P. H . . . . .Glen Cove, N. Y.
Guthrie, Ed. B. . . . . .Buffalo, N. Y.
Hall, A. F . . . . .Boston, Mass.
Halsey, F. A. . . . . .Quebec, Can.
Hammond, G. W. . . . . .Yarmouthville, Me.
Hand, F. L . . . . .Philadelphia, Pa.
Hardie, Robt. . . . . .New York City.
Haskins, Harry S . . . . .Philadelphia, Pa.
Hawkins, Jno. T. . . . . .Taunton, Mass.
Hayward, F. H . . . . .New York City.
Hemenway, F. F. . . . . .New York City.
Henning, G. C. . . . . New York City.
Henthorn, J. T. . . . . .Providence, R. I.
Hershey, M. E. . . . . .Harrisburg, Pa.
Hibbard, Thos . . . . .West Roxbury, Mass.
Hill, Wm. . . . . .Collinsville, Conn.
Holland, Jno. . . . . .Dover, N. H.
Holloway, J. F. (*Past President*). . . . . .New York City.
Howard, C. P. . . . . .Hartford, Conn.
Howe, H. M . . . . .Boston, Mass.
Hunt, C. W. . . . . .New York City.

Hunt, Robt. W. (*President*)..................Chicago, Ill.
Hunter, Geo. E..........................Elgin, Ill.
Hutton, F. R. (*Secretary*)....... ....New York City.
Hyde, C. E ....Bath, Me.
Ide, A. L............ ....Springfield, Ill.
Idell, F. E..................................New York City.
Jacobus, D. S..............................Hoboken, N. J.
Jenks, W. H ....Brookville, Pa.
Johnson, Arthur E.........................Stamford, Conn.
Jones, Ed. H ....Cleveland, O.
Kafer, Jno. C..............................New York City.
Kent, Wm....................................New York City.
King, C. C..................................New York City.
Kirchhoff, Chas............................New York City.
Kneass, S. L............ ....Philadelphia, Pa.
Knickerbacker, Jno ....Troy, N. Y.
Lane, H. M..................................Cincinnati, O.
Lane, J. S...................................Chicago, Ill.
Lewis, Wilfred.............................Philadelphia, Pa.
Lipe, C. E.......................... ....Syracuse, N. Y.
Low, Fred. R...............................New York City.
Lowe, Wm. V.................. ....Fitchburg, Mass.
McBride, Jas ....Brooklyn, N. Y.
McEwen, J. H ....Ridgway, Pa.
McLeod, Howard D........................Pittsburgh, Pa.
Mackinney, W. C....... ....Philadelphia, Pa.
Main, C. T...................... ....Lawrence, Mass.
Manning, Chas. H.........................Manchester, N. H.
Mansfield, A. K...........................Salem, O.
Mason, H. I.................................Cuyahoga Falls, O.
Matheson, W. G ....Amherst, N. S.
May, De Courcy............................Philadelphia, Pa.
Miller, Alex ....Jersey City, N. J.
Miller, Jas. S...............................Erie, Pa.
Miller, T. S.................................New York City.
Miller, E. F.................................Cambridgeport, Mass.
Moran, Dan'l E ....New York City.
Morgan, T. R...............................Alliance, O.
Morgan, T. R., Jr..........................Alliance, O.
Moulthrop, Leslie..........................New Haven, Conn.
Muller, E. A ....Cincinnati, O.
Müller, T. H.................................Philadelphia, Pa.
Nason, C. W. (*Manager*)........ ....New York City.
Nicholson, Stephen........................Providence, R. I.
Nicholson, T. W............................Providence, R. I.
Nicolls, Chas. H ....Newark, N. J.
Norris, R. Van A.................. ....Wilkesbarre, Pa.
Odell, W. H.................... ....Yonkers, N. Y.
Oviatt, D. B.................................Atlanta, Ga.
Painter, Wm.... ....Baltimore, Md.
Park, W. R..................................Boston, Mass.

Parker, C. D. . . . . Worcester, Mass.
Parker, C. H. . . . . Cambridgeport, Mass.
Parkhurst, E. G . . . . Hartford, Conn.
Parks, E. H. . . . . Providence, R. I.
Passel, Geo. W . . . . Cincinnati, O.
Payne, D. W. . . . . Elmira, N. Y.
Peabody, Cecil H. . . . . Boston, Mass.
Peard, Jas. J. . . . . Hartford, Conn.
Penney, E. . . . . Waynesboro, Pa.
Philbrick, F. E. . . . . Waterville, Me.
Phillips, Franklin. . . . . Newark, N. J.
Philp, C. von. . . . . Bethlehem, Pa.
Pickering, T. R. . . . . Portland, Conn.
Pike, W. A. . . . . Minneapolis, Minn.
Plummer, F. J . . . . Norwich, Conn.
Pollock, Jas. . . . . Wilkesbarre, Pa.
Ponten, Anders . . . . New York City.
Porter, H. F. J. . . . . Pittsburgh, Pa.
Porterfield, H. A. . . . . Pittsburgh, Pa.
Powel, S. W. . . . . Hamilton, O.
Quint, A. D. . . . . Hartford, Conn.
Redwood, I. I. . . . . Brooklyn, N. Y.
Reist, H. G. . . . . Lynn, Mass.
Remick, O. P. . . . . Boston, Mass.
Rice, Richard H. . . . . Providence, R. I.
Richards, Chas. B. . . . . New Haven, Conn.
Richards, Francis H. . . . . Hartford, Conn.
Roberts, Wm . . . . Waltham, Mass.
Robertson, R. A., Jr. . . . . Providence, R. I.
Robinson, G. H. . . . . Providence, R. I.
Roelker, H. B. . . . . New York City.
Rogers, W. A. . . . . Waterville, Me.
Roney, W. R . . . . Chicago, Ill.
Ross, E. L. . . . . Indian Orchard, Mass.
Russel, Walter S. . . . . Detroit, Mich.
Russell, C. M . . . . Massillon, O.
Sancton, E. K. . . . . Scranton, Pa.
Schwanhausser, Wm. . . . . Brooklyn, N. Y.
Schwarz, F. H. . . . . Lawrence, Mass.
See, Horace (*Past President*). . . . . New York City.
Sellers, Coleman, Jr. . . . . Philadelphia, Pa.
Sharp, Joel. . . . . Salem, O.
Sheldon, F. P. . . . . Providence, R. I.
Shepherd, Wm. G. . . . . Chester, Pa.
Sims, Gardner C. . . . . Providence, R. I.
Slater, A. B. . . . . Providence, R. I.
Slater, H. C. . . . . Boston, Mass.
Smith, Chas. P. . . . . Cambridgeport, Mass.
Smith, Geo. H . . . . Providence, R. I.
Smith, Jesse M. . . . . Detroit, Mich.
Smith, Oberlin (*Past President*). . . . . Bridgeton, N. J.

Smith, Scott A. . . . . Providence, R. I.
Snell, H. I. . . . . Philadelphia, Pa.
Sowter, I. G. . . . . Detroit, Mich.
Spaulding, Hollon C. . . . . Boston, Mass.
Sperry, Chas. . . . . Port Washington.
Springer, J. H. . . . . Hamilton, O.
Stearns, Albert. . . . . Brooklyn, N. Y.
Stetson, Geo. R. . . . . New Bedford, Mass.
Stiles, N. C. . . . . Middletown, Conn.
Stillman, F. H. . . . . New York City.
Stratton, W. H. . . . . Providence, R. I.
Suplee, H. H. . . . . Stamford, Conn.
Swasey, Ambrose. . . . . Cleveland, O.
Sweeney, J. M. . . . . Wheeling, W. Va.
Sweet, Jno. E. (*Past President*). . . . . Syracuse, N. Y.
Tallman, F. G. . . . . Beaver Falls, Pa.
Tatnall, Jas. E. . . . . Bethlehem, Pa.
Taylor, Stevenson . . . . Hoboken, N. J.
Terrell, Chas. E. . . . . Brooklyn, N. Y.
Thomson, Jno. . . . . New York City.
Thorne, W. H. . . . . Philadelphia, Pa.
Tompkins, S. . . . . Charlotteville, Va.
Towle, Wm. M. . . . . Ithaca, N. Y.
Tribe, Jas. . . . . Providence, R. I.
Trump, E. N. . . . . Syracuse, N. Y.
Tynan, J. W. . . . . Savannah, Ga.
Vernon, W. G . . . . Philadelphia, Pa.
Wales, C. M. . . . . New York City.
Wallace, Wm. . . . . Ansonia, Conn.
Walworth, A. C. . . . . Boston, Mass.
Ware, J. A. . . . . Worcester, Mass.
Warren, Jno. E. . . . . Cumberland Mills, Me.
Watts, Geo. W. . . . . Philadelphia, Pa.
Weaver, S. B . . . . Chicago, Ill.
Webber, W. O. . . . . Erie, Pa.
Webster, Jno. H. . . . . Boston, Mass.
Weeks, Geo. W. . . . . Clinton, Mass.
Weightmann, W. H. . . . . New York City.
Wheeler, F. Meriam. . . . . New York City.
Wheelock, Jerome. . . . . Worcester, Mass.
Whitehead, Geo. E. . . . . Providence, R. I.
Whitney, B. D. . . . . Winchendon, Mass.
Whitney, Wm. M. . . . . Winchendon, Mass.
Whitney, E. H. . . . . Providence, R. I.
Wiggin, W. H. . . . . Worcester, Mass.
Wilbraham, Thos . . . . Philadelphia, Pa.
Wilcox, Jno. F. . . . . Pittsburgh, Pa.
Wiley, Wm. H. (*Treasurer*). New York City.
Williamson, W. C. . . . . Philadelphia, Pa.
Winship, J. G. . . . . New York City.
Wood, De Volson (*Vice President*). . . . . Hoboken, N. J.

Woodbury, C. J. H. .... Boston, Mass.
Woolson, O. C .... Newark, N. J.
Worthington, C. C. .... New York City.
Wright, J. Q. .... Fitchburg, Mass.
Wyman, H. W .... Worcester, Mass.

This list numbering 279 should be supplemented by 21 names of guests, professional friends and candidates for election, making an aggregate of 300 gentlemen in attendance upon the convention. There were 95 ladies in the party, and, so far as known, this record constitutes the highest mark for attendance attained by any engineering convention anywhere.

The opening session was called to order at ten o'clock by Col H. A. Du Villard, Chairman of the Local Committee, who introduced His Honor, Charles Sidney Smith, Mayor of the City of Providence, who made an address of welcome, to which the President of the Society, Mr. Robert W. Hunt of Chicago, made a fitting response.

The professional papers were at once taken up, those allotted for the morning being entitled: Two Rope Haulage Systems, by R. Van A. Norris; A Belt Dynamometer, by S. P. Watt; A New Belt Testing Machine, by G. I. Alden; and Test of a Triple Expansion Engine, by John T. Henthorn. These received discussion (as appears in the sequel) by Messrs. T. S. Miller, E. F. C. Davis, J. F. Holloway, William Kent, Wilfred Lewis, and D. S. Jacobus.

The session concluded with business matters, the first being the report of the tellers of election, upon the four ballots taken during the interval since the last convention, upon candidates for membership. Their report was as follows:

The undersigned were appointed a Committee of the Council to act as tellers, under Rule 13, to count and scrutinize the ballots cast for and against the candidates proposed for membership in the Society of Mechanical Engineers, and seeking election before the XXIIId meeting of the Society, in June, 1891.

They would report that they have met upon the designated days in the office of the Secretary, and proceeded to the discharge of their duties.

There were 474 votes cast in the ballot upon the Pink List, of which ten were thrown out because of informalities, the member voting having omitted to sign his name to his ballot.

There were 458 votes cast in the ballot upon the Yellow List, of which 15 were thrown out because of informalities.

There were 451 votes cast in the ballot upon the Orange List, of which 17 were thrown out because of informalities.

There were 439 votes cast in the ballot upon the Green List, of which 18 were thrown out because of informalities.

They would certify for formal insertion in the Records of the Society to the election of the appended-named persons, to their respective grades, upon lists numbered respectively 1, 2, 3 and 4.

CARLETON W. NASON,
STEPHEN W. BALDWIN,
Tellers.

AS HONORARY MEMBERS.

Bessemer, Sir Henry. Léauté, Henri.

AS MEMBERS.

Ayer, James I.
Agnew, J. H.
Baldwin, F. Ruel.
Benham, Elijah B.
Billings, F. C.
Bogle, John M.
Brashear, John A.
Brown, Robert A.
Bushnell, Fred. N.
Camacho, Leopoldo A.
Clapp, Geo. H.
Clemens, Ernest V.
Conant, Hezekiah.
Conway, Geo. M.
Covell, Harry N.
Darling, Edward.
Davidson, George.
Davids, Chas. H.
Doble, William A.
Farmer, Thomas, Jr.
Ferguson, J. W.
Fish, Charles H.
Fitts, James H.
Fleming, Wm. R.
Gabriel, Wm. A.
Gentry, F. W.
Giles, C. E.
Gleaves, R. Taylor.
Godfrey, E. S.
Granger, Wm. S.
Grant, W. H. E.
Green, M. A.
Green, Wm. O.
Gwilliam, Geo. T.
Haberlin, Hermann.
Hall, Julien A.
Harkness, Wm.
Hart, Fred'k L.
Helvey, Geo. H.
Hodge, Harry S.
Hunter, Jos. L.
Hunting, Alfred A.
Janson, Ernest N.
Jones, Frank Cazenove.
Johnson, Ed. E.
Johnson, J. B.
King, Chas. G. Y.
Knickerbacker, John.
Locke, Sylvanus D.
Lord, Hiram F.
McGeorge, John.
Maddocks, Wm.
Makepeace, Chas. R.
Mason, H. I.
Matheson, Wm. G.
Matthews, Edwin S.
Mitchell, Albert E.
Moore, M. F.
Norton, Chas. H.
Oviatt, David B.
Paine, Charles.
Payne, S. F.
Peard, Jas. J.
Pentz, Albert D.
Perry, Jas. H.
Philbrick, Frank B.
Piercy, Edgar M.
Pitkin, Stephen H.
Plummer, Frank J.
Quint, Alanson D.
Reeve, Chas. D.
Repath, Charles H.
Rider, Frank A.
Ridsdale, T. W.
Rinman, Gustav O.
Ritchie, William.
Rites, Francis M.
Sakata, S.
Sargent, Chas. E.
Searle, Jos. M.
Shipley, Thomas M.
Slater, Alpheus B.
Sneddon, Jas. P.
Stanton, John.

Stetson, A. B.	Taylor, John T.	Wagner, Emile F.
Stryker, Francis A.	Turner, W. C.	Weaver, Seth B.
Sulzer, Jacob Carl.	Veit, Richard C.	Winship, J. G.
Sylven, Walfrid T.	Vose, Clarence.	Williams, D. Curtis.
Witman, Noel B.	Worthen, Wm. H.	

FOR PROMOTION TO FULL MEMBERSHIP.

Burchard, Anson W.	Conrad, Hugh V.	Huson, Winfield S.
Laird, John A.	Lipps, Henry, Jr.	

FOR ASSOCIATES.

Beach, Giles.	Fenner, Herbert N.	Moulthrop, Leslie.
	Wilcox, Edwin B.	

AS JUNIORS.

Anderson, F. Paul.	Hibbard, Herb't Wade.	Patitz, Martin.
Blankenship, R. M.	Hobbs, Franklin W.	Patitz, Max.
Brill, Geo. M.	Hough, David Leavitt.	Prosser, Joseph C.
Brown, Robert S.	Isbelle, Howard L.	Reading, Robert B.
Campbell, Gordon.	Kimball, Clement F.	Richmond, K. C.
Case, Theodore N.	Krause, Julius.	Royce, Daniel.
Church, E. D., Jr.	McClelland, Edwd. S.	Schaeffer, John V.
DeLancy, Darragh.	Marr, Geo. H.	Slatter, H. C.
Echeverria, Ricardo J.	Meeker, Warren H.	Temple, Robert K.
Fisher, Elbert C.	Myers, Geo. F.	Walcott, Morgan.
Hammer, Oscar.	Nathan, Alfred.	Wilcox, Wallace J.
Harris, Benj'n M.	Overpeck, Ora E.	Williams, Otis L.

SUMMARY.

Honorary Members	2
Members	98
Promotions	5
Associates	4
Juniors	36
Increase in Membership	140

Second notice was given under the Rules of a Proposed Amendment to Article 31 of the Rules, whereby the Nominating Committee would be expected to present two nominees instead of one for each office falling vacant at each election.

The Committee on a Standard Method for Testing the Efficiency of Locomotive Engines reported progress. The Committee on Standard Method for Conducting Duty Trials of Pumping Engines reported that, pursuant to the action taken at Richmond, it had incorporated into the draft of the original report the suggested improvements which the Committee had accepted, and that the report in its final form has formed one of the papers of the Richmond Meeting as No. CCCLXXXI., and with the discussion as No. CCCCXXXVII. appears in Volume XII. of the Transactions.

The Society's Committee on Standard Units of Measurement reported resolutions adopting the Henry as the unit of electrical self-induction; requesting that a committee be appointed to memorialize the Director of the Mint in reference to the possible deterioration in value of the standard Troy pound, and requesting the privilege of the use of circular letters in the Society's name. These resolutions were on motion referred to the Council to consider and report back to the Society

The Society's Committee on Standard Methods of Testing Materials reported through its Secretary and Reporter, Mr. Gus. C. Henning, as follows:

Mr. Gustavus C. Henning.—The Committee would report that considerable work has been done in the last six months. Over 500 letters have been sent to engineers and manufacturers in this country and Europe, and numerous replies have been received, particularly from abroad. Only a few letters were sent to France, as the necessary lists of names and addresses have not been available. The one prominent point of replies from abroad is the possibility of co-operation, to give the work of the Committee an international character and value. The German engineers have suggested that definite action be taken at the Conference to be held in Vienna in 1892. The Committee expect to make a report in November which will show fully what has been done. Recent investigations in France, Germany, and Russia have developed the possibility of putting all testing on a scientific basis (as it has been shown that shape has a definite and determinate influence on certain qualities of material) in the shape of test-pieces. The first steps have now been taken to express elongation on the basis of a standard shape, from which that of any similar one can be determined. It has been discovered that there is a relation between results obtained from test-pieces of different shapes. The development of this fact, first stated by Barba in 1880, has demonstrated its more general applicability. The Committee have issued a supplementary Report, now in the hands of the printer, and which it is expected will be issued in a few days.

This is a translation of the work done by the last Conference in Berlin, held in October last, and in the appendix to that Report is an investigation into the comparative elongation of different shapes of test-pieces. These investigations were made at the Imperial Laboratory Department of Roads and Railways in St. Petersburg. This appendix in connection with the other work will form a voluminous report which may be ready for the next annual meeting in November, although if not then ready it is hoped members will have a little patience on account of the great amount of work to be done.

Mr. William Kent.—In connection with this Report I would call attention to the fact, that I received not long ago from the Secretary of the American Society of Civil Engineers a circular letter which showed that that Society is at work on this same subject through a committee. I think it will be very unfortunate if the different engineering societies should present any divergences in their work resulting from any lack of co-operation with each other. Such a course would delay the establishment of a standard, if not defeat it.

Mr. Henning.—The Chairman of that Committee of the American Society of Civil Engineers, of which our President happens to be a member, has addressed us on this very subject, and the Committee have under consideration the matter of co-operation, because they are well aware of the fact that there can be no difference of opinion on this subject. The work in hand is simply questions of fact, and when facts are sifted the same conclusions must be reached. The reports may be identical, but they cannot be antagonistic. I do not believe that there will be any conflicting points in the two reports. The Committee is attempting to have a joint consideration of the subject, and will very surely take the necessary steps to have a joint meeting of the Committee of the Society of Civil Engineers and our Committee shortly.

The President.—I am glad that Mr. Henning has made that statement. I can assure you that there is not the slightest danger of the Civil Engineers' Committee's Report being issued in advance of yours, and in view of the steps taken by the Chairman of that Committee to bring about this joint consideration, certainly we need not expect any clashing. It must be very gratifying, gentlemen, to have a report from such a committee as yours, and one which promises such a result. It has been my honor to be on a number of such committees, and I think we have all known of cases where much good would have been done if some one had been able to devote to the work the energy which has been shown in your present Committee. I congratulate this society upon its Committee.

The President announced the appointment, under Article 31 of the Rules, of the Committee to nominate officers for the Society for the next year. The Committee is as follows:

T. J. Borden	Fall River, Mass.
Thos. Allison	Port Carbon, Pa.
T. Spencer Miller	New York City.
F. Meriam Wheeler	Montclair, N. J.
John Thomson	Brooklyn, N. Y.

The first session then adjourned.

In the afternoon, leaving the Narragansett Hotel shortly after 2 o'clock, the ladies of the party were taken by carriages to the works of the Gorham Manufacturing Company in Elmwood, while the members and guests went out to Earl Street by special train at 2:40 o'clock. The arrival of the two detachments was well timed. There were fully 200 men and 65 ladies in the party, which was said to be the largest visiting company the Gorham establishment has ever entertained. Under the guidance of the company's officials, the visitors were taken through the several departments and were afforded an excellent opportunity to see all the elaborate product of the works in process. The tour was completed a few minutes after 4 o'clock. Each lady of the party was handed a souvenir silver spoon in a handsome box. The spoon was a memento of the manufactory, and had a view of the buildings etched on the bowl. The ladies were then taken to Roger Williams Park, where a luncheon was served at What Cheer Café, while the special train followed the 4:17 o'clock regular train into the city, stopping at Acorn Street to allow members to visit the plants of the Nicholson File Company and the Rhode Island Locomotive Works.

In the evening the second session was convened in St. John's Hall at 8 o'clock. The papers of the evening were by D. S. Jacobus, entitled "Comparison of the Economy of Compound and Single Cylinder Corliss Engines, each expanding about sixteen times;" by De Volson Wood, entitled "Flexure of Thin Elastic Rings;" by Fred. A. Halsey, entitled "The Premium Plan of Paying for Labor," and two papers on the "Applications of Hirn's Analysis to Engine Testing," one by Cecil H. Peabody and the other by R. C. Carpenter. The debate on these papers was participated in by Messrs. Thurston, Peabody, Wheeler, Alden, Jacobus, Hawkins, Davis, Webber, Ball, Kent, Suplee, Gantt, Almond, Norris, McBride, Weightman, Ashworth and Hunt, and this in spite of the intense heat of the day and of the room. At the close of the discussion the Council reported, through the Secretary, that it had considered the report of the Society's Committee on Standard Units of Measurement, and recommended a resolution referring the Report back to the Committee for further information, in view of the fact that adequate information was not at hand in reference to the matters of which it treats. This action, on motion, was taken.

Mr. Frederick Meriam Wheeler, acting under Article 45 of the Rules, presented the following amendment to the Rules by written notice:

PROVIDENCE, R. I., *June* 16, 1891.

MR. PRESIDENT:

I beg to give this notice in writing (agreeably to the requirements of the Rules of our Society), of amendments that I shall propose at the next (annual) meeting, namely: that the initiation fee for members and associates shall be increased from $15 to $25, and their annual dues from $10 to $15, and for juniors the dues shall be $10, and initiation fee $15.

In other words, I shall move, at the next meeting, to amend Article 18 of the Rules of the American Society of Mechanical Engineers, to amend the figures $15, and figures $10 (as per printed copy of said Rules), to read $25 and $15, respectively, for members, and $15 and $10 for juniors.

I shall also make a motion to amend the figure $5, as printed on the fifth line of Article 18 of the Rules, to read $10, so that a junior, when promoted to full membership, shall pay the additional initiation fee of $10—which amount, in addition to the original initiation fee of $15, paid by said junior, shall make his payments ultimately the same as made by a full member. The fee to be paid at one time for life-membership in the Society should be raised from $150 to $200 pursuant to the above changes.

FREDERICK MERIAM WHEELER.

This called for no action at this meeting. The session then adjourned.

SECOND DAY. JUNE 17TH.

The third session was convened at half-past nine in the morning, in St. John's Hall. The papers presented and discussed were by R. C. Carpenter, "Notes Regarding Calorimeters;" by J. Burkitt Webb, on "Jet Propulsion and the Performance of a Steam Reaction Wheel;" by Daniel Royse, "Heat Transmission through Cast-iron Plates;" by R. H.

Thurston, on "Steam-engine Efficiencies;" by W. W. Bird, on "The Effect of the Steam-jacket on Cylinder Condensation;" by H. L. Gantt, on "Steel Casting;" and two by W. A. Rogers, on "Dividing an Index Plate into One Thousand Equal Parts," and "An Additional Contribution to the Perfect Screw Problem." The debate on these papers was by Messrs. Peabody, Barrus, Nason, Henning, Wood, Suplee, Wheeler, Kneass, Sweet, Davis, Thurston, Denton, Hawkins, Oberlin Smith, Gobeille, Kent, Parker, Howe, Morgan. Messrs. Bissell and Wheeler spoke to the Topical Query in reference to speeds of caloric engines; Messrs. Webber, Idell, Stillman, Capen, and Wheeler spoke on the subject of speeds of Corliss dash-pots; Messrs. Webber, Jewett, Davis, Gantt, and Henning discussed the question of better economy by slow melting in a cupola furnace.

The President reported to the Society the action of the Council, by which members participating in the Society's excursions to shops and manufacturing establishments were requested to refrain from bringing cameras and photographic appliances into such establishments. It was recognized, on the one hand, that the Society as an organization had received and would receive invitations to inspect works from which as individuals they would be debarred, and this fact should preclude any procedure which might be construed (even mistakenly) as an effort to appropriate for individual use what might there be seen. While recognizing the usual impracticability of photography in shops, yet to avoid even the appearance of discourtesy, visitors would be requested to leave their cameras outside of the establishments which the Society should hereafter visit. The Society's good feeling in this matter manifested itself in applause as the President concluded.

The Secretary also presented from the Society's Committee of Conference with Committees of the other Engineering Societies, the Report and Resolution accepted by the joint committees in session at Chicago, May 15, 1891. This Report and Resolution were as follows:

REPORT.

This body shall be called General Committee of Engineering Societies, Columbian Exposition. The objects of this Committee are:

1st. To provide on behalf of the Engineering Societies represented on this Committee, an Engineering Headquarters for members of all Engineering Societies of the world, who may visit Chicago during the World's Columbian Exposition of 1893.

2d. To promote an International Engineering Congress to be held in Chicago in 1893, under the auspices of the World's Congress Auxiliary of the World's Columbian Exposition.

The Report continues in reference to the organization of the Committee and its officers, and announces the election of Mr. Octave Chanute, as President, and the members of the Executive Committee to consist of E. L. Corthell, E. M. Izard, William Forsyth, C. L. Strobel, Robert W. Hunt, J. W. Cloud, and D. J. Whittemore. Mr. J. W. Weston has been chosen secretary *pro tem*, and Mr. T. J. Korner as treasurer. The Committee passed the following resolutions:

Resolved, That it is the sense of this Committee that the importance of Engineering entitles it to the place of an independent department in the World's Congresses to be held in 1893 under the auspices of the World's Columbian Exposition.

The Society, hearing this report of its representatives, passed the following resolutions:

Resolved, That it is the sense of the Society, to approve heartily of the idea proposed, of an Engineering Headquarters at the Columbian Exhibition of 1893.

Resolved, That until the details of the Engineering Congress which it is proposed to hold shall be more clearly formulated, the Society deems it premature to express an opinion as to its advisability.

During this session, the ladies of the party had been escorted in carriages to visit points of interest on the east side of the city. After dinner, the entire party was escorted to the shops of the W. A. Harris Steam Engine Co., the Brown & Sharpe Mfg. Co., and the Armington & Sims Engine Works.

In the evening, at nine o'clock, a reception was tendered to the Society and its ladies by representative citizens of Providence, in Spink's Hall, on Broad Street. Governor H. W. Ladd of Rhode Island received the guests, and music, a collation, and dancing concluded the evening. The promoters of this pleasant feature of the entertainment at Providence were the leading members of the Advance Club of the city, and their supporters.

THIRD DAY. JUNE 18TH.

The fourth and concluding session for papers and Society business was convened in St. John's Hall at half-past nine in the morning. The papers were those by Messrs. H. M. Howe, on "Manganese Steel;" James E. Denton, on the "Performance of a Worthington Pumping Engine against a Head equivalent to 2,000 Feet of Water;" F. W. Gordon, on "A Blast Furnace Blowing-engine;" Jas. McBride, on "Some Experiments with a Screw-bolt"; and W. R. Roney, on "Mechanical Stokers." Prof. Denton, in presenting his paper, made use of a projection lantern and screen, and the debaters of the session were Messrs. Henning, Rogers, Hunt, Wheeler, Davis, Kent, Holloway, Ashworth, Howe, McBride, and Lewis. The Topical Query as to the use of a water pyrometer was discussed by Messrs. Barrus, Durfee, Barnes, Rogers, Howe, Kent, and Nason.

This completed the professional programme of the Convention.

Mr. E. F. C. Davis, member of the Society's Committee on Standard Flanges for Pipes, Valves, etc., spoke of the embarrassments under which the Committee were and are now laboring by reason of changes of residence and professional engagement of members of the Committee which had made it impracticable for the members to get together, and while the individual efforts of the Chairman of the Committee had enabled them to make a report of some progress when called on, yet the work of the Committee as a whole, since its appointment in 1887, had not been what the Society had a right to expect. He would, therefore, ask that the present Committee should be discharged, by vote of the Society, from further consideration of the question referred to them. On motion, the present Committee was discharged, and the President was directed to appoint a new Committee on Standard Flanges, subject to the approval of the Council.

The following series of resolutions were then presented in short and fitting preambles, and were passed by acclamation.

Mr. Oberlin Smith.—Inasmuch as the sum of human happiness, as contained in the hearts of the members of this Society, has been steadily growing since they have been in Providence, I wish to offer the following resolutions:

Resolved, That the warmest thanks of this Society are due and are hereby tendered to his Honor, Charles Sidney Smith, the Mayor of Providence, for his kind and earnest welcome to this beautiful and busy city—a city which has been the mother of many engineers whose works have become an integral part of the history of the civilization of the nineteenth century.

(The resolution was seconded and unanimously adopted.)

Mr. C. J. H. Woodbury.—While the members of this Society and their ladies have been viewing many points of engineering interest around Providence, I think they could not fail to be struck with the perfection of the street railway service, conforming, as it does in a remarkable degree, to the difficult topographical conditions of the city. Providence was, I believe, the first city to meet the question of internal railway transportation of its citizens in a broad and strong sense, with a proper consolidation of interests, and with such arrangements as would meet the convenience of all, and, therefore, I take pleasure in offering the following resolution:

Resolved, That the members and ladies of the American Society of Mechanical Engineers hereby express their hearty thanks to the Street Railway Companies of Providence for the courtesy of their roads, so generously tendered during the meeting of this Society.

(The resolution was seconded and unanimously adopted.)

I have another resolution very much in the same vein. Providence, like all other metropolises, owes its position of leadership to its relations with the surrounding country, which is tributary to it, and that leadership is certainly in this instance maintained by its facilities of transportation, both of the railways centring in the city, the methods of freight exchange, which are being largely extended at the present time, and also its position as the head of navigation on Narragansett Bay.

Resolved, That the American Society of Mechanical Engineers hereby desire to tender their thanks to the New York, Providence and Boston Railway Company for their kindness in furnishing special trains for the excursions of the Society.

(The resolution was seconded and unanimously adopted.)

Mr. Holloway.—The pleasant duty has been assigned to me of formulating, as best I might, the thanks of this Society to the various manufacturing establishments of Providence and vicinity for the kind invitations extended to us to visit their works. I have felt that a mere formal resolution to that end would, at best, but slightly express the sense of obligation that I am sure we all feel to the gentlemen who have opened to our inspection engineering works, famous not only in our own country, but in all the countries of the world in which the skill, ingenuity, and enterprise of the engineer and artisan have at all been recognized. Coming, as many of us do, from distant cities, it has been not only a source of pleasure, but as well of pride, to see establishments whose products have in many ways reflected credit on the name of the American engineer. Mr. President, I therefore offer the following resolutions:

Resolved, That the thanks of this Society be tendered to the various manufacturing establishments in Providence and vicinity, who have so generously invited and so warmly welcomed us to their works; that while it would be impossible to name in detail the numerous and well-known establishments to which we are thus indebted, we are constrained by the unanimous voices of the ladies of our party to express to the gentlemen connected with the Gorham Manufacturing Company a special vote of thanks, not only for the privilege of witnessing the elegant display of their beautiful wares, but especially for their thoughtful kindness in the presentation of a choice souvenir, that will serve for years to come as a pleasant reminder of a most enjoyable time.

Resolved, That while our limited time has prevented us from visiting in a body very many interesting engineering works of Providence, as well as the works in Pawtucket and Bristol, to which we were kindly invited, we wish to express to the gentlemen connected with these various establishments our warmest thanks for the opportunity thus tendered us, and to express the hope that the fame and renown they now enjoy and well deserve will last as long as will the remembrance of the members of this Society of our most enjoyable visit to this the capital of a small but famous State.

(The resolutions were seconded and unanimously adopted.)

Mr. Jesse M. Smith.—Much has been said already about the wonderful things that have been done in Providence, and of the wonderful things that we have seen, but how these things have been brought about is not so well known. The Advance Club of Providence is the prominent factor in this; therefore I desire to offer the following resolution:

Resolved, That to the Advance Club of Providence, an organization which has already accomplished so much for this city, and whose future is so full of promise, the members of the American Society of Mechanical Engineers and their ladies express their thanks for the thoughtful kindness which has permitted them to meet socially His Excellency Herbert W. Ladd, Governor of Rhode Island, and the good citizens of Providence, who have made our visit one to be so pleasantly remembered.

(The resolution was seconded and unanimously adopted.)

Mr. Russell.—Mr. President, I desire to offer the following resolutions:

HISTORICAL SOCIETY.

The American Society of Mechanical Engineers, assembled at Providence this June 17, 1891, wishes most earnestly to express its thanks for the many invitations of freedom to Hall, Library, Shop, and Household of the good citizens of Providence. The acceptance of one particular invitation will be especially remembered. Your Historical Society, in its new fire-proof quarters, where every shelf and ledge teems with most valuable collections, and where even the historical names of its present officers, Rodgers, Perry, and Everett, lend interest to it, was visited by a special committee, and other members of the Engineers will have missed much if the limited time spent in Providence prevents their examining your treasures and mementos of old colonial history.

PROVIDENCE PUBLIC LIBRARY.

The American Society of Mechanical Engineers, at this, its June Meeting of 1891, wishes to thank the Trustees and Managers of the Providence Public Library for the kind invitation of welcome to its privileges. Some of us have taken advantage of these, and know how much the others have lost. We regret that our stay is so short that a proper examination of your treasures cannot be made by all of us.

PROVIDENCE FRANKLYN SOCIETY.

The American Society of Mechanical Engineers have received with marked favor the kind invitation of Mr. Charles Salsbury, in behalf of the Providence Franklyn Society, and would express its many thanks for this kindness shown. We are indebted to the Franklyn Society, as we are to many other Providence invitations, for pleasure that our short stay almost entirely prohibits, but wish to thank you with all courtesy possible.

LIBRARY OF BROWN UNIVERSITY.

The American Society of Mechanical Engineers at their Providence Meeting in June, 1891, have received with especial favor the kind invitation, through Mr. Gyld, for the welcome of its members to the privileges of the Library of Brown University. Our short stay in your beautiful city will prevent the acceptance of your hospitality but by a few. However, most of us have passed the building, and pronounce it one of the very best of its kind that we have ever seen.

BROWN UNIVERSITY.

The American Society of Mechanical Engineers congratulates itself that this its largest meeting, and the largest meeting, too, of any similar organization in the country, was held this year with "Brown." In the same proportion that our strength and membership have grown, so has Brown University, and although it has been without an "Angell" for twenty years, yet an "Andrews" leads it now with a greater strength of membership and usefulness than it has ever known before. We engineers thank you for the welcome given us, and have visited the many college buildings, and have strolled under the magnificent elms of your college green with pleasure. We look for a time when many of the happy, howling crowd of boys, noted for their college song, and passing by our hotel this 17th day of June, 1891, will be as successful at engineering as they are at base-ball.

(The resolutions were seconded and unanimously adopted.)

Mr. Ashworth.—Mr. President, I desire to offer the following resolution:

Resolved, That the American Society of Mechanical Engineers most heartily return their thanks to Commander Theodore F. Jewell, U. S. N., and his associate officers, in charge of the Torpedo Station at Newport, for the prompt and generous response to the suggestions of the Local Committee in inviting us to witness the action of the torpedoes and the manœuvring of the famous despatch boat *Stiletto.*

Resolved, That in this action we recognize the confirmation of the hearty and thorough coöperation of this great arm of the nation's servants with scientific bodies in their efforts to develop and apply the mechanical appliances which are alike powerful both in peace and in war.

(The resolutions were seconded and unanimously adopted.)

Mr. Smith.—To still further express our gratitude, I offer the following resolution:

Resolved, That the warmest thanks of the ladies attending this convention be tendered to Mr. George H. Smith, who, by his courteous and indefatigable labors in their behalf, in conducting them upon various delightful excursions, has so much added to the pleasure of their sojourn in this beautiful city.

Also the following resolution:

Resolved, That the sincere thanks of this Society are due and are heartily tendered to the Local Committee of Entertainment, who, by their earnest labors, their courtesy, and their pockets, have made so pleasant our long-to-be-remembered visit to the beautiful and hospitable city, which, in many of its engineering aspects, has become, as it were, a Mecca to which engineers from home and abroad seek inspiration in the cultured refinements of their noble craft.

The President.—Gentlemen, in putting these resolutions, I suggest that we take a rising vote.

(The resolutions were adopted unanimously by a rising vote.)

President Hunt, in closing the session, gave expression to his sense of appreciation that everything in the Convention had moved with such unity of good feeling; announced that the autumn meeting would take place in New York City, and pronounced the meeting adjourned.

Excursion Days.

The afternoon of Thursday was devoted to a sail down the Narragansett Bay to Rocky Point in the steamer *Bay Queen,* where a Rhode Island clam-bake was served. The *menu* covered Little Neck clams, clam chowder, clam broth, baked blue fish, and tautog and fried eels. After the baked clams came lobster and clam fritters, with lettuce and cucumbers, strawberries, ice-cream, and coffee, sauterne and cigars. Leaving Rocky Point again the boat steamed down toward Newport, while in transit receiving a call from the Herreshoff boat *Stiletto.* This boat circled round and round the speedy excursion boat with the speed of a railway train, and was a very remarkable exhibition. At the naval station in Newport Harbor, a series of submarine torpedoes was exploded near shore for the interest of the party with great success.

Returning up the bay to Providence, a visit was paid to the Elm Street station of the Narragansett Electric Lighting Co., and later, in the hotel where most of the ladies were, an informal dance was organized in the cleared dining saloon. In spite of the mist and rain, the day was voted a great success.

On Friday, visits were made by the Society in a body to the Corliss Steam Engine Works, the Rhode Island Tool Co., the Hope Pumping Station, the American Ship Windlass Co., and the power station of the Providence Cable Railway Co. In the afternoon a special train conveyed the party to Pawtucket, where under the guidance of Mr. Edwin Darling, member of the Society and superintendent of the water-works, the Society visited the various stations of the water-works. Luncheon was served in a tent at the largest station, presided over by His Honor the Mayor of Pawtucket, supported by other representative gentlemen. The forbidding weather precluded visits to the many points of manufacturing interest in Pawtucket, so that after the meal the company listened to speeches until the hour for return by train.

CCCCXXXIX.*

TWO ROPE HAULAGE SYSTEMS.

BY R. VAN A. NORRIS, WILKESBARRE, PA.
(Junior Member of the Society.)

WITHIN the past two years the Susquehanna Coal Company have put into operation two rope haulage systems, designed by their chief engineer, J. H. Bowden, which are believed to have some novel features. The first of these was erected in December, 1888, to take the empty mine cars from No. 5 breaker back to No. 4 slope. The cars are hoisted from the slope in trips of five cars each (see map, Fig. 181), and run by gravity to the breaker, where they are hoisted to the top in self-dumping cages, lowered, then bumped off and back-switched onto the empty track. The problem was to transport 600 cars per day up a grade varying from $1\frac{1}{2}°$ to $7°$, and around a reverse curve, for a distance of about 600 feet, while the coal for shaft and slope boilers, about 40 cars per day, was hauled from the breaker across these tracks, so that arrangements had to be made to allow the mine locomotive to run to the breaker on the empty track. On these accounts, and because of the number of switches in use, it was decided to run the rope overhead ten feet in the air to clear the locomotive. The rope had to be driven from the breaker engine, an $18'' \times 48''$ plain slide-valve engine, which was found to have ample power to spare, and it was considered advisable to have the "take-up" on the driving end. This was done by mounting the driving sheave on a truck, and driving by means of a square shaft sliding in a cast-iron sleeve to which the driving pinion was keyed (Fig. 182), the end of the square shaft working in a clamp box provided with wheels running on an angle-iron track. The tension was put on the rope by an iron bucket pulling on a three-to-one chain tackle, the bucket being loaded with scrap iron. The rope was a three-quarter-inch diameter steel hemp-centre rope, 19 wires to the strand, and was driven by a

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. of the *Transactions*.

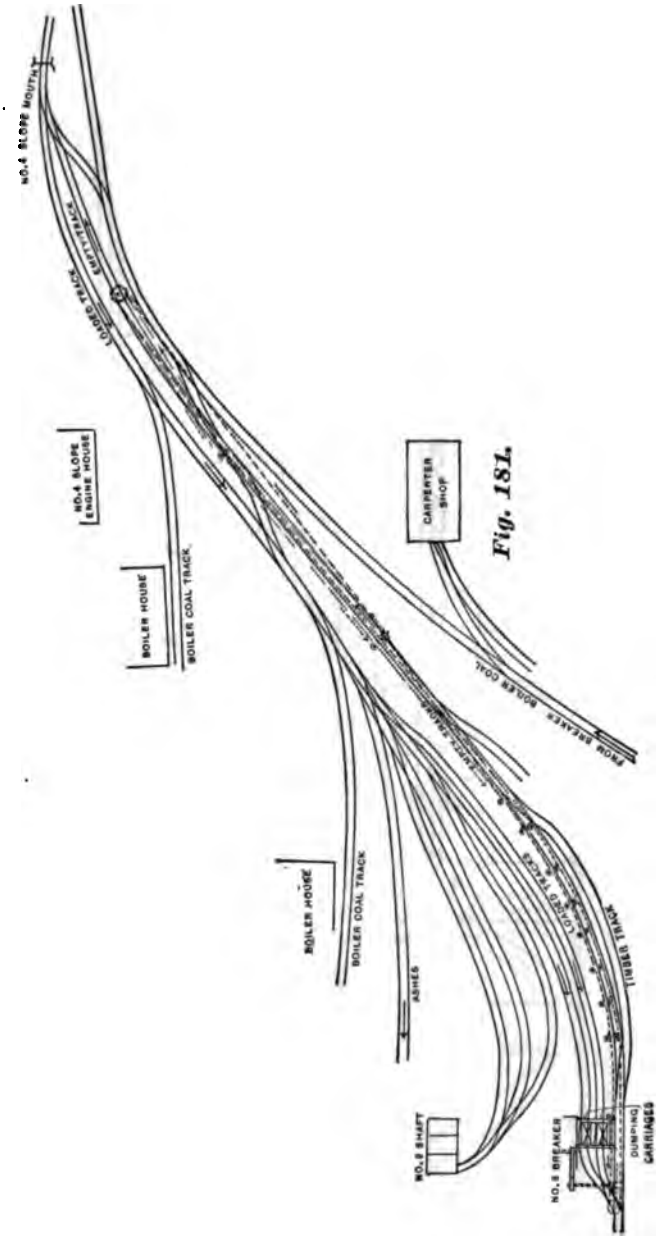


Fig. 181.

plain six-foot sheave (Fig. 183), with the groove filled with babbitt metal, an experiment which it was thought would give the rope a grip without the wear on it incident to a V-groove, and as the rope merely took a half turn on the sheave, some such plan was considered necessary. This babbitt lining has proved a complete success, the rope at first cutting it to the shape of the strands, but afterward wearing smooth; it has shown but three-sixteenths inch wear in two years of use, and has slipped only twice to my

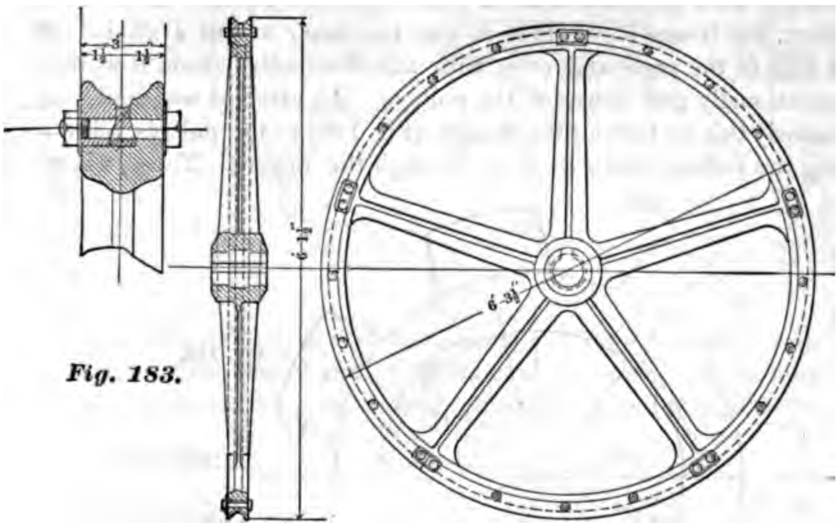
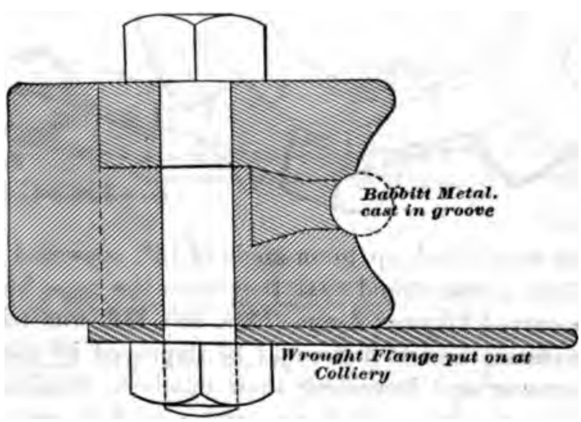


Fig. 183.



knowledge, and then with a new rope freshly oiled, when an attempt was being made to haul a loaded car.

The end sheave at the slope was similar to the driving sheave in every particular, and shows no appreciable wear on the babbitt lining.

The rope was at first run as shown by the dotted lines (Fig. 181), but owing to the pull across the car at the slope-end, which tended to derail cars, it was afterward changed to the position shown by the full line, the return rope crossing over the pulling one. Great difficulty was experienced in keeping the rope up to height. At first the 16-inch sheaves (Fig. 184) with very small flanges were used, the tension of the rope being relied on to keep it up, but it was found that it was necessary to put a strain of 5 tons in the rope, and even with this destructive strain it would occasionally pull down off the pulleys. An attempt was made to remedy this by bolting the flanges (Fig. 188) to the pulleys, allowing the pulling chain to drop through the fingers. These, when

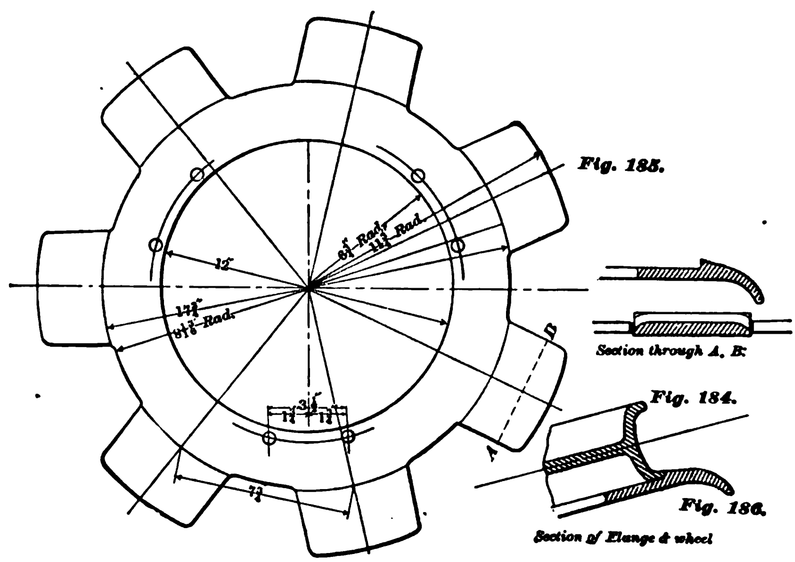


Fig. 185.

Section through A. B.

Fig. 184.

Fig. 186.

Section of Flange & wheel

the sheaves were tilted up to an angle of 15°, answered the purpose well, but it was found that they wore the rope badly, and finally the curved fingered flange (Figs. 185, 186) was tried, with complete success, though that could be improved by making the fingers narrower and increasing their number. Similar fingers were finally put on the end-sheave, and a solid flange 3 inches wide on the driving sheave (Fig. 183). The tension on the rope was finally reduced to 3,200 lbs.

The matter of attaching the pulling chain to the rope was one of no little difficulty. The first attempt was a patent clamp cone (Fig. 187), with the chain attached to a clamp in the middle, allowing the rope to revolve without tangling the chain, which returned merely took a half turn on the sheave, some such plan was considered necessary. This babbitt lining has proved a complete success, the rope at first cutting it to the shape of the strands, but afterward wearing smooth; it has shown but three-sixteenths inch wear in two years of use, and has slipped only twice to my

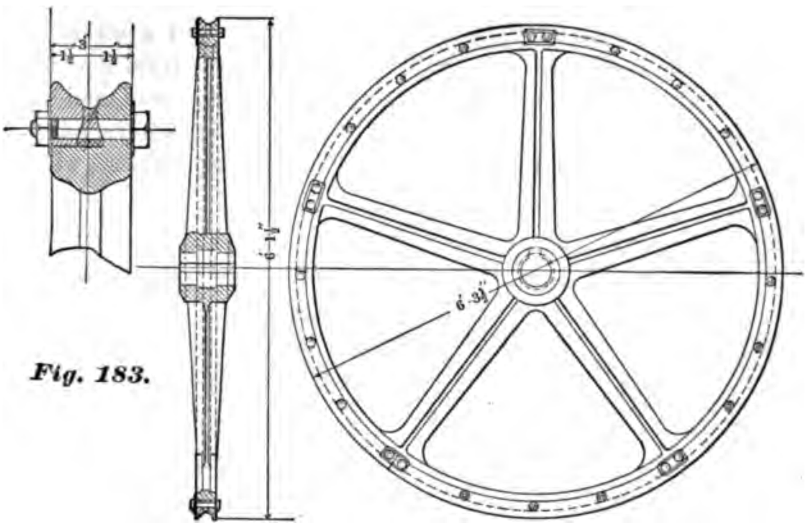
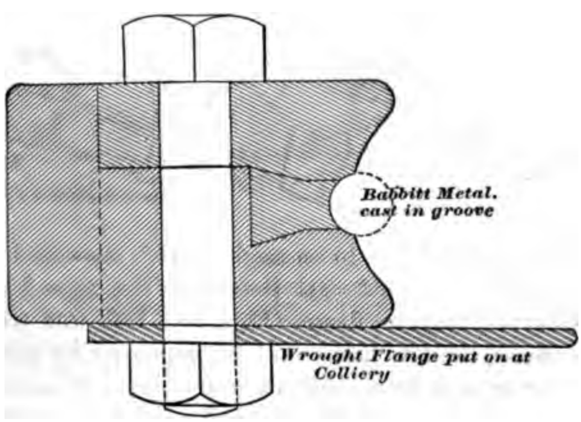


Fig. 183.



knowledge, and then with a new rope freshly oiled, when an attempt was being made to haul a loaded car.

The end sheave at the slope was similar to the driving sheave in every particular, and shows no appreciable wear on the babbitt lining.

The rope was at first run as shown by the dotted lines (Fig. 181), but owing to the pull across the car at the slope-end, which tended to derail cars, it was afterward changed to the position shown by the full line, the return rope crossing over the pulling one. Great difficulty was experienced in keeping the rope up to height. At first the 16-inch sheaves (Fig. 184) with very small flanges were used, the tension of the rope being relied on to keep it up, but it was found that it was necessary to put a strain of 5 tons in the rope, and even with this destructive strain it would occasionally pull down off the pulleys. An attempt was made to remedy this by bolting the flanges (Fig. 188) to the pulleys, allowing the pulling chain to drop through the fingers. These, when

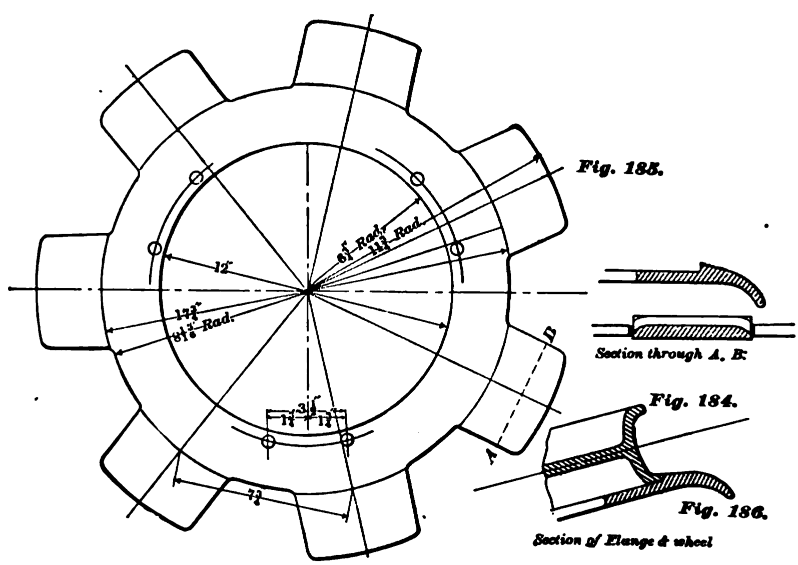


Fig. 185.

Section through A. B.

Fig. 184.

Fig. 186.

Section of Flange & wheel

the sheaves were tilted up to an angle of 15°, answered the purpose well, but it was found that they wore the rope badly, and finally the curved fingered flange (Figs. 185, 186) was tried, with complete success, though that could be improved by making the fingers narrower and increasing their number. Similar fingers were finally put on the end-sheave, and a solid flange 3 inches wide on the driving sheave (Fig. 183). The tension on the rope was finally reduced to 3,200 lbs.

The matter of attaching the pulling chain to the rope was one of no little difficulty. The first attempt was a patent clamp cone (Fig. 187), with the chain attached to a clamp in the middle, allowing the rope to revolve without tangling the chain, which returned to the breaker hanging from the rope. It was not supposed that the small leverage, $2\frac{5}{16}$ inches, would kink the rope to any extent,

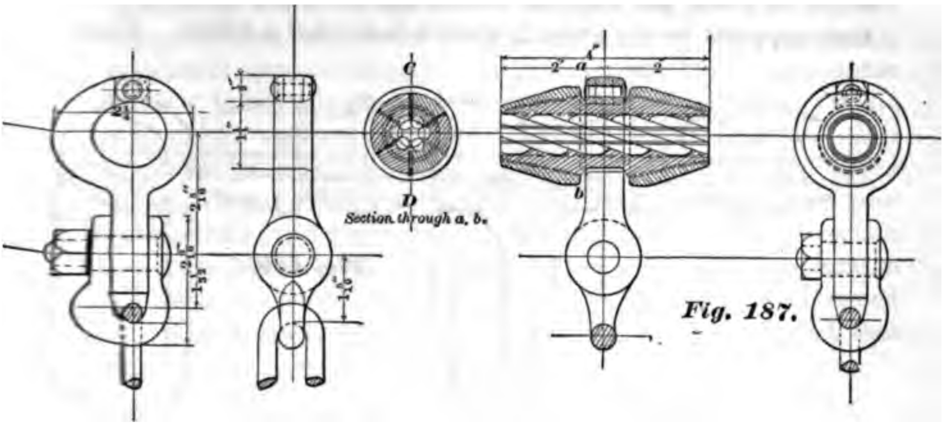


Fig. 187.

but this supposition proved erroneous, the rope breaking with exasperating regularity and promptness at the cone, and the long hanging chain catching on various impediments and bringing the

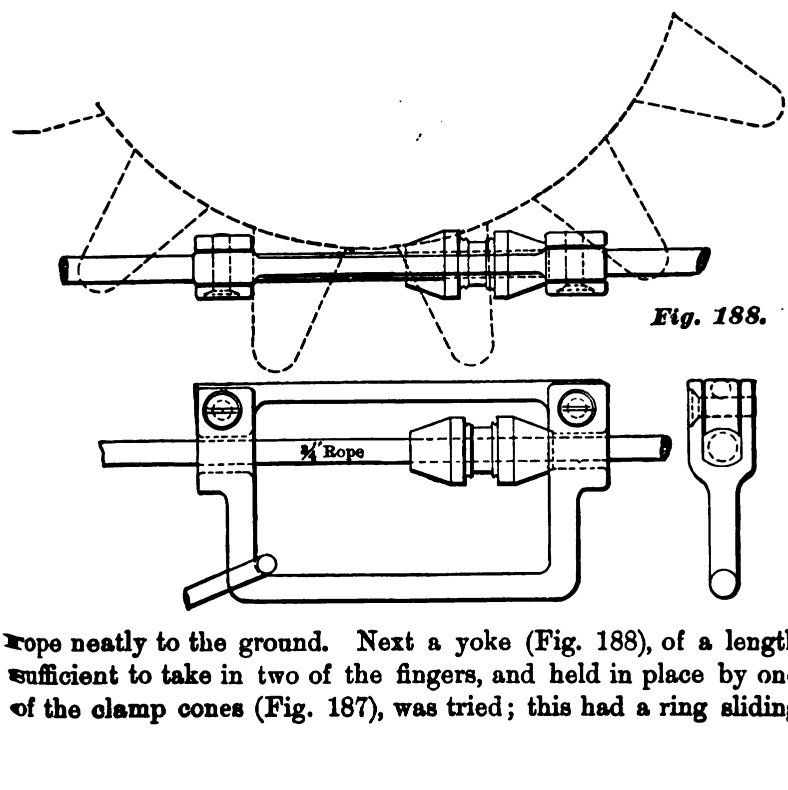


Fig. 188.

rope neatly to the ground. Next a yoke (Fig. 188), of a length sufficient to take in two of the fingers, and held in place by one of the clamp cones (Fig. 187), was tried; this had a ring sliding

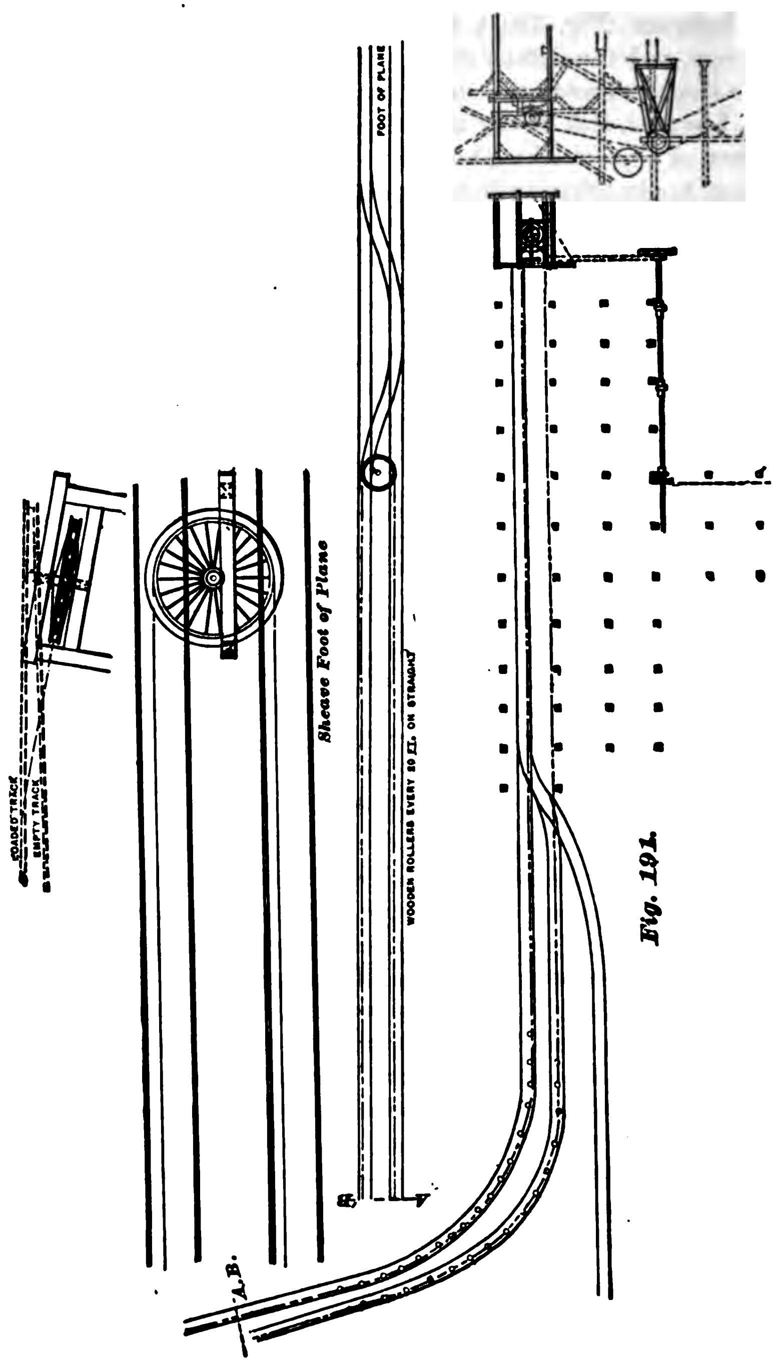

FOOT OF PLANE
Sheave Foot of Plane
WOODEN ROLLERS EVERY 20 FT. ON STRAIGHT
LOADED TRACK
EMPTY TRACK
A.B.

Fig. 191.

link between (Fig. 190a), the cones being securely leaded to the rope, and the pulling chains, hooked into the links, removed at the slope and sent back on the loaded trips to the breaker. The rope, however, still shows a tendency to break at the cones. A number of ways of taking hold of the car were tried, half a dozen forms of hooks being made, tried, and discarded; either they would slip off the cars or they would dig into the wood so that they could not be readily detached. Finally a half-inch pin was driven in the side-bar of each car, and the end of the pulling chain provided with a ring which was slipped over the pin, giving a certain and easy hooking and disengagement, but with the disadvantage of pulling the car from the side. The rope was originally designed to run at a speed of 250 feet per minute, but it was found that

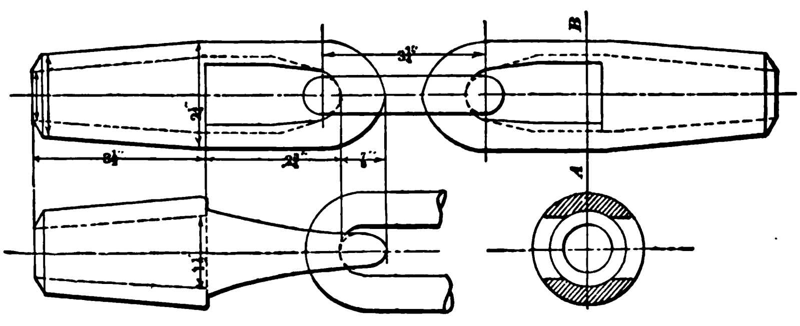


Fig. 190a.

in hooking onto the side of the cars, the jerk tended to derail them, and the pulley on the engine shaft was changed to give a speed of 180 feet per minute, which proved satisfactory.

To provide for instant stoppage of the rope in case of accident, and whenever there were no cars to move, or the locomotive was passing, a Dodge hub friction clutch was placed on the driving shaft, and two three-quarter-inch hemp ropes, one for starting and one for stopping, were carried the full length of the haul and attached to the clutch lever. The whole has now been in successful operation for two years, two boys doing the work which previously required five mules and their drivers. At first only one car was pulled at a time, but now two are attached, the rope being connected to the rear one. The jerk of starting was found by dynamometer tests to be about 25% of the weight of the cars, and the tractive force, corrected for grade, exerted in carrying

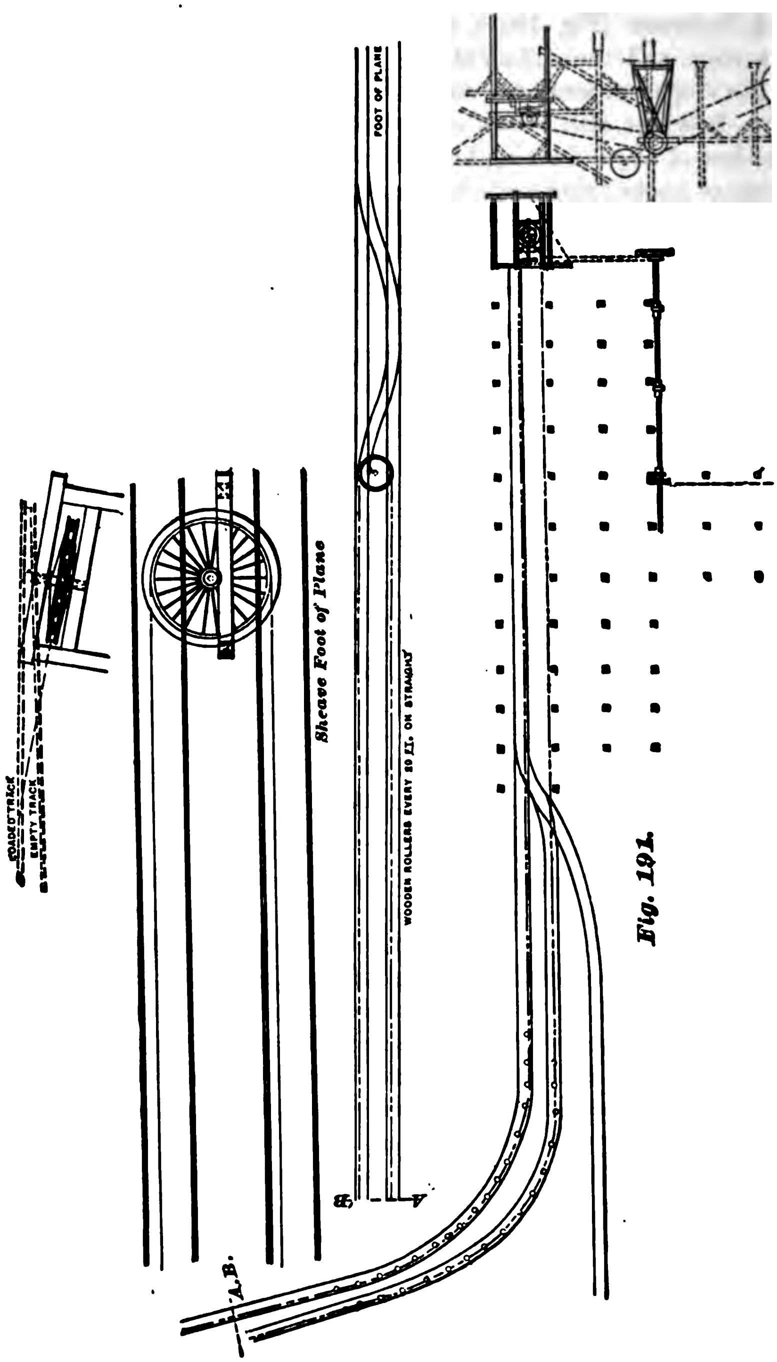
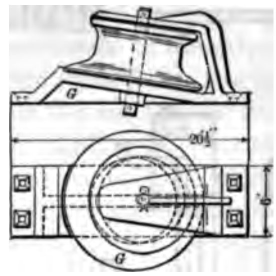

FOOT OF PLANE
Sheave Foot of Plane
WOODEN ROLLERS EVERY 30 FT. ON STRAIGHT
LOADED TRACK
EMPTY TRACK
A.B.
A
B

Fig. 191.

have a V-groove turned in it would have consumed several days, so a tool was made to the shape of the groove required and bolted to the timber of the carriage, the machine, rope and all, run very slowly, and the groove turned out in two hours to the shape shown by the dotted lines, when it was found to drive perfectly, though, of course, it is none too easy on the rope; the tension was, however, reduced to 3,000 lbs. without interfering with the driving. The cars, 12 in all, were attached to the rope by a grip (Fig. 196), catching clamp cones (Fig. 197) * similar to those used on the No. 5 breaker haul; these are constructed of six brass segments cast on a piece of the rope, so that their inside surfaces are an exact reproduction of its exterior, and clamped to it by steel cones screwed on the threaded exterior of the segments; these are readily put on and removed, so far have shown no tendency to slip, and give perfect satisfaction in every way. The moving rope is thrown into the socket (*a*, Fig. 196) by a wooden lever slipped under the car, and the finger (*b*) allowed to spring into place, holding it there until the cone comes along, and, striking the socket, carries the car forward with it, the blow being cushioned by the springs (*c*), each compressing with 600 lbs., against which the grip arm works, as well as by the spring hub in the driving sheave. The latter works admirably, the rope coming almost to a full stop and then gently starting the car.



Curve Roller.
Fig. 194.

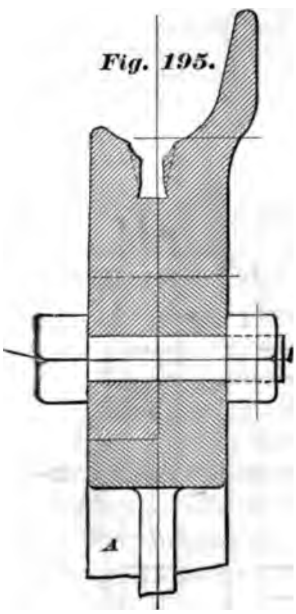


Fig. 195.

The rope is dropped by the sliding out of the finger *b*, which is effected by an uncoupler beside the tracks. It was originally intended to place this inside the track, and to shove the pin out with the arm *d* by having that slide against a curved piece of wood, but it was found that *d* had a tendency to bend, and besides was in the way between the rails. It was consequently removed

* Patented by the chief engineer, J. H. Bowden, October 16, 1888.

lbs., being interposed between the lugs and the hub, allowing a motion of four feet on the circumference of the six-foot diameter rope-driving wheel, to take up the shock of starting. The rope take-up was similar to that at No. 5 breaker, except that the square shaft was shorter and the end unsupported, allowing a "take-up" of twelve feet of rope (Fig. 192). The end-sheave, which was of necessity eight feet in diameter, the distance between centres of tracks, was put in a pit under the tracks and tilted in both directions to bring the two ropes out at equal distances from

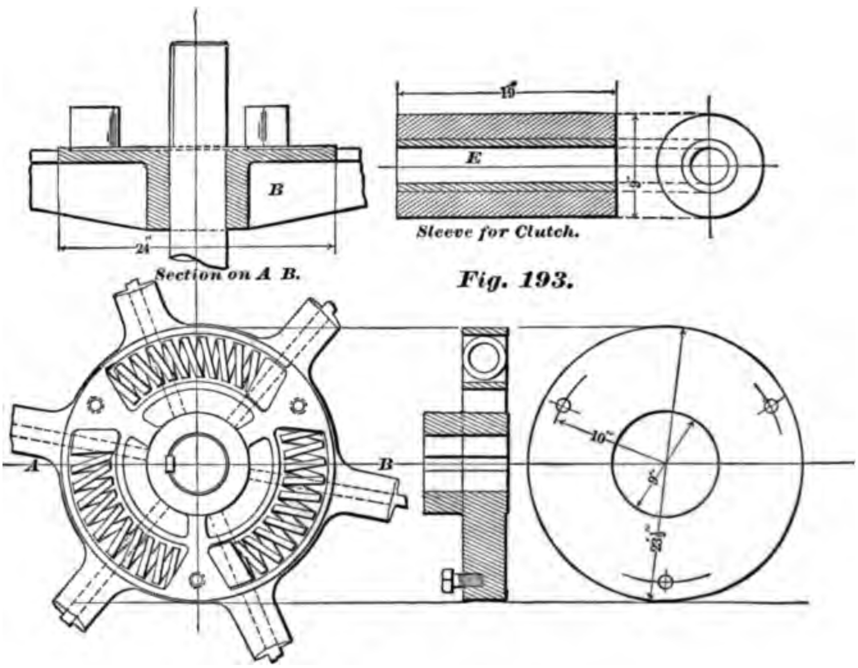


Fig. 193.

it, on the empty and loaded tracks, three feet vertically apart. The rope was carried around the curve on the curve pulleys (Fig. 194), set about four-foot centres on the curve; these work very well, though they already show considerable wear. The throat of the driving sheave in this instance was made of the shape shown (Fig. 195), the groove being filled with hemp; when the machine was started, however, it was found that the driving sheave did not sufficiently grip the rope, a tension of 8,000 lbs. on the rope being just sufficient to move the cars, but not to start them on grade. To take the sheave down and send it to the shops to and the uncoupler applied direct to the handle *E*. This was not at first successful, as the finger could only be moved $2\frac{5}{8}$ inches, and the car had a clearance of over an inch, including end motion of axle and clearance of wheels on track. Finally the solid uncoupler was successfully replaced by a $3 \times \frac{3}{8}$ steel spring, which was strong enough to drop the rope, but not to bend the handle of the grip.

From the drawing it will be noticed that the loaded cars can be picked up anywhere under the breaker, while the empties are unhooked outside and run in by gravity, the empty rope passing under the track. The loaded rope passes one switch rail; which is effected by grading the track so that the natural position of the rope is below the bottom of the rail at this point, the rope being raised in the grip through the open self-acting latch shown, this

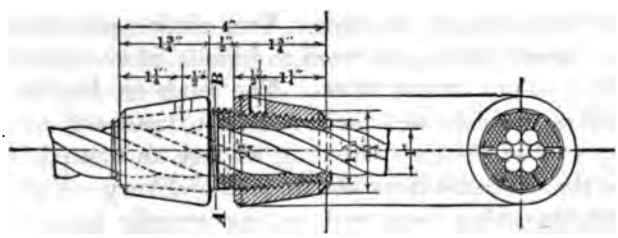


Fig. 197.

latch only closing when the empty cars are running in. A friction clutch operated by ropes extending the entire length of the haul is provided for starting and stopping, as in the No. 5 breaker haulage plant. The cars, both empty and loaded, are run in trips of two cars each, the grips on the last car only being used; the cars are very heavy and hard running, teams of two mules each having all they could do to pull two loaded cars up the grade to the plane.

The history of these two haulage plants with their failures and successes is offered for what it is worth, not as being the best way of accomplishing the results aimed at, but with the hope that better ways may be developed by the criticism of these.

DISCUSSION.

Mr. T. Spencer Miller.—I notice in the latter part of the paper that the driving-wheel had been grooved to an angle which from the drawing seems to be 25° or 30°, and the paper mentions that this was rather hard on the rope. I am wondering to what extent the practice of pinching wire rope in V grooves is being followed now, and also the reason why they did not improve further, and use a grip wheel for driving the cable. I would like to hear something on these points.

Mr. E. F. C. Davis.—I would ask whether they have had any opportunity for comparing this system with what is called in the Schuylkill region the car-hoist system. A great many collieries there have requirements similar to this one. It has been customary to lift the cars up an incline by a constantly moving chain, hooking on the axles of the cars, which runs the cars up the necessary amount of incline to make them run down grade the rest of the road, and that involves very much less mechanism than the rope-haul, and from the fact that the cars are moving constantly there may be as many as half a dozen light cars kept in motion at one time. Two chains are used (about six inches apart) when you want to handle more cars than you could afford to put on one chain. The car-hoist has the advantage of being in constant motion all the time, and it is only necessary to feed the cars on to it as fast as wanted, and the length of the car-hoist does not limit its delivery. I would like to ask whether they have had an opportunity to contrast the economics of the two systems, as I suppose they must have considered that before they adopted the rope-haulage system.

Mr. J. F. Holloway.—I do not rise for the purpose of discussing this paper at all, but simply for the purpose of perpetrating a little ancient history. When I was a young man I was called upon to devise a plan for wire rope haulage, which, perhaps, was the first use of cable rope for that purpose in this country. It was in connection with coal mines in Maryland, and for the Maryland Mining Company, near Frostburgh and near the Borden mines. We had a slope going down into the big coal vein, with a very steep incline at the beginning, and it was thought that some contrivance in the way of hauling out coal by machinery could be used. Nothing was then known about continuous rope-hauling, but I knew something about a belt going around one pulley and over another. I devised a scheme of having an endless wire rope passing around a drum at the head of the slope, and going back to another similar drum which set on an incline, and which carried the rope back to another groove in the driving-drum, until we got two or three turns of the rope around the driving-drum, then going down on one side of the slope with the empty cars and bringing up the loaded cars on the other side. The loaded cars were attached by a pair of eccentrics on the cars, which were run on to the main line, and these grasped the wire rope which was going up the slope, and held on with the peculiarity of holding on the harder when the road was the steeper, until the car got to the top of the incline, where it ran over a little steep pitch, ran ahead, and loosed itself. It was all simple enough, and worked very well. Another peculiar advantage of the arrangement was, that to the large wheel at the bottom of the slope which made the return from one track to the other, I attached machinery for driving-pumps, so that the drainage of the mine was performed by the same power that hauled the cars up. I went back there quite a number of years afterwards, and learned that the rope was still running very nicely and successfully, and hauling the coal out at a less cost than at any mines in that vicinity. I simply mention this ancient history, because I know something about such history.

Mr. Norris.—May I ask Mr. Holloway whether he had a grip on each car, or on a special grip car?

Mr. Holloway.—On each car. At the bottom of each car were two eccentrics partially grooved, and, as the car ran on to the main line, the main driving-rope was raised up until it came opposite the groove, and these eccentrics were clamped on it. As I say, the grade varied, and the eccentric grip held on until it got to the outside, and then let go and the cars ran off down to the dump. Each car had a simple grip attachment of its own.

*Mr. R. Van A. Norris.**—The V groove in the driving-sheaves was put in as a matter of necessity, because the machine would not drive without it. It is not a plan ordinarily used about the mines, nor one which we care to adopt. A grip-wheel would not have taken our cones very safely, which had to pass around it. The car-hoist system referred to by Mr. Davis was considered in both of these cases; in the first it would have interfered seriously with the grades, and have involved a rearrangement of the breaker tracks, as well as an expensive trestle; in the second the cars would have needed to be brought up grade the length of the breaker to a car-hoist located outside, and a second hoist put in at the plane end to return the empties, so that the cost in either case would have been considerably greater than the

* Author's closure.

the practice of pinching wire rope in V grooves is being followed now, and also the reason why they did not improve further, and use a grip wheel for driving the cable. I would like to hear something on these points.

Mr. E. F. C. Davis.—I would ask whether they have had any opportunity for comparing this system with what is called in the Schuylkill region the car-hoist system. A great many collieries there have requirements similar to this one. It has been customary to lift the cars up an incline by a constantly moving chain, hooking on the axles of the cars, which runs the cars up the necessary amount of incline to make them run down grade the rest of the road, and that involves very much less mechanism than the rope-haul, and from the fact that the cars are moving constantly there may be as many as half a dozen light cars kept in motion at one time. Two chains are used (about six inches apart) when you want to handle more cars than you could afford to put on one chain. The car-hoist has the advantage of being in constant motion all the time, and it is only necessary to feed the cars on to it as fast as wanted, and the length of the car-hoist does not limit its delivery. I would like to ask whether they have had an opportunity to contrast the economics of the two systems, as I suppose they must have considered that before they adopted the rope-haulage system.

Mr. J. F. Holloway.—I do not rise for the purpose of discussing this paper at all, but simply for the purpose of perpetrating a little ancient history. When I was a young man I was called upon to devise a plan for wire rope haulage, which, perhaps, was the first use of cable rope for that purpose in this country. It was in connection with coal mines in Maryland, and for the Maryland Mining Company, near Frostburgh and near the Borden mines. We had a slope going down into the big coal vein, with a very steep incline at the beginning, and it was thought that some contrivance in the way of hauling out coal by machinery could be used. Nothing was then known about continuous rope-hauling, but I knew something about a belt going around one pulley and over another. I devised a scheme of having an endless wire rope passing around a drum at the head of the slope, and going back to another similar drum which set on an incline, and which carried the rope back to another groove in the driving-drum, until we got two or three turns of the rope around the driving-drum, then going down on one side of the slope with

CCCCXL.*

TEST OF A TRIPLE-EXPANSION ENGINE.

BY J. T. HENTHORN, PROVIDENCE, R. I.
(Member of the Society.)

THE engine which forms the subject of this paper was specially designed and built by the E. P. Allis Co., E. Reynolds, Am. Soc. M. E., General Superintendent, of Milwaukee, Wisconsin, for the Narragansett Electric Lighting Company of Providence, R. I., for which latter, the writer's firm (Remington & Henthorn) were the designing engineers. It is of the triple expansion type, working as a quarter crank machine. The performance of this engine when on its regular service at the station was made and reported upon to the parties in interest by a board composed of E. D. Leavitt, Am. Soc. M. E., and the writer, and the details of the trial from the official test are herewith presented to the membership, as a matter of engineering information.

The engine has cylinders 14", 25", and 33" diameter, and each 48" stroke. The frame of the engines is formed by two forged iron guide-rod bars, each 4½" square, which are secured to the lugs on the cylinder and main pillow block. Both guide-rod bars have a heavy support underneath at the centre, which in turn is secured to the foundation by anchor bolts. The guides are formed by cast iron V strips bolted on top and bottom of the bars, and can be removed for repairs at pleasure. The high-pressure and intermediate cylinders work tandem on one crank with the high-pressure cylinder astern, and the low-pressure cylinder works upon a second crank set at right angles to that upon the opposite end of the crank shaft. The intermediate and low-pressure cylinders have each two piston rods, each pair of rods passing through its respective cross-head, with the cross-head pin intervening between the ends of the rods. For the one piston rod of the high-pressure cylinder, connection is made to the centre of the piston of the intermediate cylinder, and is secured thereto by shoulder and

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. of the Transactions.

rope-haulage. As to economy, our ropes last about a year, and other repairs are very slight. The first rope-haul requires the services of two boys, one at each end; the second is hauled by the loaders under the breaker and by the plane tender at the plane, thus employing no extra labor; and furthermore, it delivers the cars in trips of two, the way in which they are hoisted on the plane.

CCCCXL.*

TEST OF A TRIPLE-EXPANSION ENGINE.

BY J. T. HENTHORN, PROVIDENCE, R. I.
(Member of the Society.)

THE engine which forms the subject of this paper was specially designed and built by the E. P. Allis Co., E. Reynolds, Am. Soc. M. E., General Superintendent, of Milwaukee, Wisconsin, for the Narragansett Electric Lighting Company of Providence, R. I., for which latter, the writer's firm (Remington & Henthorn) were the designing engineers. It is of the triple expansion type, working as a quarter crank machine. The performance of this engine when on its regular service at the station was made and reported upon to the parties in interest by a board composed of E. D. Leavitt, Am. Soc. M. E., and the writer, and the details of the trial from the official test are herewith presented to the membership, as a matter of engineering information.

The engine has cylinders 14", 25", and 33" diameter, and each 48" stroke. The frame of the engines is formed by two forged iron guide-rod bars, each $4\frac{1}{2}$" square, which are secured to the lugs on the cylinder and main pillow block. Both guide-rod bars have a heavy support underneath at the centre, which in turn is secured to the foundation by anchor bolts. The guides are formed by cast iron V strips bolted on top and bottom of the bars, and can be removed for repairs at pleasure. The high-pressure and intermediate cylinders work tandem on one crank with the high-pressure cylinder astern, and the low-pressure cylinder works upon a second crank set at right angles to that upon the opposite end of the crank shaft. The intermediate and low-pressure cylinders have each two piston rods, each pair of rods passing through its respective cross-head, with the cross-head pin intervening between the ends of the rods. For the one piston rod of the high-pressure cylinder, connection is made to the centre of the piston of the intermediate cylinder, and is secured thereto by shoulder and

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. of the Transactions.

nut. The wrought iron cross-head is what might be termed of the steamboat type, having the pin at the centre and each end turned to fit the shoes or slide of the guide rod.

The details in the construction of the cylinders are shown in section on Figs. 198, 199, and 200, where it will be seen that each is steam-jacketed by forcing a working barrel or liner into

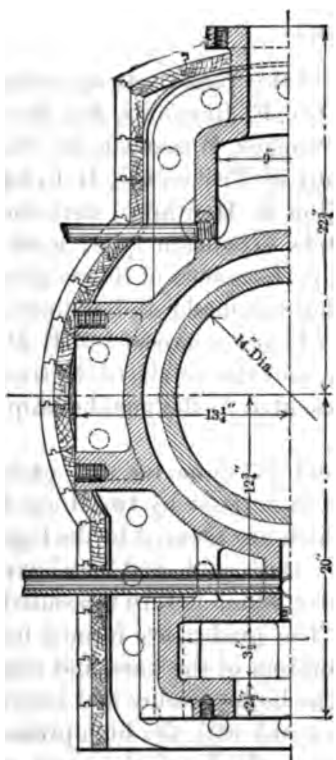


Fig. 199.

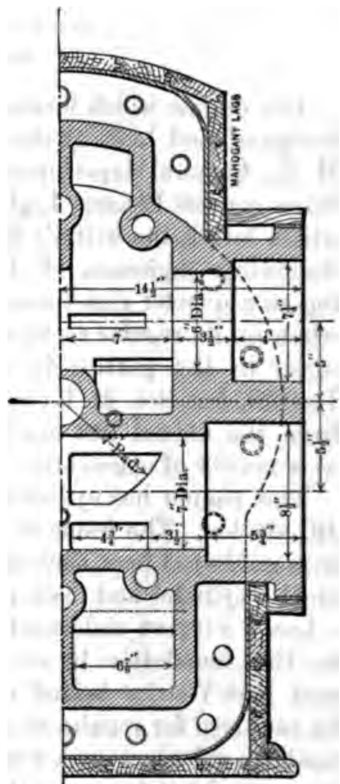


Fig. 198.

the cylinder proper, and then each jointed end made tight by staking a copper ring into a recess or groove cut equally in width in the end of the liner and cylinder casting. The heads of each cylinder form the casing for the Corliss type of valve, and through which steam and exhaust passages communicate with corresponding steam passages in the cylinder casting. All of the steam-valves are double ported to admit of a greater port opening with a minimum valve travel. With the exception of the surface forming the steam-chest for the valves, the heads are not steam jacketed.

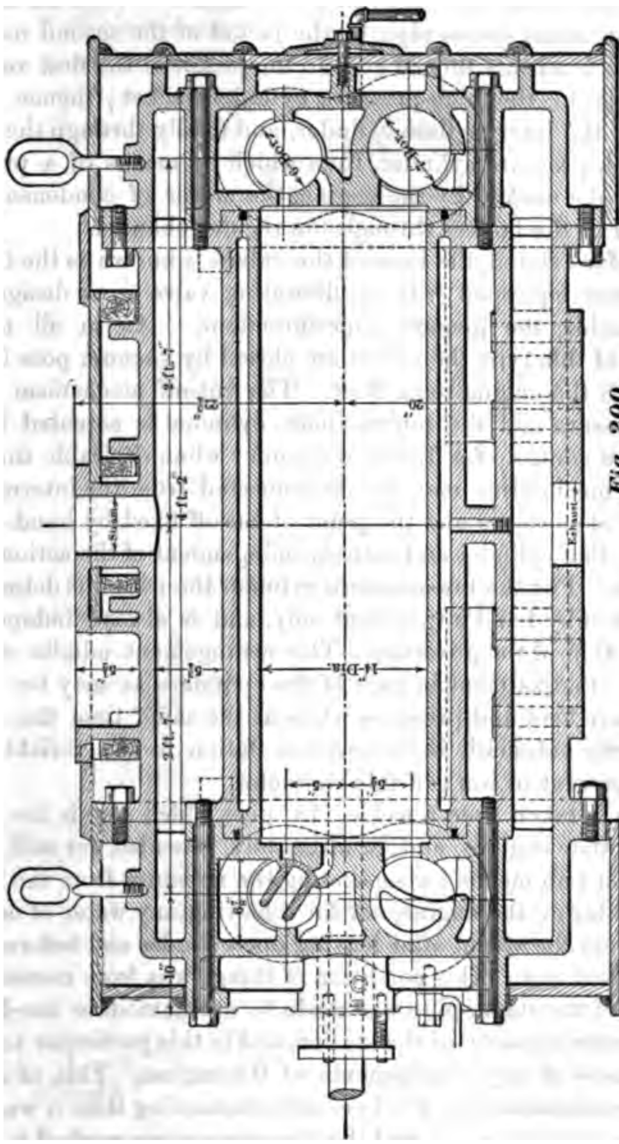


Fig. 200.

The receivers, of which there are two, are each made of welded wrought iron tube, 20″ in diameter, which is enclosed in a similar tube 24″ in diameter, thus leaving $1\frac{3}{4}$″ space between the outer and inner shell, through which circulates the jacket steam at boiler pressure. The course of this steam through the receivers and jackets of the several cylinders is as follows: Steam is taken from the main steam-pipe to the jacket of the second receiver; the drain from this thence goes to the jacket of the first receiver, from this to the high-pressure cylinder jacket; thence to the jacket of the intermediate cylinder, and finally through the jacket of the low-pressure cylinder, from which by means of a pump of short stroke worked by the engine, the water of condensation is returned to the boilers through the regular channel.

As before stated, the type of the valves is known as the Corliss. These are equipped with a liberating valve gear designed by E. Reynolds, the general superintendent. As in all modern engines of this type the valves are closed by vacuum pots located level with the engine-room floor. The cut-off mechanism of the high-pressure and the intermediate cylinder is actuated by and under the control of a fly-ball governor; when desirable this controlling mechanism may be disconnected from the intermediate cylinder at pleasure and the point of cut-off fixed by hand-adjustment for that cylinder and entirely independent of the action of the governor. For the low-pressure cylinder the cut-off is determined at pleasure by hand adjustment only, and is always independent of the action of the governor. This arrangement admits of such a nicety of adjustment in each of the cylinders as may be desirable for any load and pressure, while at the same time the engine is perfectly automatic in its accommodation to the variable load always present in work of this character.

• Steam is taken from a welded 18" pipe which also is the feeder for two other engines, and is ultimately intended for still more. There were no means for separating the moisture from the steam before entering the engine, nor for removing any water of condensation from the steam after leaving one cylinder and before entering the next one. This condition of things was from necessity, as the size of the steam-pipe was made to accommodate one-half of the ultimate capacity of the station, and in this particular test was far in excess of any requirements of the engine. This, of course, caused condensation in the line, notwithstanding that it was covered by a plastic cement, and these circumstances worked together to the detriment of the engine very materially.

The engine exhausts into the condenser through a 26-inch cast-iron pipe, which also is common to all engines of the station. The

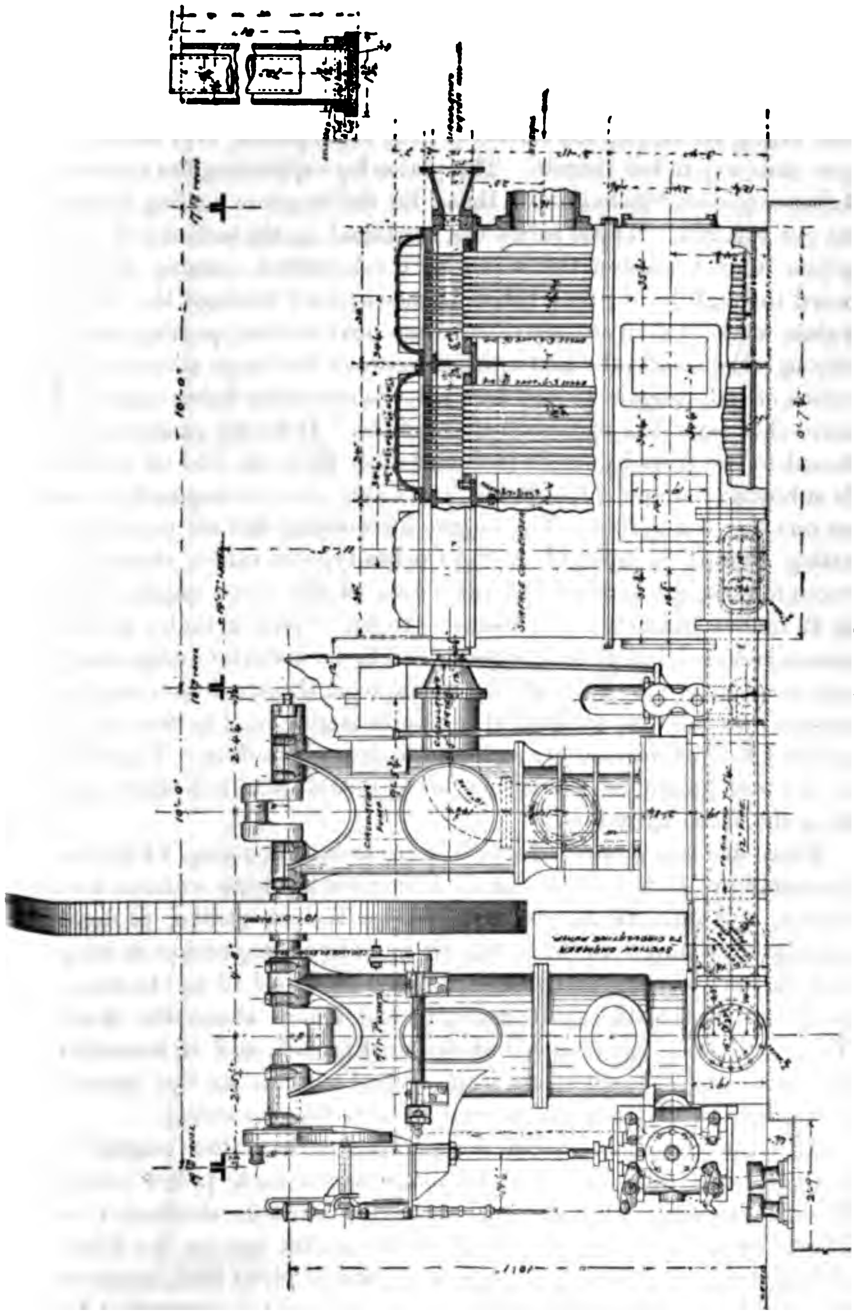


Fig. 201.

condensing apparatus is entirely independent of the main engine, and consists of a condenser 9 feet long, 6 feet wide, and 8 feet high, and contains 2,496 square feet of cooling surface. The tubes, which are $\frac{13}{16}$ inch and $1\frac{1}{4}$ inches in diameter, placed one inside of the other, are suspended vertically from brass plates, with their upper ends expanded therein. The plates for supporting the smaller tubes are some 8 inches above those for the larger or cooling pipes, as per Fig. 201. These tubes are arranged in six sections of 208 pipes in each section, the water of condensation passing downward through the $1\frac{1}{4}$-inch tubes, thence upward through the small tubes, where it then crosses over to the next section, passing down through the small tube and upward between the large and smaller tubes, where it again passes forward, the operation being repeated until the water has traversed six sections. It finally passes overboard to the river by an 18-inch cast-iron pipe, the end of which is submerged about 4 feet below low water, and consequently acts as one leg of a siphon. The engine for working the air and circulating pumps is upright, of the Corliss type of valves, closed by vacuum pots, the same as the valve gear of the large engine, and is 12 inches diameter and 16-inch stroke. This cylinder is not steam-jacketed, but is well lagged with plastic material and mahogany wood covering. The exhaust steam from this condenser engine passes into the first receiver of the main engine, and is thus again utilized in the intermediate and low-pressure cylinders. This fact of the test precluded the laying out of a combined indicator card from the three cylinders.

There are two pumps, of which there is one air-pump, 24 inches diameter by 16-inch stroke, of the lifting bucket type without foot valves, and one, 16 inches diameter by 16-inch stroke, plunger circulating pump. Between the air and circulating pumps is located, on the overhead crank shaft, a plain fly-wheel 10 feet in diameter, the centre of the shaft being 10 feet 1 inch above the floor. This condensing apparatus is shown on Fig. 201, and is intended to receive the exhaust steam from several engines, as the growth of the station may require, as well as from the one tested.

All oil used in lubricating the working parts of the engine is drained back into an oil well located under each pillow-block. From these wells, after passing through a suitable strainer, it is drawn by a small pump, worked by the rocker arm on the high-pressure side, and forced through a system of pipes back again to the working parts. The supply at any one point is controlled by

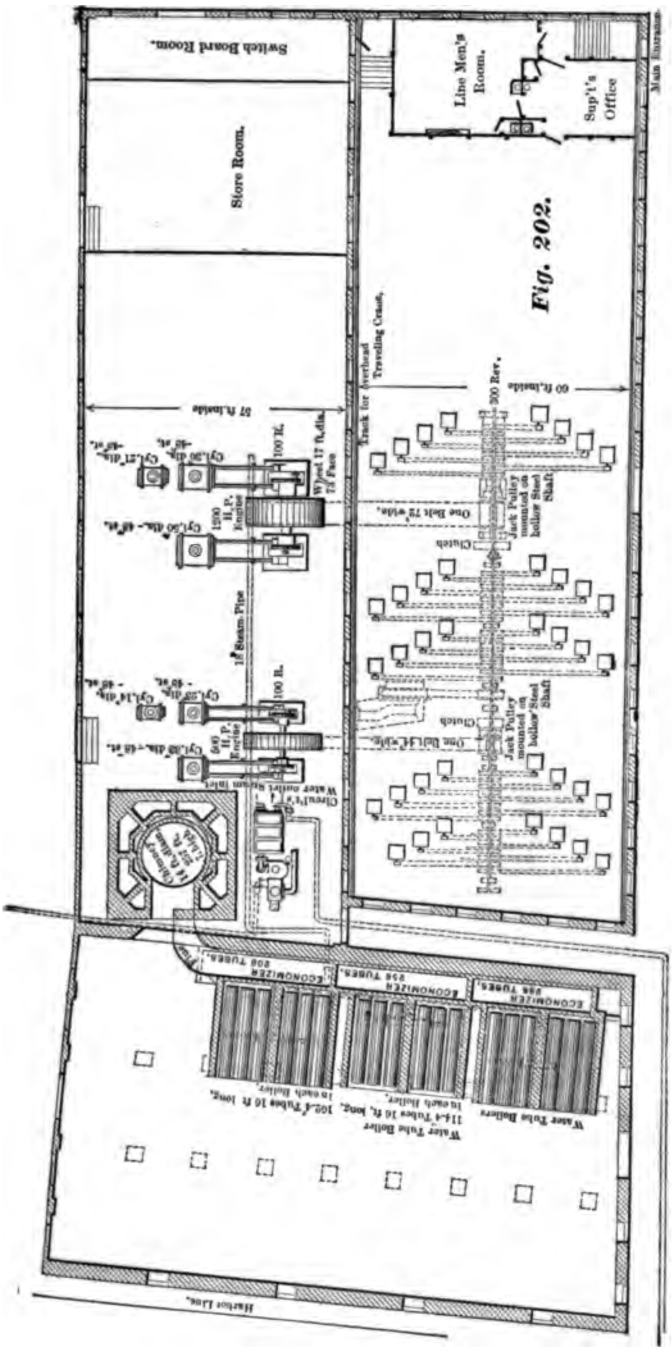

Switch Board Room.
Store Room.
Line Men's Room.
Sup't's Office
Main Entrance
Track for Overhead Traveling Crane.
60 ft. inside
57 ft. inside
Jack Pulley mounted on hollow Steel Shaft
Clutch
Economizer
Water Tube Boilers
Harbor Line.

Fig. 202.

a valve, and so a steady and constant flow of oil is maintained at all places requiring lubrication. This same system is in use on our

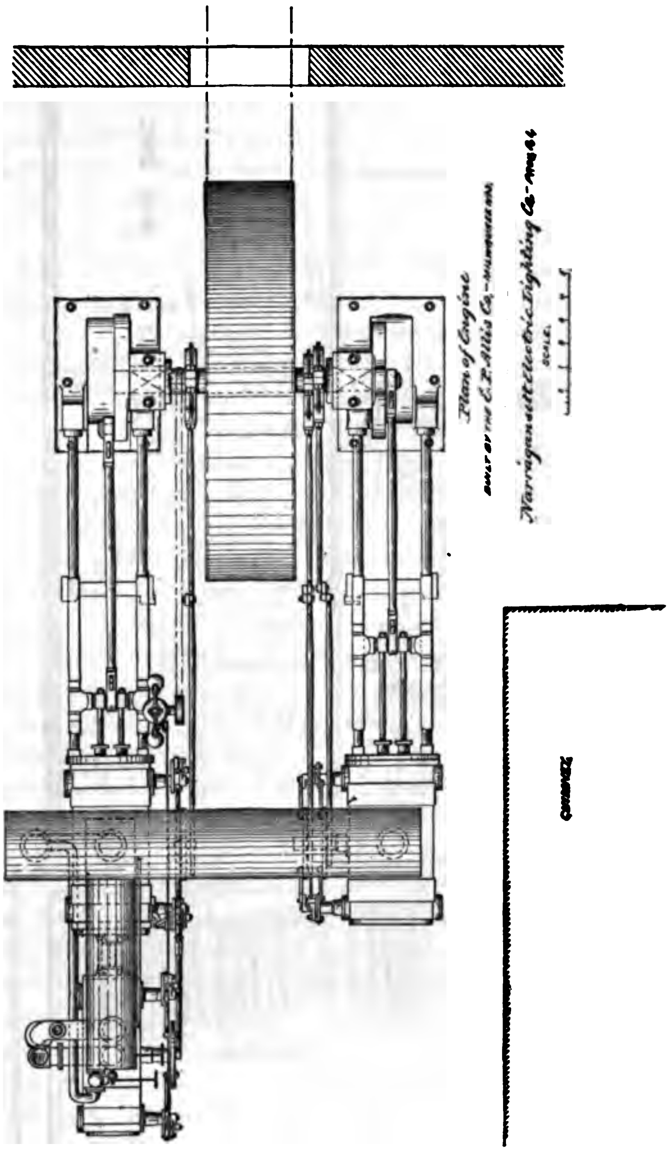
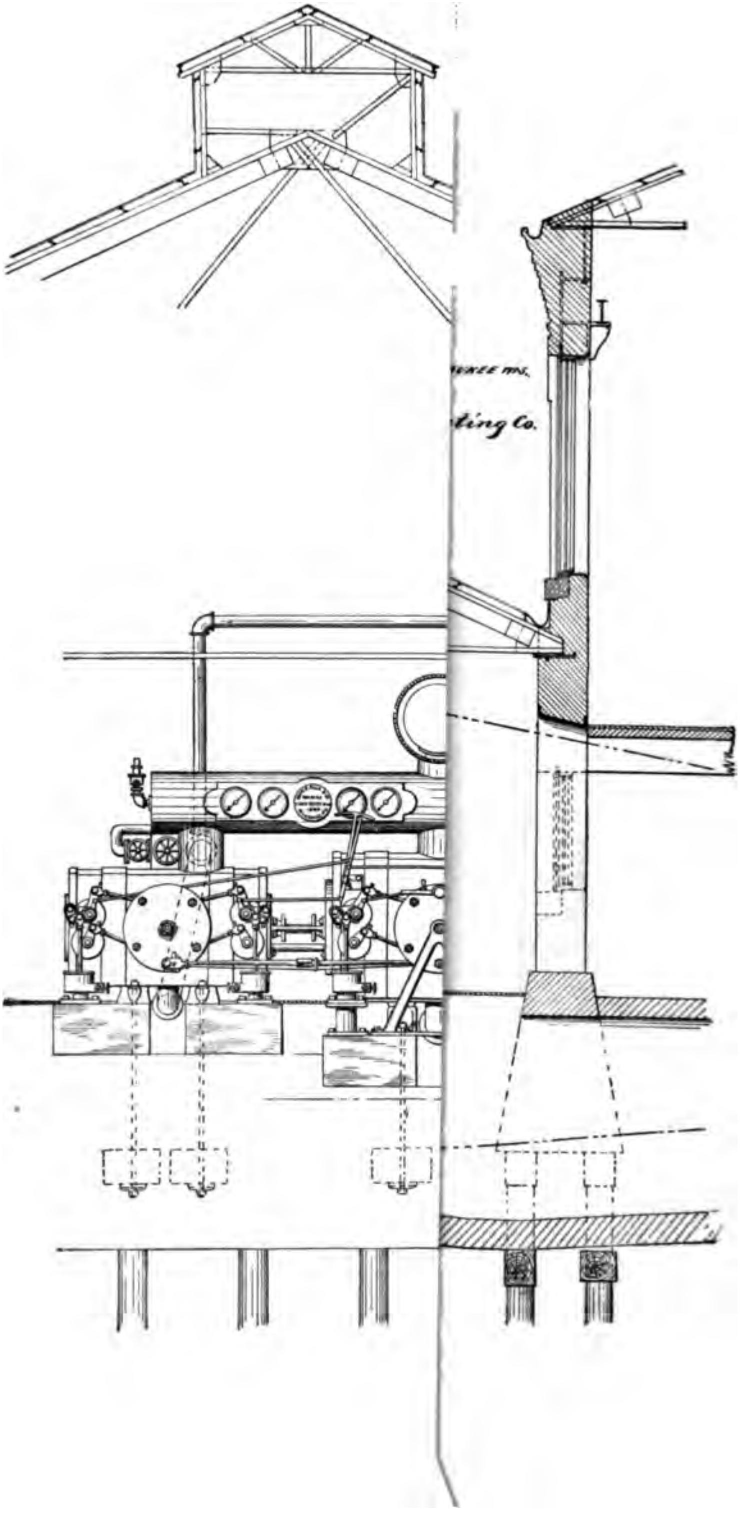


Fig. 203.

large engine (1,200 horse-power) of the same type and build, and which runs at the same speed, 100 revolutions per minute, with very satisfactory results.



In connection with this paper a general plan of the electric station, where the engines are in daily operation, is given in Fig. 202, which plan shows the location of the engines, boilers, chimneys, etc., and their connections. There is also given a plan and elevation of the engine, Figs. 203 and 204, which plainly show the construction and general arrangement of the parts.

METHOD OF TEST.

It was found inexpedient to measure the feed-water on account of leakage from the boilers, and, therefore, arrangements were made to weigh the water discharged from the engine after passing through the surface condenser. The ordinary hot-well discharge was blanked off from its regular course, and the necessary pipes and valves fitted to discharge the water from the surface condenser into two weigh-tanks alternately. The water, after being drained from each of the alternate weigh-tanks, was then taken by the feed-pump and discharged through a meter as a check on the weighing, when it finally passed to the river. The temperature of the hot-well water was taken at the time of each weighing of the water tank.

The jacket-water pump of the engine was disconnected, and during the test the drainage from the receivers and jackets was regulated by hand, so as to keep the water at a given height, as shown by a glass gauge in the system of jacket piping, to prevent steam from being discharged with the jacket water. Two weigh-tanks were provided for measuring this water. Each tank was partially filled with cold water from the city mains, and its weight and temperature noted. The condensed steam from the jackets and receivers was then allowed to run in until the tank was nearly full, when it was turned into the other tank, which had previously been supplied with cold water, and the weight and temperature of the mass again noted. The water was then emptied into the tank from which the feed-pump drew its supply, and was passed through the meter with the hot-well water.

As the condensing water, drawn from the river, was salt water, it afforded a means of detecting any leakages which might be in the surface condenser from the salt-water side. Therefore, the hot-well water was frequently tested by nitrate of silver, but no leakage was found during the test. The condenser was also tested previous to the engine test and found to be perfectly tight.

Indicators were put on all steam cylinders of main and auxiliary engines, one on each end of each cylinder, and all connected as close to the cylinders as possible, all pipe connections being direct, without bends or angle cocks. The motion for indicators was taken from the engine cross-heads by pantagraphs, and their accuracy of motion established before the test was commenced. The six indicators on the main engine were electrically connected, so that all cards of each set were taken simultaneously. An electric current from batteries was employed to bring the pencil point of the indicator motion up to the paper and to make the contact with it upon the drum through the agency of a suitable magnet mounted upon the instrument. All indicator springs used on the engine were tested under steam, both before and after the engine test, and the mean of the two scales taken. The springs used on the small engine were similarly tested, but only once. All springs were tested by a test steam-gauge, which itself had immediately before been tested by a mercury column.

Previous to the engine test the steam and vacuum gauges were tested and found to be correct. Subsequent to testing the engine, however, the vacuum gauge got out of order, so that its readings were incorrect. It was, consequently, tested after the engine test, and its correction for the pressures noted was found to be an average of 1 inch. The steam-gauge at the engine was compared, while the engine was standing, with the test-gauge used with the calorimeter experiments. The application of the resulting correction did not make the readings agree with the test-gauge, probably on account of the pipe leading to the gauge not having a full head of water in it when the engine was running. The average reading of the test-gauge is therefore given in the synopsis of results, instead of the average from the regular engine gauge.

A half-inch perforated brass pipe was inserted horizontally in the vertical main steam-pipe close to the engine throttle, through which steam was collected for test by calorimeter (Fig. 205), and all such pipes leading to the calorimeter were carefully covered by hair felting. The calorimeter was of the "barrel" type, made of galvanized iron, with double walls separated about three inches and the intervening space insulated by hair felt. It was provided with a set of rotating buss paddles mounted on a brass rod for equalizing the temperature of the water (Fig. 206). The steam entered horizontally through the side of the tank, thence down the side, and finally into a half-inch brass pipe coiled in the bot-

Fig. 206.

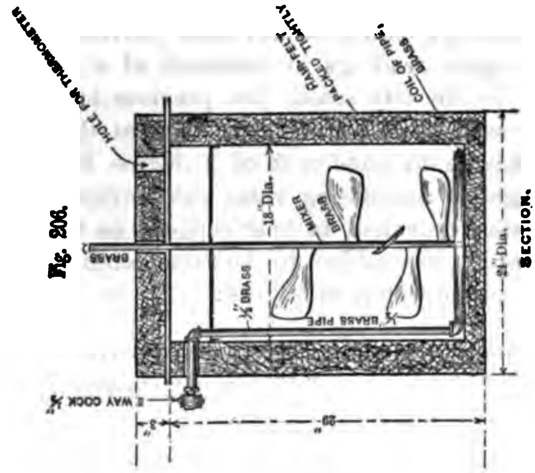
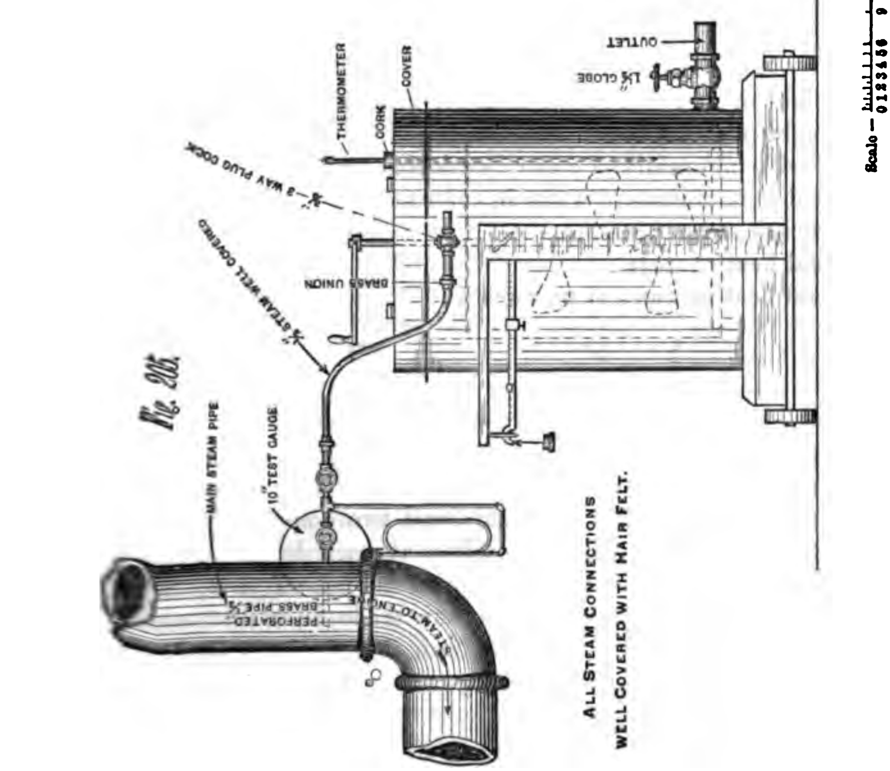


Fig. 205.



tom of the barrel, and to which it was permanently secured. The barrel was mounted on a platform scale provided with a jockey weight which gave readings to one-tenth of a pound, the scale being sealed by the city sealer just previous to the test. The thermometer used was a reliable one tested at the Kew Observatory, and readable to one-fourth of a degree Fahrenheit. The observed weight of condensing water was corrected to include a weight of water equivalent in heat capacity to the metal of the calorimeter, which was subject to the same conditions, changes of temperature, as the water, as follows:

14.56 lbs. iron × specific heat 0.094	1.369
6.07 " brass × " " 0.110	.668
Equivalent weight of water	2.037 lbs.

For determining the percentage of moisture in the steam furnished the engine, the following formula is used:

Where

W, Weight of cold water (corrected for heat capacity of barrel).
w, Weight of steam blown in.
t, Initial temperature of water in calorimeter.
t_1, Final temperature of water in calorimeter.
P, Average steam pressure.
H, Total heat of steam.
T, Total heat in water at pressure P.

$$\frac{1}{H-T}\left[\frac{W}{w}(t_1-t)-(T-t_1)\right]$$

All preparations were made for testing the engine on July 13, and on the afternoon of that day the engine was loaded successively with 400, 450, and 500 arc lights, and two complete sets of indicator cards from all cylinders taken at each load. A long pointer was attached to the high-pressure cut-off arm, and when the above load was on, marks were made on a stationary board, so that variations of the loads from that desired could be detected. The horse-power shown by the various cards was as follows:

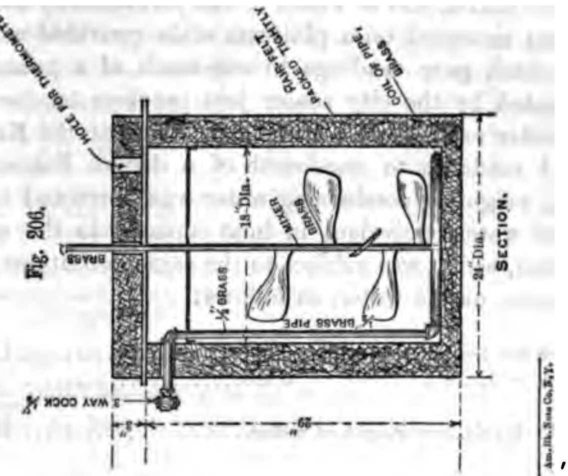


Fig. 206.

SECTION.

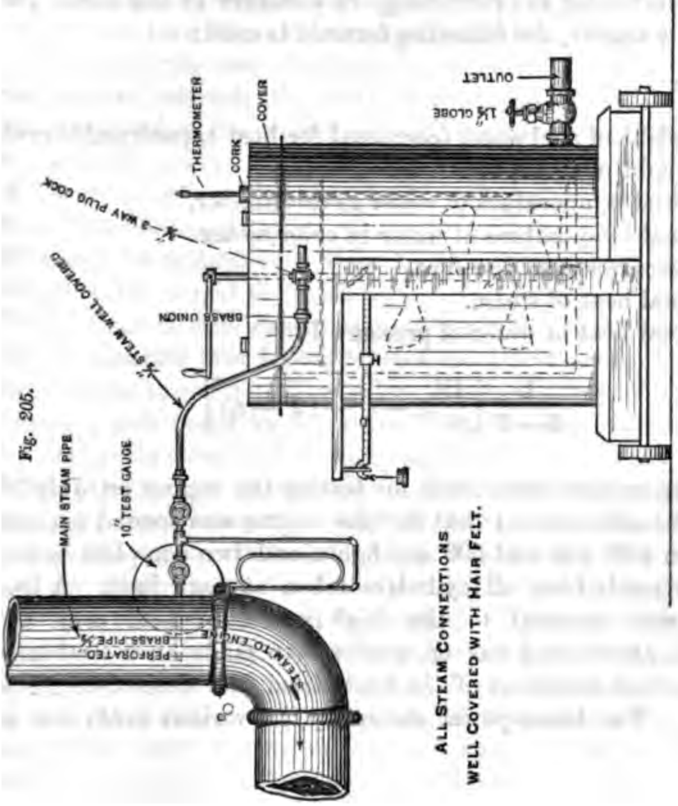


Fig. 205.

ALL STEAM CONNECTIONS WELL COVERED WITH HAIR FELT.

Scale — 0 1 2 3 4 5 6 9 12

tom of the barrel, and to which it was permanently secured. The barrel was mounted on a platform scale provided with a jockey weight which gave readings to one-tenth of a pound, the scale being sealed by the city sealer just previous to the test. The thermometer used was a reliable one tested at the Kew Observatory, and readable to one-fourth of a degree Fahrenheit. The observed weight of condensing water was corrected to include a weight of water equivalent in heat capacity to the metal of the calorimeter, which was subject to the same conditions, changes of temperature, as the water, as follows:

14.56 lbs. iron × specific heat 0.094	1.369
6.07 " brass × " " 0.110	.668
Equivalent weight of water	2.037 lbs.

For determining the percentage of moisture in the steam furnished the engine, the following formula is used:

Where

W, Weight of cold water (corrected for heat capacity of barrel).
w, Weight of steam blown in.
t, Initial temperature of water in calorimeter.
t_1, Final temperature of water in calorimeter.
P, Average steam pressure.
H, Total heat of steam.
T, Total heat in water at pressure P.

$$\frac{1}{H-T}\left[\frac{W}{w}(t_1-t)-(T-t_1)\right]$$

All preparations were made for testing the engine on July 13, and on the afternoon of that day the engine was loaded successively with 400, 450, and 500 arc lights, and two complete sets of indicator cards from all cylinders taken at each load. A long pointer was attached to the high-pressure cut-off arm, and when the above load was on, marks were made on a stationary board, so that variations of the loads from that desired could be detected. The horse-power shown by the various cards was as follows:

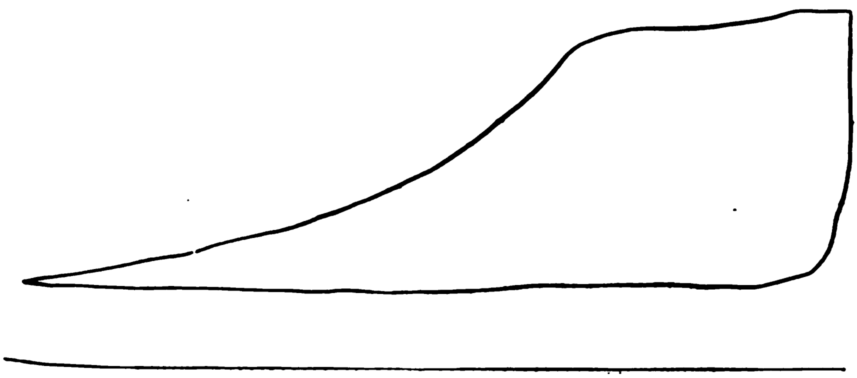


FIG. 208.

No. 10.—July 14, 1890. 12:50 A.M.
High-pressure cylinder, back end.

Diameter cylinder	14 inches
Stroke	48 inches
Throttle	5 inches
Indicator scale	63.93
Calculations :	
Area	3.15
Length	4.40
M. E. P	45.71
I. H. P	84.4

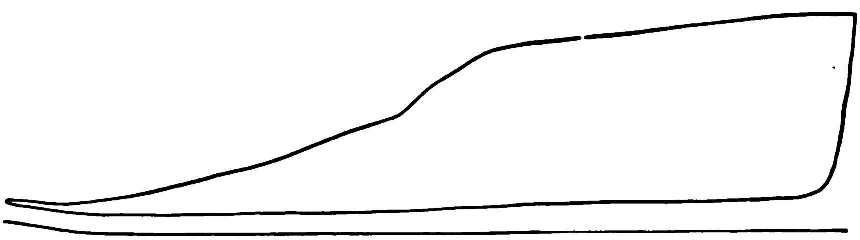


FIG. 209.

No. 10.—July 14, 1890. 12:50 A.M.
Intermediate pressure cylinder, crank end.

Diameter cylinder	25 inches
Stroke	48 inches
Indicator scale	23.39
Calculations :	
Area	2.37
Length	4.45
M. E. P	12.44
I. H. P	71.4

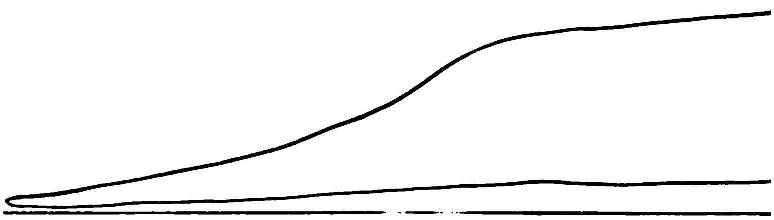


FIG. 210.

No. 10.—July 14, 1890. 12:50 A.M.
Intermediate pressure cylinder, back end.
Diameter cylinder........................ ....25 inches
Stroke..48 inches
Indicator scale 23.43
Calculations :
  Area.. 2.27
  Length...................................... 4.45
  M. E. P.....................................11.95
  I. H. P.....................................69.5

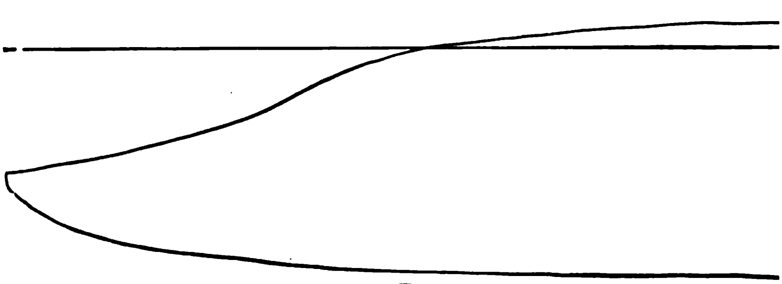


FIG. 211.

No. 10.—July 14, 1890. 12:50 A.M.
Low-pressure cylinder, back end.
Diameter cylinder..............................38 inches
Stroke...48 inches
Indicator scale 10.16
Calculations:
  Area 4.52
  Length...................................... 4.60
  M. E. P..................................... 9.976
  I. H. P......... ....102.3

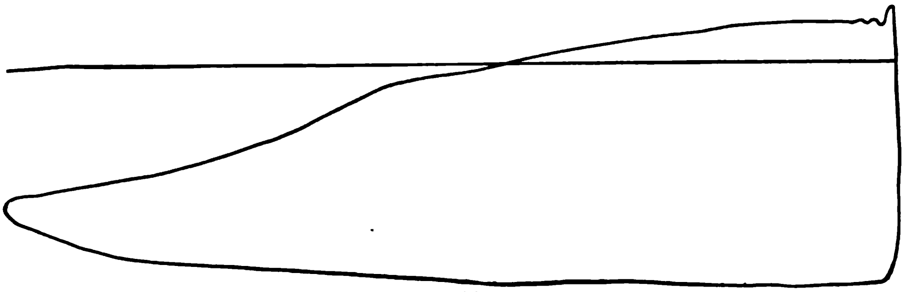


FIG. 212.

No. 10.—July 14, 1890. 12:50 A.M.
Low-pressure cylinder, crank end.

Diameter cylinder	33 inches
Stroke	48 inches
Indicator scale	10.65
Calculations:	
Area	4.33
Length	4.65
M. E. P	9.894
I. H. P	100.0

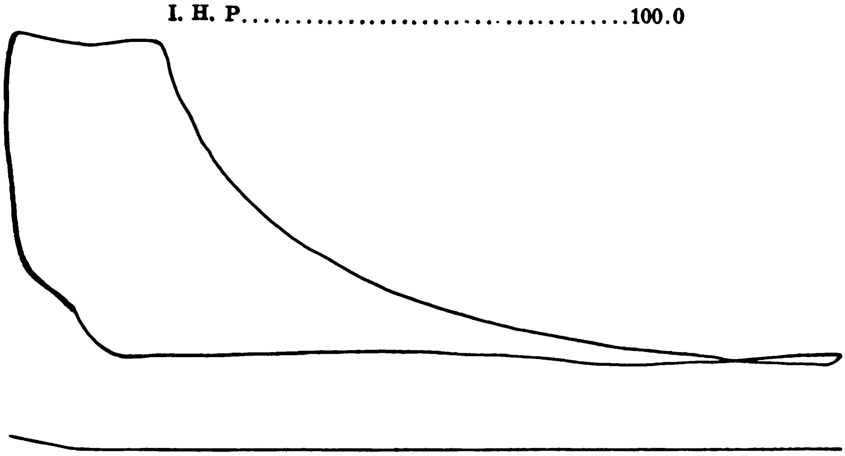


FIG. 213.

No. 10.—July 14, 1890. 12:50 A.M.
Condensing cylinder, bottom end.

Diameter cylinder	12 inches
Stroke	16 inches
Revolutions	61
Indicator scale	57.82
Calculations:	
Area	2.11
Length	4.20
M. E. P	29.02
I. H. P	8.1

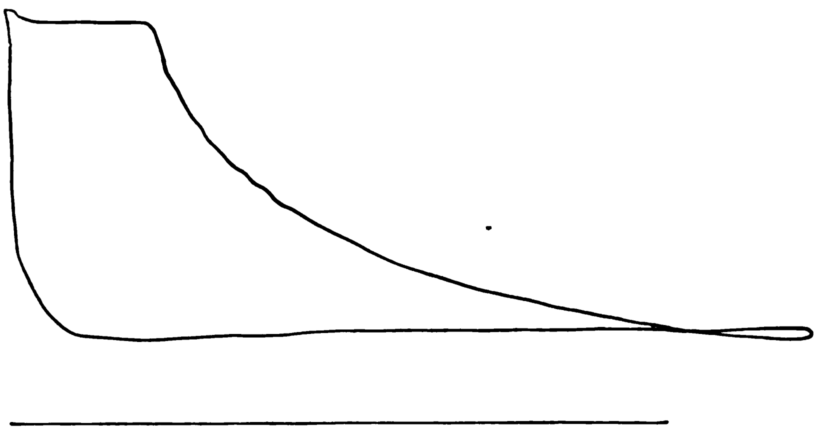


FIG. 214.

No. 10.—July 14, 1890. 12:50 A.M.
Condensing cylinder, top end.

Diameter cylinder	12 inches
Stroke	16 inches
Revolutions	61
Indicator scale	56.42
Calculations:	
Area	2.41
Length	4.15
M. E. P	32.72
I. H. P	8.8

In determining the duty of this engine as given in the following general summary, the power developed in the engine of the condensing apparatus and that of the main engine has been used collectively, and thus treated as one engine; and this factor, when divided into the total water, which includes all that discharged from the surface condenser, and that also from the two receivers and the three cylinder jackets, gives as a result 12.94 lbs. of water per indicated horse-power; this is accomplished with exceedingly wet steam and of 125 lbs. pressure only.

The engines were originally designed, however, to carry a working steam pressure of 160 lbs., as that was fixed by the mechanical engineers of the electric station as the ultimate pressure for all machines. And I feel quite confident that, with that pressure and dry steam, only 12.6 lbs. of water would be required per indicated horse-power.

GENERAL DIMENSIONS OF ENGINE.

Diameter of fly-wheel	17 ft.
Width of pulley face	45 in.
Width of belt	44 "
Diameter of crank-pins	5 "
Length of crank-pins	6 "
Diameter of main bearing	10 "
Length of main bearing	18 "
Centre to centre of engines	12 ft.

14-INCH CYLINDER.

Steam-ports (2)	$\frac{7}{8}$ x $9\frac{1}{4}$ in.
Exhaust-port (1)	$3\frac{1}{8}$ x $9\frac{1}{4}$ "
Diameter of steam-ports	6 "
Diameter of exhaust-ports	7 "
Thickness steam space in jacket	1 "
Diameter piston rod	$2\frac{13}{16}$ "
Diameter steam inlet	5 "
Diameter exhaust outlet	7 "

25-INCH CYLINDER.

25" cylinder steam-ports (2)	$1\frac{1}{4}$ x 18 in.
" " exhaust-port (1)	$8\frac{1}{16}$ x 18 "
" " diameter steam and exhaust ports	7 "
" thickness of steam space in jacket	1 "
Diameter piston rods	$2\frac{13}{16}$ "
Diameter steam inlet	8 "
Diameter exhaust outlet	10 "

33-INCH CYLINDER.

Steam-ports (2)	$1\frac{1}{4}$ x $27\frac{1}{4}$ in.
Exhaust-port (1)	$3\frac{1}{8}$ x $27\frac{1}{4}$ "
Diameter steam-ports	6 "
Diameter exhaust	7 "
Thickness of steam space in jacket	1 "
Diameter of piston rods	$2\frac{13}{16}$ "
Diameter steam inlet	10 "
Diameter exhaust outlet	12 "

RESULT OF TEST.

Test began at	9.30 P.M.
Test ended at	7.30 A.M.
Duration of test	10 hrs. 0 m.
Revolutions of main engine, total	59,376
" " " " maximum per minute	99.33
" " " " minimum " "	99.10
" " " " average " "	99.12
" " condenser engine, average per minute	61.29

Steam pressure at engine throttle, per gauge, maximum	128 lbs.
" " " " " " " minimum	110 "
" " " " " " " average	125.2 "
" " " first receiver, " " "	26.98 "
" " " second " " " "	1.70 "
Vacuum at condenser, maximum	26.7 in.
" " " minimum	25.8 "
" " " average	26.5 "
Barometer, average	30.08 "
Temperature of injection water, average	72.00° F.
" " discharge " "	86.06° F.
" " hot-well " "	109.06° F.
" " engine-room, "	90.00° F.
Indicated horse-power, main engine, maximum	529.7
" " " " minimum	502.2
" " " " average	515.70
" " condenser" maximum	17.6
" " " " minimum	14.8
" " " " average	16.41
" " aggregate "	532.11
Water discharged from hot-well, total	63,684 lbs.
" " " jackets, "	10,653 "
" " " engine, "	74,337 lbs.
Percentage of total steam used in jackets	14.33%
Water entrained in steam, per calorimeter test	7.39%
" " " " " " "	5,497 lbs.
Net steam used by engine	68,840 lbs.
Steam per indicated horse-power per hour	12.94 lbs.

TABLE NO. I.

RECORD OF EXPERIMENTS WITH A TRIPLE-EXPANSION ENGINE AT THE NARRAGANSETT STATION, PROVIDENCE, R. I., JULY 13 AND 14, 1891.

ENGINE-ROOM DATA.

Hour.	Pressures.								Temperatures.		
	Steam at throttle. Cor. = − 3.		Steam at first receiving. Cor. = − ½.		Steam at second receiving. Cor. = −1.75.		Vacuum. Cor. = − 1″.				
	Observed.	Actual.	Observed.	Actual.	Observed.	Actual.	Observed.	Actual.	Injection water.	Discharge water.	Engine-room.
P.M.											
9:30	125	122.0	27.0	26.5	3.5	1.75	26.8	25.8			
9:45	125	122.0	26.5	26.0	3.5	1.75	27.4	26.4	72	86.0	93.0
10:00	123	120.0	27.5	27.0	3.5	1.75	27.5	26.5	72	86.0	93.0
10:15	126	123.0	27.0	26.5	3.5	1.75	27.4	26.4	72	86.0	93.0
10:30	126	123.0	27.0	26.5	3.4	1.65	27.4	26.4	72	86.0	93.0
10:50	126	123.0	27.5	27.0	3.5	1.75	27.4	26.4	72	86.0	93.0
11:10	125	122.0	27.5	27.0	3.5	1.75	27.5	26.5	72	86.5	93.0
11:30	125	122.0	27.0	26.5	3.4	1.65	27.5	26.5	72	86.5	92.0
11:50	125	122.0	27.0	26.5	3.4	1.65	27.5	26.5	72	86.0	91.0
A.M.											
12:10	125	122.0	27.5	27.0	3.3	1.55	27.4	26.4	72	86.0	91.0
12:30	112	109.0	29.0	28.5	4.0	2.25	27.4	26.4	72	86.5	90.5
12:50	126	123.0	27.4	26.9	3.5	1.75	27.4	26.4	72	86.5	92.0
1:10	125	122.0	27.5	27.0	3.5	1.75	27.5	26.5	72	86.0	91.0
1:30	127	124.0	27.5	27.0	3.4	1.65	27.5	26.5	72	86.0	90.0
1:50	126	123.0	27.5	27.0	3.4	1.65	27.5	26.5	72	86.0	91.0
2:10	126	123.0	27.5	27.0	3.4	1.65	27.5	26.5	72	86.0	90.0
2:30	127	124.0	28.0	27.5	3.5	1.75	27.5	26.5	72	86.0	90.0
2:50	129	126.0	27.0	26.5	3.2	1.45	27.5	26.5	72	86.0	90.0
3:10	128	125.0	27.0	26.5	3.0	1.25	27.5	26.5	72	86.5	90.0
3:30	128	125.0	27.5	27.0	3.5	1.75	27.5	26.5	72	86.0	90.0
3:50	128	125.0	28.0	27.5	3.5	1.75	27.5	26.5	72	86.5	91.0
4:10	127	124.0	28.0	27.5	3.9	2.15	27.5	26.5	72	86.0	91.0
4:30	128	125.0	28.5	28.0	3.6	1.85	27.6	26.6	72	86.0	89.0
4:50	128	125.0	28.5	28.0	3.6	1.85	27.6	26.6	72	86.0	90.0
5:10	127	124.0	27.5	27.0	3.5	1.75	27.6	26.6	72	86.0	88.0
5:30	127	124.0	27.5	27.0	3.5	1.75	27.6	26.6	72	86.0	87.0
5:50	127.5	124.5	27.5	27.0	3.5	1.75	27.6	26.6	72	86.0	86.0
6:10	128	125.0	27.5	27.0	3.4	1.65	27.6	26.6	72	86.0	86.0
6:30	129	126.0	27.5	27.0	3.4	1.65	27.7	26.7	72	86.0	87.0
6:50	127	124.0	27.5	27.0	3.4	1.65	27.7	26.7	72	86.0	87.0
7:10	127	124.0	27.0	26.5	3.1	1.35	27.7	26.7	72	85.0	87.0
7:30	127	124.0	27.0	26.5	3.1	1.35	27.7	26.7	72	86.0	87.0
Av.....		123.1		26.92		1.70		26.5	72	86.06	90.08

TABLE NO. II.

RECORD OF EXPERIMENTS WITH A TRIPLE-EXPANSION ENGINE AT THE NARRAGANSETT STATION, PROVIDENCE, R. I., JULY 13 AND 14, 1890.

REVOLUTIONS.

Main Engine.										Condensing Engine.
First Counter.					Second counter.					
Hour.	Reading of counter.	Difference.	Time interval.	Revolutions per minute.	Hour.	Reading of counter.	Difference.	Time interval.	Revolutions per minute.	Revolutions per minute.
9:30					9:31					62.0
9:45	1490	1490	15	99.33	9:46	1489	1489	15	99.26	62.0
10:00	2975	1485	15	99.00	10:02	3073	1584	16	99.00	61.0
10:15	4462	1487	15	99.13	10:16	4461	1388	14	99.14	61.5
10:30	5949	1487	15	99.13	10:31	5948	1487	15	99.13	61.0
10:50	7931	1982	20	99.10	10:51	7930	1982	20	99.10	61.0
11:10	9914	1983	20	99.15	11:11	9913	1983	20	99.15	61.0
11:30	11895	1981	20	99.05	11:31	11894	1981	20	99.05	61.0
11:50	13876	1981	20	99.05	11:51	13876	1982	20	99.10	61.5
12:10	15858	1982	20	99.10	12:11	15858	1982	20	99.10	61.5
12:30	17839	1981	20	99.05	12:31	178[illegible]	1980	20	99.00	61.0
12:50	19818	1979	20	98.95	12:51	19818	1980	20	99.00	61.0
1:10	21800	1982	20	99.10	1:11	21800	1982	20	99.10	61.5
1:30	23782	1982	20	99.10	1:31	23781	1981	20	99.05	61.5
1:50	25765	1983	20	99.15	1:51	25764	1983	20	99.15	61.0
2:10	27748	1983	20	99.15	2:11	27747	1983	20	99.15	61.0
2:30	29730	1982	20	99.10	2:31	29729	1982	20	99.10	61.0
2:52	31913	2183	22	99.09	2:51	31713	1984	20	99.20	61.0
3:10	33699	1786	18	99.23	3:11	33698	1985	20	99.25	61.5
3:30	35681	1982	20	99.10	3:31	35680	1982	20	99.10	61.0
3:50	37665	1984	20	99.20	3:51	37664	1984	20	99.20	61.5
4:10	39648	1983	20	99.15	4:11	39646	1982	20	99.10	61.0
4:30	41630	1982	20	99.10	4:31	41629	1983	20	99.15	61.5
4:50	43612	1982	20	99.10	4:51	43611	1982	20	99.10	61.5
5:10	45595	1983	20	99.15	5:11	45594	1983	20	99.15	61.5
5:30	47577	1982	20	99.10	5:31	47576	1982	20	99.10	61.0
5:50	49558	1981	20	99.05	5:51	49557	1981	20	99.05	61.5
6:10	51543	1985	20	99.25	6:11	51542	1985	20	99.25	61.5
6:30	53525	1982	20	99.10	6:31	53524	1982	20	99.10	61.5
6:50	55509	1984	20	99.20	6:51	55508	1984	20	99.20	61.5
7:10	57493	1984	20	99.20	7:11	57492	1984	20	99.20	61.5
7:29	59376	1883	19	99.15	7:30	59374	1882	19	99.05	61.0
Average . . .				99.12					99.12	61.29

TABLE NO. III.

RECORD OF EXPERIMENTS WITH A TRIPLE-EXPANSION ENGINE AT THE NARRAGANSETT STATION, PROVIDENCE, R. I., JULY 13 AND 14, 1890.

WEIGHT OF WATER DISCHARGED FROM HOT-WELL.

Tank No. 1.					Tank No. 2.				
Time when full.	Weight empty.	Weight full.	Net weight of water.	Temp.	Time when full.	Weight empty.	Weight full.	Net weight of water.	Temp.
P.M.									
9:43	213	946	733	110	9:36	265	911	646	105
9:57	213	938	725	110	9:50	259	937	678	109
10:10	215	947	732	110	10:03	256	954	698	110
10:24	214	950	736	110	10:17	262	967	705	110
10:38	217	944	727	110	10:31	262	980	718	110
10:52	212	986	774	110	10:45	259	968	709	110
11:06	212	985	773	110	10:58	258	966	708	110
11:20	213	929	716	110	11:12	258	972	714	110
11:33	213	991	778	110	11:26	261	979	718	110
11:47	212	950	738	110	11:41	260	979	719	110
12:00	216	952	736	110	11:54	259	971	712	110
12:15	211	964	753	110	12:07	255	967	712	110
12:27	213	938	725	110	12:21	256	970	714	110
12:43	213	980	767	110	12:35	256	968	712	110
12:56	213	965	752	110	12:49	258	976	718	110
1:10	213	922	709	110	1:03	260	971	711	110
1:25	213	939	726	110	1:17	260	982	722	110
1:38	213	966	753	109	1:32	260	1020	760	110
1:51	213	925	712	109	1:45	260	1025	765	109
2:05	213	933	720	109	1:59	260	974	714	109
2:20	213	942	729	109	2:12	260	980	720	109
2:33	213	975	762	109	2:26	260	1009	749	109
2:47	213	952	739	109	2:40	260	972	712	109
3:00	213	932	739	109	2:53	260	998	738	109
3:14	213	935	722	109	3:07	260	974	714	109
3:29	213	1010	797	109	3:21	260	977	717	109
3:43	213	946	733	109	3:35	260	983	723	109
3:56	213	966	753	109	3:49	259	979	720	109
4:10	213	969	756	109	4:03	260	1050	790	109
4:25	213	974	761	109	4:17	260	977	717	109
4:38	213	976	763	109	4:30	260	977	717	109
4:51	213	946	733	108	4:45	260	976	716	109
5:05	213	945	732	108	4:58	260	989	729	108
5:19	213	961	748	108	5:12	260	985	725	108
5:32	213	955	742	108	5:25	260	974	714	108
5:46	213	943	730	108	5:40	260	1021	761	108
6:00	213	974	761	108	5:55	260	976	716	108
6:15	213	941	728	108	6:08	260	992	732	108
6:29	213	940	727	108	6:22	260	1049	789	108
6:42	213	927	714	108	6:35	260	981	721	108
6:56	213	940	727	108	6:49	260	991	731	108
7:10	213	957	744	108	7:03	260	1072	812	108
7:25	220	944	724	108	7:18	259	998	739	108
					7:30	259	930	671	108
Totals.........	9171	41020	31849	4692		11421	43256	31835	4796

Total water from hot-well = 63684 lbs.
Average temperature of same = 109.06° Fahr.

TABLE NO. IV.

RECORD OF EXPERIMENTS WITH A TRIPLE-EXPANSION ENGINE AT THE NARRAGANSETT STATION, PROVIDENCE, R. I., JULY 13 AND 14, 1890.

WEIGHT OF JACKET WATER.

Tank No. 3.								Tank No. 4.							
Time when full.	Weight with cold water.	Temp. of cold water.	Weight with jacket water.	Temp. with jacket water.	Weight empty.	Net weight cold water.	Net weight jacket water.	Time when full.	Weight with cold water.	Temp. of cold water.	Weight with jacket water.	Temp. with jacket water.	Weight empty.	Net weight cold water.	Net weight jacket water.
9:47	341.0	78	500.0	166	91.0	250.0	159.0	9:38	375.0	76	495.0	150	95.0	280.0	120.0
10:04	325.0	82	498.0	182	90.0	235.0	173.0	9:55	350.0	78	490.0	160	99.5	250.5	140.0
10:23	323.0	81	500.0	178	96.0	227.0	177.0	10:12	327.0	78	487.0	177	98.5	228.5	160.0
10:42	327.0	84	580.0	179	89.0	238.0	173.0	10:34	371.5	78	525.0	162	108.0	263.5	158.5
11:00	318.0	82	485.0	174	92.5	225.5	167.0	10:51	301.0	82	465.0	184	99.5	201.5	164.0
11:20	316.0	84	502.0	182	92.5	223.5	186.0	11:10	344.5	80	513.5	172	99.0	245.5	169.0
11:42	309.5	82	500.0	184	89.5	220.0	190.5	11:30	341.0	82	515.5	172	99.0	242.0	174.5
11:59	336.0	82	495.5	170	88.0	248.0	159.5	11:51	311.5	80	483.0	180	99.0	212.5	171.5
12:17	306.0	81	476.0	176	90.0	216.0	170.0	12:07	313.0	80	371.0	174	99.5	218.5	58.0
12:36	309.0	82	477.0	170	108.0	201.0	168.0	12:25	354.0	80	493.5	162	98.0	256.0	189.5
12:55	260.5	94	456.0	205	90.0	170.5	195.5	12:44	360.5	76	511.0	164	102.5	258.0	150.5
1:14	328.0	82	494.0	174	90.5	237.5	166.0	1:08	331.0	80	476.0	168	97.5	233.5	145.0
1:33	313.0	93	500.0	181	91.0	222.0	187.0	1:25	346.0	78	520.0	172	155.0	191.0	174.0
1:50	375.0	84	504.0	156	92.5	282.5	129.0	1:42	325.0	98	504.0	192	155.5	169.5	179.0
2:09	291.0	86	482.0	188	90.0	201.0	191.0	2:00	320.0	102	485.5	194	104.0	216.0	165.5
2:27	310.0	82	500.0	183	89.0	221.0	190.0	2:22	340.5	82	492.0	168	253.0	87.5	151.5
2:46	307.0	84	485.5	182	90.0	217.0	178.5	2:37	366.0	120	514.0	190	126.5	239.5	148.0
3:05	322.0	82	492.0	176	88.5	233.5	170.0	2:54	324.0	94	496.0	183	101.0	223.0	172.0
3:22	318.5	80	498.0	176	94.0	224.5	169.5	3:14	340.0	79	502.0	172	124.5	215.5	162.0
3:41	309.0	82	492.0	181	92.0	217.0	183.0	3:32	336.0	86	479.0	170	101.5	234.5	143.0
3:59	305.0	83	490.0	187	92.5	212.5	185.0	3:50	332.5	79	491.5	173	99.0	233.5	159.0
4:18	323.0	82	495.0	178	94.5	228.5	172.0	4:09	343.5	80	504.5	170	99.5	244.0	161.0
4:37	320.0	80	496.0	179	90.0	230.0	176.0	4:28	309.0	80	473.5	180	99.5	209.5	164.5
4:54	302.0	82	480.5	185	89.0	213.0	178.0	4:45	326.5	83	489.5	172	98.5	228.0	163.0
5:15	309.0	80	495.0	182	91.0	218.0	186.0	5:05	324.5	80	487.0	175	98.0	226.5	162.5
5:33	322.5	80	501.0	169	92.0	230.5	178.5	5:23	341.0	78	501.0	170	99.0	242.0	160.0
5:52	322.0	82	499.0	176	91.5	230.5	177.0	5:43	318.0	78	493.0	180	97.5	220.5	173.0
6:11	293.5	83	471.5	186	96.0	197.5	178.0	6:02	331.0	78	498.5	174	99.0	232.0	167.5
6:29	312.0	84	495.0	184	94.0	218.0	183.0	6:19	363.0	76	507.0	160	98.0	265.0	144.0
6:43	316.0	82	487.0	179	95.0	221.0	171.0	6:37	335.0	77	485.0	168	99.0	236.0	150.0
7:05	303.5	82	487.0	186	94.0	209.5	183.5	6:54	333.0	78	491.0	172	99.0	234.0	158.0
7:22	310.0	82	485.0	181	90.0	220.0	175.0	7:18	336.0	80	476.5	164	99.0	237.0	140.5
								7:30	321.0	78	402.5	186	95.0	226.0	81.5
Totals........	10083.0	2648	15709.0	5735	2948.5	7139.5	5626.0		11091.5	2718	16136.5	5660	3596.0	7495.5	5087.0

TABLE NO. V.

RECORD OF EXPERIMENTS WITH A TRIPLE-EXPANSION ENGINE AT THE NARRAGANSETT STATION, PROVIDENCE, R. I., JULY 13 AND 14, 1890.

TABLE OF CALORIMETER TESTS.

Time.	Weight of cold water corrected for heat capacity of barrel = W.	Weight of steam blown in = w.	Steam pressure at beginning.	Steam pressure at end.	Average steam pressure = P.	Initial temperature of water = t.	Final temperature of water = t_1.	Total heat in water at pressure $P = T$.	Total heat of steam = H.	$t_1 - t$	$T - t_1$	$H - T$	Percentage of dry steam $= \frac{1}{H-T}\left[\frac{W}{w}(t_1 - t) - (T - t_1)\right]$.	Percentage of moisture.
P.M. 9:45	165.0	8.4	125	125	125.00	73.25	125.00	356.0	1221.6	51.75	231.00	865.6	$\frac{1}{865.6}\left(\frac{165.0 \times 51.75}{8.4} - 231.00\right) = 90.75$	9.25
10:15	162.4	6.7	124	124	124.00	73.25	115.00	355.4	1221.4	41.75	240.40	866.0	$\frac{1}{866.0}\left(\frac{162.4 \times 41.75}{6.7} - 240.40\right) = 89.07$	10.93
10:45	172.1	8.4	125	124	124.50	73.25	124.00	355.7	1221.5	50.75	231.70	865.8	$\frac{1}{865.8}\left(\frac{172.1 \times 50.75}{8.4} - 231.70\right) = 93.32$	6.68
11:15	175.7	9.3	125	125	125.00	72.75	125.75	356.0	1221.6	53.00	230.25	865.6	$\frac{1}{865.6}\left(\frac{175.7 \times 53.00}{9.3} - 230.25\right) = 89.09$	10.91
11:45	173.8	9.6	124	125	124.50	73.00	129.75	355.7	1221.5	56.75	225.95	865.8	$\frac{1}{865.8}\left(\frac{173.8 \times 56.75}{9.6} - 225.95\right) = 92.55$	7.45
12:15	172.4	8.8	110	110	110.00	76.75	129.75	347.0	1218.9	53.00	217.25	871.9	$\frac{1}{871.9}\left(\frac{172.4 \times 53.00}{8.8} - 217.25\right) = 94.16$	5.84
12:45	171.1	9.9	124	124	124.00	73.00	133.00	355.4	1221.4	60.00	222.40	866.0	$\frac{1}{866.0}\left(\frac{171.1 \times 60.00}{9.9} - 222.40\right) = 94.06$	5.94
1:15	172.3	10.2	126	125.5	125.75	73.25	134.00	356.4	1221.7	60.75	222.40	865.3	$\frac{1}{865.3}\left(\frac{172.3 \times 60.75}{10.2} - 222.40\right) = 92.90$	7.10

TABLE NO. V.—*Continued.*

Time.	Weight of cold water corrected for heat capacity of barrel = W.	Weight of steam blown in = w.	Steam pressure at beginning.	Steam pressure at end.	Average steam pressure = P.	Initial temperature of water = t.	Final temperature of water = t_1.	Total heat in water at pressure $P = T$.	Total heat of steam = H.	$t_1 - t$	$T - t_1$	$H - T$	Percentage of dry steam $= \frac{1}{H-T}\left[\frac{W}{w}(t_1 - t) - (T - t_1)\right]$.	Percentage of moisture.
P.M. 1:45	182.4	11.0	125	125	125.00	73.00	135.00	356.0	1221.6	62.00	221.00	865.6	$\frac{1}{865.6}\left(\frac{182.4 \times 62.00}{11.0} - 221.00\right) = 93.24$	6.76
2:15	174.3	10.2	126	125	125.50	73.00	133.50	356.3	1221.6	60.50	222.90	865.3	$\frac{1}{865.3}\left(\frac{174.3 \times 60.50}{10.2} - 222.80\right) = 93.72$	6.28
2:45	182.3	10.7	127	127	127.00	72.75	132.50	357.1	1221.9	59.75	224.60	864.8	$\frac{1}{864.8}\left(\frac{182.3 \times 59.75}{10.7} - 224.60\right) = 91.74$	8.26
3:15	176.3	10.6	128	128	128.00	72.75	135.50	357.7	1222.0	62.75	222.20	864.3	$\frac{1}{864.3}\left(\frac{176.3 \times 62.75}{10.6} - 222.20\right) = 95.04$	4.96
3:45	182.5	10.6	128	127.5	127.75	73.10	133.75	357.6	1222.0	60.65	223.85	864.4	$\frac{1}{864.4}\left(\frac{182.5 \times 60.65}{10.6} - 223.85\right) = 94.91$	5.00
4:15	185.0	11.1	127	127.5	127.25	73.00	134.25	357.3	1221.9	61.25	223.05	864.6	$\frac{1}{864.6}\left(\frac{185.0 \times 61.25}{11.1} - 223.05\right) = 92.29$	7.71
4:45	179.1	10.6	126	127	126.50	72.60	133.50	356.8	1221.8	60.90	223.30	865.0	$\frac{1}{865.0}\left(\frac{179.1 \times 60.90}{10.6} - 223.30\right) = 93.15$	6.85
5:15	179.5	10.7	126	126	126.00	72.50	133.50	356.6	1221.7	61.00	223.10	865.1	$\frac{1}{865.1}\left(\frac{179.5 \times 61.00}{10.7} - 223.10\right) = 92.50$	7.50
5:45	182.9	11.0	127	126	126.50	72.60	133.80	356.8	1221.8	61.20	223.00	865.0	$\frac{1}{865.0}\left(\frac{182.9 \times 61.20}{11.0} - 223.00\right) = 91.86$	8.14
6:15	180.3	10.4	127	127	127.00	72.75	132.30	357.1	1221.9	59.55	224.80	864.8	$\frac{1}{864.8}\left(\frac{180.3 \times 59.55}{10.4} - 224.80\right) = 93.37$	6.62

<table>
<tr><td>6:45</td><td>184.1</td><td>10.9</td><td>128</td><td>127</td><td>127.50</td><td>72.50</td><td>133.00</td><td>357.4</td><td>1221.9</td><td>60.50</td><td>224.40</td><td>864.5</td><td>$\frac{1}{864.5}\left(\frac{184.1 \times 60.50}{10.9} - 224.40\right) = 92.25$</td><td>7.75</td></tr>
<tr><td>7:15</td><td>186.5</td><td>11.1</td><td>127</td><td>127</td><td>127.00</td><td>72.50</td><td>133.25</td><td>357.1</td><td>1221.9</td><td>60.75</td><td>223.85</td><td>864.8</td><td>$\frac{1}{864.8}\left(\frac{186.5 \times 60.75}{11.1} - 223.85\right) = 92.16$</td><td>7.84</td></tr>
<tr><td>Totals</td><td>3540.0</td><td>200.2</td><td></td><td></td><td>2503.75</td><td>1461.55</td><td>2620.10</td><td>7121.4</td><td>24431.6</td><td>1158.55</td><td>4501.80</td><td>17310.2</td><td>1852.13</td><td>147.86</td></tr>
<tr><td>Aver.</td><td>177.0</td><td>10.0</td><td></td><td></td><td>125.18</td><td>73.08</td><td>131.00</td><td>356.1</td><td>1221.6</td><td>57.92</td><td>225.10</td><td>865.5</td><td>92.61</td><td>7.39</td></tr>
</table>

TABLE NO. VI.

Record of Experiments with a Triple-expansion Engine at the Narragansett Station, Providence, R. I., July 13 and 14, 1890.

TABLE OF INDICATED HORSE-POWER.

No. of Indicator Card.	Hour.	Rev. Per Min.		Main Engine.												Condenser Engine.				Total.	
				14''x 48''. High Pressure Cylinder.				25''x 48''. Intermediate Cylinder.				33''x 48''. Low Pressure Cylinder.				12''x 16''.				I. H. P.	I. H. P.
		Main Eng.	Cond. Eng.	Back End. H. P. Constant =0.01866.		Crank End. H. P. Constant =0.01790.		Back End. H. P. Constant =0.05874.		Crank End. H. P. Constant =0.05799.		Back End. H. P. Constant =0.10367.		Crank End. H. P. Constant =0.10216.		Bottom. H. P. Constant =0.004569.		Top. H. P. Constant =0.004417.		Main Eng.	Cond. Eng.
				M. E. P.	I. H. P.	M. E. P.	I. H. P.	M. E. P.	I. H. P.	M. E. P.	I. H. P.	M. E. P.	I. H. P.	M. E. P.	I. H. P.	M. E. P.	I. H. P.	M. E. P.	I. H. P.		
	P. M.																				
1	9:50	99.33	62.0	45.84	84.9	48.71	86.6	11.86	69.2	12.37	71.2	9.936	102.3	9.873	100.2	27.93	7.9	33.18	9.0	514.4	16.9
2	10:10	99.00	61.5	47.43	87.6	48.90	86.6	11.36	66.1	12.37	71.0	9.824	100.8	9.873	99.8	28.39	7.9	32.10	8.7	511.9	16.6
3	10:30	99.10	61.0	46.73	86.4	49.98	88.6	11.88	69.2	12.54	72.0	9.956	102.2	9.873	99.9	29.66	8.2	30.63	8.2	518.3	16.4
4	10:50	"	61.0	43.60	80.6	45.92	81.4	11.67	67.8	12.79	73.5	9.916	101.8	10.000	101.2	29.14	8.1	33.40	9.0	506.3	17.1
5	11:10	"	61.0	43.34	80.2	44.39	78.7	11.95	69.5	12.51	71.9	9.844	101.1	9.958	100.8	28.73	8.0	30.18	8.1	502.2	16.1
6	11:30	"	61.0	46.41	85.8	49.60	87.9	12.02	69.9	13.10	75.2	10.027	102.9	10.054	101.8	29.49	8.2	31.90	8.6	523.5	16.8
7	11:50	"	61.5	45.20	83.6	47.12	83.6	11.67	67.8	12.84	73.8	9.956	102.2	9.873	99.9	31.45	8.8	32.56	8.8	510.9	17.6
	A. M.																				
8	12:10	"	61.5	44.11	81.6	45.85	81.3	11.41	66.4	12.58	72.3	9.895	101.6	10.022	101.4	28.39	7.9	30.64	8.3	504.6	16.2
9	12:30	99.00	61.0	43.34	80.1	44.65	79.1	12.37	71.9	13.10	75.2	10.292	105.6	10.512	106.3	31.53	8.5	33.79	9.1	518.2	17.6
10	12:50	"	61.0	45.71	84.4	46.62	82.6	11.95	69.5	12.44	71.4	9.976	102.3	9.894	100.0	29.02	8.1	32.72	8.8	510.2	16.9
11	1:10	99.14	61.5	45.58	84.3	46.93	83.3	11.74	68.3	12.75	73.2	9.874	101.4	9.980	101.0	29.43	8.2	30.47	8.2	511.5	16.4
12	1:30	"	61.5	44.11	81.6	46.62	82.7	11.95	69.6	12.89	74.0	9.976	102.5	10.213	103.4	30.65	8.6	30.35	8.2	513.8	16.8
13	1:50	"	61.0	47.31	87.5	47.69	84.6	11.76	68.5	12.49	71.8	9.956	102.3	10.032	101.6	29.26	8.1	32.50	8.7	516.3	16.8
14	2:10	"	61.0	44.49	82.3	44.33	78.6	11.67	67.9	12.58	72.3	9.976	102.5	10.118	102.5	28.39	7.9	32.72	8.8	506.1	16.7
15	2:30	"	61.0	45.01	83.2	47.76	84.7	12.00	69.8	12.54	72.0	9.915	101.9	10.075	102.0	29.43	8.2	32.72	8.8	513.6	17.0
16	2:50	"	61.0	45.20	83.6	44.78	79.4	11.74	68.4	12.54	72.0	9.907	102.7	10.161	102.9	28.73	8.0	31.20	8.4	509.0	16.4
17	3:10	"	61.5	47.50	87.9	48.90	86.7	12.21	71.1	12.96	74.4	9.907	102.7	10.192	103.2	30.18	8.5	31.82	8.6	526.0	17.1
18	3:30	"	61.0	44.75	82.8	45.28	80.3	12.07	70.2	12.77	73.4	10.027	103.0	10.139	102.7	26.71	7.4	33.40	9.0	512.4	16.4
19	3:50	"	61.5	47.82	88.5	48.84	86.6	12.16	70.7	13.12	75.4	10.089	103.7	10.352	104.8	29.08	8.1	31.14	8.4	529.7	16.5
20	4:10	"	61.0	43.79	81.0	46.11	81.8	12.04	70.1	12.65	72.7	10.242	105.2	10.458	105.9	29.84	8.3	32.61	8.8	516.7	17.1
21	4:30	"	61.5	43.92	81.3	47.12	83.6	11.97	69.7	12.61	72.4	10.089	103.7	10.394	105.2	27.98	7.8	31.03	8.4	515.9	16.2
22	4:50	"	61.5	47.82	88.5	47.38	84.0	12.35	71.9	12.49	71.8	10.180	104.6	10.352	104.8	26.94	7.5	28.66	7.7	525.6	15.2
23	5:10	"	61.5	47.63	88.1	49.15	87.2	12.02	70.0	12.70	73.0	9.976	102.5	10.257	103.8	28 74	8.0	30.02	8.1	524.6	16.1
24	5:30	"	61.0	44.37	82.1	45.47	80.7	11.93	69.4	12.68	72.8	10.048	103.2	10.257	103.8	27.12	7.5	29.45	7.9	512.0	15.4
25	5:50	"	61.5	47.05	87.1	48.08	85.3	12.09	70.4	12.82	73.6	9.976	102.5	10.192	103.2	26.71	7.5	29.90	8.1	522.1	15.6
26	6:10	"	61.5	45.96	85.0	46.55	82.6	12.30	71.6	13.10	75.3	10.027	103.0	10.234	103.6	23.01	6.4	31.42	8.5	521.1	14.9
27	6:30	"	61.5	47.31	87.5	49.85	88.4	12.28	71.5	13.00	74.7	9.915	101.9	10.161	102.9	27.06	7.6	29.68	8.0	526.9	15.6
28	6:50	"	61.5	47.05	87.0	47.37	84.0	11.97	69.7	12.58	72.3	9.804	100.7	9.979	101.0	23.42	6.5	30.75	8.3	514.7	14.8
29	7:10	"	61.5	47.88	88.6	48.20	86.7	12.23	71.2	12.54	72.0	9.804	100.7	10.075	102.0	26.83	7.5	29.06	7.9	521.2	15.4
30	7:30	"	61.0	45.14	83.5	45.09	80.0	11.93	69.4	12.44	71.5	9.936	102.1	10.096	102.2	27.75	7.7	29.73	8.0	508.7	15.7
Averages.		99.12	61.29	45.71	84.54	47.13	83.62	11.95	69.57	12.69	72.94	9.981	102.53	10.121	102.50	28.33	7.93	31.32	8.48	[illegible]	[illegible]

DISCUSSION.

Prof. R. H. Thurston.—This is, I believe, the best work yet eported from an engine in regular and ordinary service, and I hink "breaks the record." Such an extraordinary and excepional performance should, of course, be checked and corroboated by repetition; but I have no doubt that it may be accepted s practically correct, as here given. I understand that the ngine is guaranteed to come down to a still lower figure, and hen working at its intended pressure and load, its economy, heoretically, should be several per cent. better. It seems perectly possible, therefore, that the guarantee may be fulfilled. ince, as stated by the reporter of these figures, the steam was eriously wet, it is certain that the jackets must have had an mportant influence in securing this high efficiency of the mahine, notwithstanding the fact that the speed was 100 revoluions a minute. The fact illustrates once more, the now fre- uently and generally observed fact that, for any considerable xpansion, in seeking high economy, *efficient* jacketing must be ecured. That 14% of the steam applied should have transerred its heat to the engine through the jackets, simply onfirms this deduction. These results and their conseuences are the more satisfactory that they are obtained from n engine of but moderate size as compared with many in use, nd in the face of some decidedly disadvantageous circumtances. The heads, although not covered by a jacket in the rdinary sense of the word, are yet well jacketed, in effect, by he valve chests. I am especially interested in this arrangenent, as I endeavored to secure the adoption of exactly this lan many years ago, in engines on the designs of which I was hen working, and was unable to do so, although well convinced hat good jacketing and a minimum clearance, in engines of this lass, would permit the adoption of high ratios of expansion with resulting advantage in economy of steam, where, otherwise, such ratios would be impracticable. I am glad to see a youth's udgment thus confirmed. Here is another engine of which complete thermal analysis would be instructive.

Mr. Geo. H. Barrus.—In view of the work of the Duty Trial Committee, who recommend that the economy of a pumping engine be referred to heat units rather than to coal or water consumption, and to a practice which is becoming more general in working out the results of engine tests, I would suggest that results of this interesting trial would be more complete if there were added to the quantities given in the table, on page 662, the number of *heat-units* consumed by the engine per I. H. P. per hour, and the same per minute. I have worked out these quantities roughly, using, so far as I can determine from the records, the proper data, and I make the number of thermal units consumed per I. H. P. per hour 14,038, and the quantity consumed per I. H. P. per minute 234.

Prof. D. S. Jacobus.—In connection with this paper of Mr. Henthorn's, it may be interesting to know the results obtained in a test made on a triple-expansion engine by Prof. M. Schröter, of the Munich Polytechnic School, in 1889. The results obtained in those tests, which were made in the most careful way, fall directly in line with those given by Mr. Henthorn, and thus add another case which may be used to predict the probable gain due to the triple-expansion over the compound engine.

The engine was located at an iron works in Augsburg, and had the following dimensions: Diameter of cylinder, 11.1", 18.0" and 27.6"; stroke of all pistons 39.4". The boiler pressure was about 145 pounds per square inch. Total horse-power of engine about 200 revolutions per minute, 70. Five tests of from 5 to 6 hours' duration were made, which agreed very well with each other, and gave an average of 12.6 lbs. of steam per hour per horse-power, including all the steam condensed in the receivers and jackets. The cylinders and receivers were jacketed with steam at full boiler pressure. A separator was placed on the steam pipe near the engine, and the water drained from it was weighed and deducted from the water weighed to the boilers in order to obtain the water used by the engine.

Mr. Wm. Kent.—I see that the paper states that the presence of moisture in the steam most seriously affected the economic action of the steam with which it was mixed, and amounted to much more than there credited for water thus found. I think there is a chance for a difference of opinion on that point, and I would like to call the attention of the writer of that paper to it, and ask him to please state if he has proofs of the statement. I think possibly Prof. Jacobus can say something about the effect of water in the cylinder.

Prof. Jacobus.—Prof. Denton made special tests bearing on this point by forcing water into the steam-pipe leading to a small horizontal high-speed engine. In this case there was no appreciable difference in the rate of steam used per hour per horse-power. In other words, on deducting the water in the steam from the water used per hour, the remainder was the same as the amount of dry steam ordinarily used by the engine. The engine on which the experiments were made had a 7 × 14 inch cylinder. It is not proper to assume, with our present state of knowledge, that the results obtained on so small an engine will apply to Mr. Henthorn's case. A high ratio of expansion will also probably alter the action of the water. We hope to be able in a short time to make experiments on this point.

Mr. Henthorn.—In regard to the moisture found, it was considered by Mr. Leavitt and myself that it did have a serious effect on the economy of the engine—much more than the amount of credit which was given to the engine for the quantity of water found. It may be, as Mr. Kent states, open to various judgments, but we seriously considered the matter and concluded that it acted as a handicap to the engine's performance.

Mr. F'k Meriam Wheeler.—I desire to ask Mr. Henthorn one or two questions, but before doing so, I would like to make a few remarks about some of the details of this triple-expansion engine. For instance, the arrangement of the valves in the heads of the cylinders, thereby giving the minimum clearance; also, the high rotative speed, practically 100 revolutions a minute. The piston travel of 800 feet, of course, is not uncommon, but so high a number of revolutions for the Corliss type of valve gear is not often found. The question I wished to ask Mr. Henthorn was regarding the independent pumping engine which operates the air and circulating pumps. I notice that the horse-power of this engine was somewhat over 3% of the main engine,—in other words, over 16 H.P. Now I want to ask if any attempt was made to run this engine slowly and yet maintain satisfactory vacuum and temperatures of circulating and feed water? Three per cent. is rather a high figure for any independent engine of that kind. In the best modern naval practice, all auxiliary engines of the ship rarely exceed 2%, including the blowing engines, the steering-engines, and all steam pumps as well as the independent air and circulating pumps. Mr. Henthorn remarks in his paper that they expected to get an economy of 12.6 lbs. weight of steam per I.H.P., while the test showed approximately 13 lbs. It is plain to me that they would have exceeded the high result desired by not running the independent engine for the condenser as fast as they did. Being interested also in the matter of surface condensers, I note with curiosity the arrangement of placing the tubes vertically. Now this handicaps the condenser considerably. My own experience, and that of Mr. Clark in England, shows that tubes placed horizontally will do at least 15% more work. This is very apparent if you stop to consider a moment, for the steam after being condensed blows down the tubes and so covers them for nearly their entire length with a film of water. Consequently, their capacity for condensing is thus very much reduced. I thought perhaps this feature might have something to do with the temperatures of the circulating and feed water as shown, and which are somewhat unusual. For instance, the initial temperature of the circulating water was 72°, while the discharge was only 86°, thereby raising the temperature but 14°. In good marine practice, we always expect to get anywhere from 40° to 50° rise, which means, of course, very much less circulating water. A suitable by-pass valve can always be provided, thus enabling the correct regulation of the amount of circulating water, according to the season of the year, without regard to the speed of the air and circulating pumping-engine.

Mr. Henthorn.—In regard to the size of the condensing apparatus, which required, as the gentleman states, about 3% of the whole, I would state that the condensing apparatus as a whole was designed for an ultimate capacity of about 2,500 H.P., which included the engine which was tested and reported upon, and for the use of other engines which were to be put into the station as necessity required. We have recently added a second engine of the same type, of 1,200 H.P., and this also is connected to the same condensing apparatus which is in use for the 500 engine. So that the percentage of power developed by the engine of the condensing apparatus to the whole power would be reduced very materially when the station requires from 2,000 to 2,500 H.P.

Mr. Wheeler.—I might say that I have no doubt you could reduce that more than one-half. I have seen an independent engine carrying the air and circulating pumps produce a good result, and yet developing only a little less than 1% of the power of the main engine.

Mr. Henthorn.—Our conditions would bring it a little less than 1% if 2,000 H.P. had been developed in the station, and for which our condensing apparatus is ample.

CCCCXLI.*

A BLAST FURNACE BLOWING ENGINE.

BY FRED. W. GORDON, PHILADELPHIA, PA.
(Member of the Society.)

It is desired to present for comment and criticism a form of blast furnace blowing engine whose details are the result of an effort to design a standard form for such machinery.

The blowing cylinder is 84 inches in diameter, the steam-cylinder 42 inches in diameter, and the stroke 60 inches. Some years ago, blowing engines were ordinarily made with a stroke of 7 feet, and subsequently were made with a remarkably short stroke. It seems that the public demand for things brings about an averaging of mechanics' notions, and for good-sized engines, the favorite stroke is now 60 inches. The bed-plate is box form, 13 feet long, 6 feet 3 inches wide, and 2 feet 3 inches deep, the metal entirely surrounding the rectangular section, excepting the opening for the removal of the cores. The two cross girders, forming the ends of the bed-plate, and the side as far as the bosses for the steam-cylinder, are 14 inches wide. Besides the cross and box girders, there are two ribbed plate cross girders which stiffen the entire frame at a point where the steam-cylinder is bolted to the bed-plate. The bottom of the bed-plate is flat, to facilitate the bedding and to secure strength in the masonry. There are eight foundation bolts, each 2 inches in the body, with 2½ inch threads, passing through this bed-plate, and through the anchor plates which extend entirely across the foundation.

The bearing for the main shaft is made of cast iron, babbitted, planed in and scraped upon the bed-plate. The seat of this bearing is well up, leaving the box frame under each bearing to receive weight of the fly-wheels and the thrust of the steam when they both come together at the bottom centre, 24 inches deep and

* Presented at the Providence Meeting of the American Society of Mechanical Engineers (1891), and forming part of Volume XII. of the *Transactions*.

22 inches wide, having 3½ inches metal on top and 2 inches on each side and bottom.

There are two housings or frames. They are hollow box castings of rectangular section. The section of each of the four legs is 22½ x 11 inches at the bottom and 16 x 11 inches at the top. They are bolted directly on top of the bed-plate, and on top of them is bolted the lower head of the blowing cylinder, and they in turn bolted to the steam-cylinder near the top head, to give ample rigidity to the whole structure, while allowance is made for the expansion and contraction of the cylinder.

There are two fly-wheels, each 18 feet in diameter, weighing 15 tons. They are made in two sections, planed at the joint and on one side of the centre flange, after which they are carefully fitted and bolted with taper bolts to the crank-plate. The crank-plates are then forced upon the shaft and thoroughly keyed to it to remain there. The crank-pin holes are then bored out by using the main shaft to carry the bearings of the boring-bar, whereby accuracy in alignment and distance is secured. This is considered highly important in this type of engine, as in crossing the centres, the slightest variation in the positions of the crank-pins will strain the whole engine. Rocking cross-heads do not relieve these strains, though they mitigate them, while with true work and occasional trammelling of the connecting rods, they never exist.

The main shaft is of wrought iron 15 inches in diameter, with bearings 24 inches long. The shaft is turned parallel, except a very slight reduction where the crank centres are forced on.

The lower head of the steam-cylinder has four legs strongly ribbed to its whole surface. The bed-plate has four pads or bases corresponding to those, and the lower cylinder-head is placed upon these and accurately adjusted in position, and is bolted to the bed-plate by four 3-inch steel bolts, passing well into the bed-plate and keyed to it. Upon this lower head the cylinder is bolted in the usual way. The distance between the housings is just sufficient to admit the cylinder, and its width, added to the width of the two housings, is the width of the bed-plate. In other words, the housings are as close together as they can possibly be to permit the withdrawal of the steam-cylinder. The length of the cross-head and shaft is reduced to a minimum, and the crank-pins are brought as close as practicable to the fly-wheels. The crank-pins are steel, 7 inches in diameter, and being accurately bored, as above referred to, never heat with ordinary attention. (Figs. 215 and 216.)

The steam piston rod is secured to the steam piston by a nut on top of the piston head. This requires an enlargement of the rod at the lower end, the head to be passed over the rod and the nut following. The purpose of this is to get at the nut and screw it up properly, which is very difficult and is likely to be neglected when the nut is on the bottom of the piston, and should this large nut get loose on the bottom, it may quickly run off, while if on

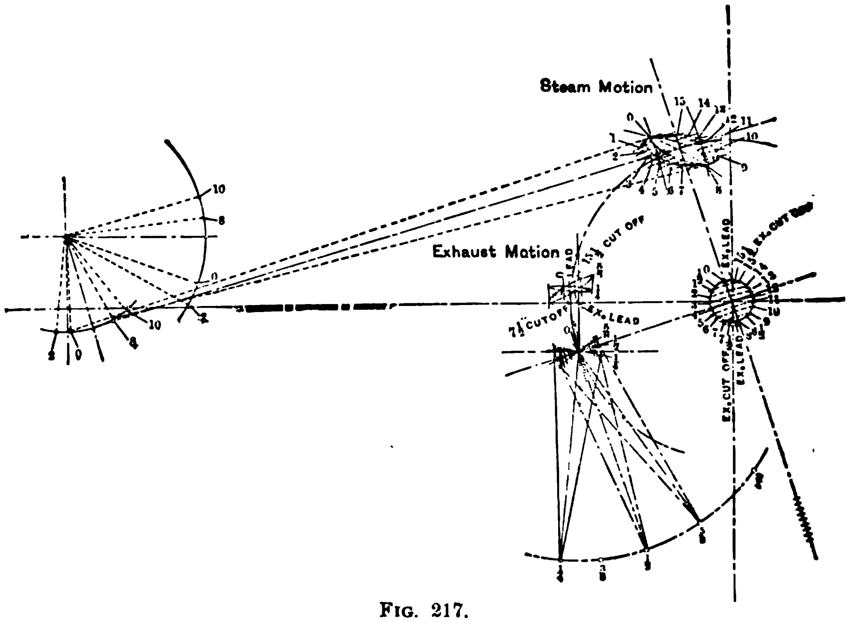


FIG. 217.

the top it will work off very slowly and pound before any damage is done. The plan is certainly somewhat more expensive, but the difference in cost is justifiable.

The steam and exhaust valves are the ordinary double balanced poppet valves. The steam-valves are made in one piece, the lower one being made small enough to drop through the seat of the upper one. They are 7½ and 7 inches in diameter area, 83 square inches or 6% of the area of the cylinder. This want of balance is to furnish a water relief through the steam-pipe, as in the lifting of an ordinary slide valve. The exhaust-valves have the lower valve the larger, which is introduced as a ring, the lower

as above referred to, never heat with ordinary attention. (Figs. 215 and 216.)

The steam piston rod is secured to the steam piston by a nut on top of the piston head. This requires an enlargement of the rod at the lower end, the head to be passed over the rod and the nut following. The purpose of this is to get at the nut and screw it up properly, which is very difficult and is likely to be neglected when the nut is on the bottom of the piston, and should this large nut get loose on the bottom, it may quickly run off, while if on

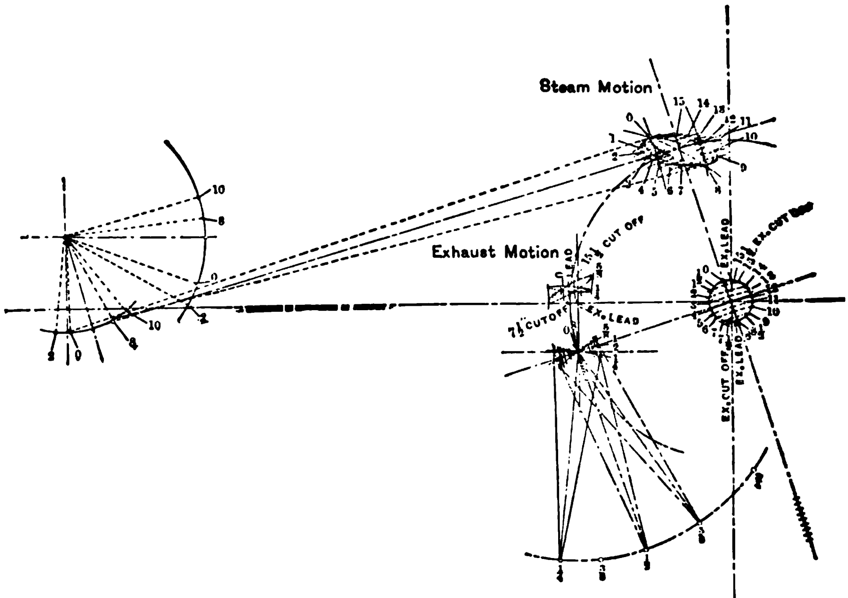


FIG. 217.

the top it will work off very slowly and pound before any damage is done. The plan is certainly somewhat more expensive, but the difference in cost is justifiable.

The steam and exhaust valves are the ordinary double balanced poppet valves. The steam-valves are made in one piece, the lower one being made small enough to drop through the seat of the upper one. They are 7½ and 7 inches in diameter area, 83 square inches or 6% of the area of the cylinder. This want of balance is to furnish a water relief through the steam-pipe, as in the lifting of an ordinary slide valve. The exhaust-valves have the lower valve the larger, which is introduced as a ring, the lower

Tool Works' manufacture. The cross-head being set to the machine, the three holes are bored without again moving it. We do not even depend on the accuracy of the machine in this case, as the cross-head is set accurately to the spindle of the machine. The shoulders for the piston rods are brought exactly to the centre of the cross-head.

The blast cylinder piston head is operated by two rods, that the strain upon a very large head may be better distributed than by the use of the central rod, and to effect better connections with the cross-head.

The connecting rods have solid ends, adjusted by keys, the brasses being very heavily backed with a wrought iron pad. Each slipper is 10 inches wide and 16 inches long, or 320 square inches,

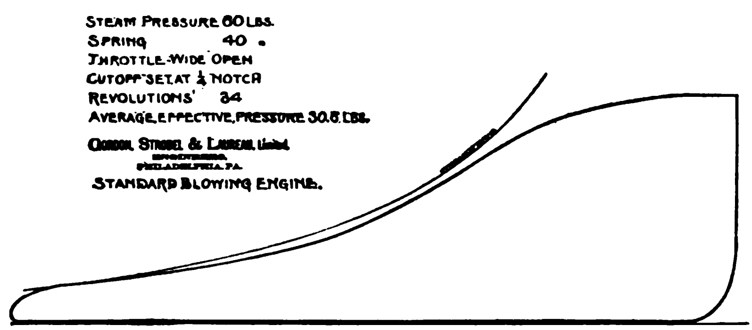


FIG. 218.

to receive the strain. The pads have no provision for setting out, as the wear is insignificant, and when it is necessary we propose to introduce a liner next the cross-head. This makes a more rigid adjustment than any screw arrangement, and has to be done so seldom that the trouble of doing it is not worth consideration.

The blowing piston head packing ring is brass, set out with springs with a long range of elasticity and light pressure. It is a single ring in segments accurately fitted together and turned true on top and bottom and on the outside. The ring wears well and keeps tight, and requires no attention until a new one is needed. The follower bolts are screwed into brass nuts in the web of the spider, as difficulty has been experienced with nuts coming off.

The valves of the blowing cylinder (Fig. 219) are so disposed that they all fall to their seats by gravity, requiring no springs, and are readily taken out and replaced in every instance without entering the cylinder, and in a moment's time. The construction is readily understood from the drawing. The outlet valves are of steel with leather on the back (not on the face). The face of the valve is ground in the lathe where it seats, the seat being turned true to receive it. The leather is used only to avoid the slamming noise

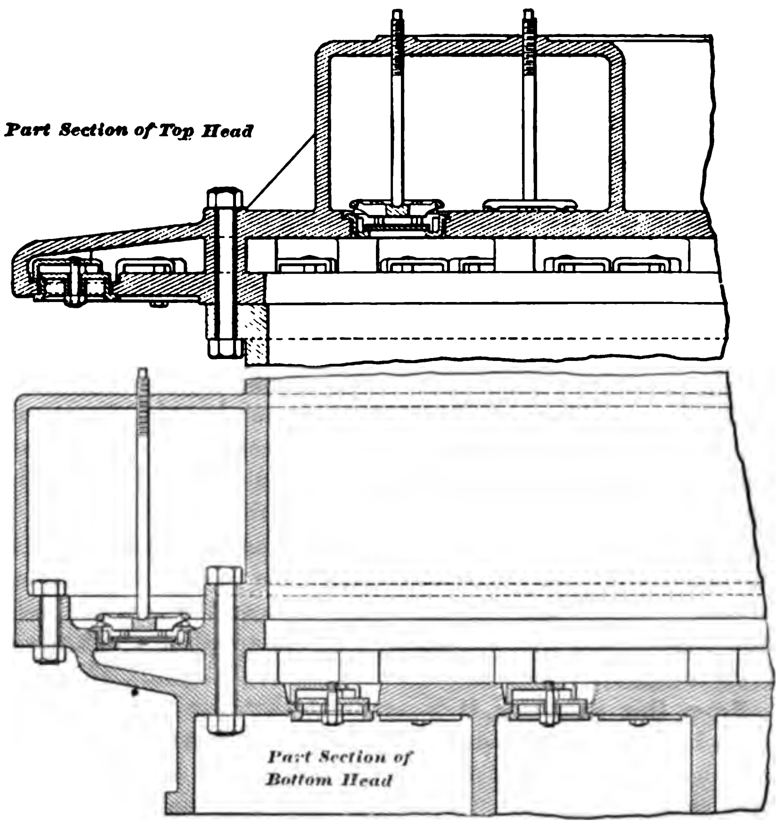


FIG. 219.

of the valve when opening. The valve gives a clear opening of 10.3 square inches, is $\frac{1}{16}$ inch thick, and weighs but $6\frac{1}{4}$ oz., which is .04 lbs. per square inch. The seat and guard are independent of the head, and together form the cage of the valve. Both are secured with a long bolt, adjusted from the outside, and are gotten at from the hand-hole openings, placed conveniently around the outlet chambers. The surface of the cylinder receiving the cages is faced as is the cage where it rests upon the cylinder. A cage can then be introduced and made secure at once, without any jointings of gum or other packing being used.

The inlet valve is of similar construction, as shown in the large drawing, but the valve is made of pure vulcanized rubber as good as the market affords; it is $\frac{3}{8}$ inch thick and $4\frac{1}{2}$ inches in diameter. The inlet valve never comes in contact with a heated atmosphere, and this gum valve (being absolutely tight) does not become heated, and is very durable.

The outlet valves are made of metal, as they are subjected under pressure to a high temperature which soon crisps leather and softens gum.

For quick motioned engines anything in the form of springs to close or ease its opening, or anything in the shape of hinges, where the wearing surfaces cannot be properly lubricated, is thought objectionable. The valve described avoids the use of hinges and wearing surfaces and springs. If the valves are heavy, or the use of springs is necessary to close them quickly, or if stiff hinges are to be overcome, there will exist a serious impediment to the flow of the air. The best which can be said of a valve is that it is only an interference to the direction of the flow, and this would be the case if it were as light as the air itself—lifting without resistance. To the inefficiency of the opening (caused by its form and the change of direction of the currents, effected by the valve), must be added the weight of the valve or its resistance to opening, to get the total difference between the pressure inside the cylinder, and the atmosphere for the speed at which the air is entering.

From this reasoning, it is apparent that the more freely the valve will open, the nearer it is to the requirements, provided it will close with sufficient promptness to prevent a loss by the return stroke. We have found that there is no liability to loss in this, as the lightest valve prepares to seat itself as the engine approaches the centre, and shuts instantly so far as the senses can appreciate it, the instant the centre is reached.

These valves are made as large as it is thought to be advisable for great durability, and are made sufficiently numerous to give the requisite effective opening with a lift of but $\frac{3}{16}$ of an inch.

From the indicator diagram of the blast cylinder (Fig. 220), it will be seen that the inlet valves show so little resistance to the in-going air, that the lines on the card are not distinguishably separated. This was taken with a 10 pound spring, the pressure by gauge being 4½ pounds and the speed 35 revolutions, the stroke of the engine being 4 feet. The outlet valves indicate a slight resistance, which cannot be measured, owing to the varying pressure in the outlet main.

Advocates of mechanically moved valves present three reasons for their use, all of which are theoretically very attractive. The first is the freedom from slamming and consequently less repairs; the second, efficiency of opening and consequently less waste of power; and the third, less dead space.

From what we have said, and by the proof of the indicator card, we need not touch upon the two first, as we think the repairs of these valves are as light as could possibly be expected in

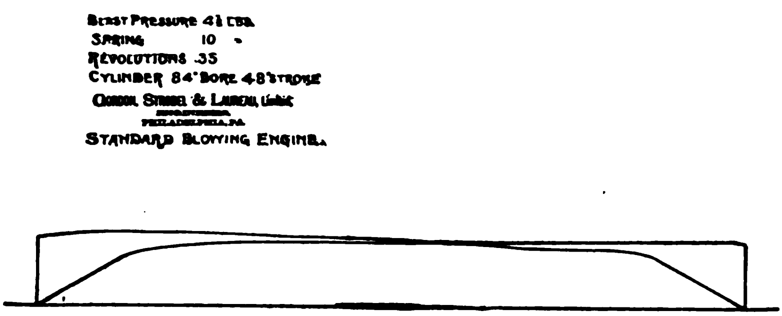


FIG. 220.

any case, and the efficiency of the opening is all that could be wished. The clearance will bear very favorable comparison with any class of construction, being but $2\frac{15}{100}$ per cent. including ⅜ inches clearance between the piston and the heads, which is rather more than is absolutely necessary; but, considering the character of the people into whose hands these engines sometimes go, we do not like to run them much closer. But leaving out this ⅜-inch clearance, the actual clearance due to the delivery or receiving chamber and valve cages is but 1½ per cent.

The engine is provided with a good substantial platform for ready access to every valve.

The total weight of this engine is 100 tons. It is designed to blow 15 pounds pressure per square inch at 40 revolutions per minute, delivering 15,000 cubic feet of blast per minute piston measurement.

44

DISCUSSION.

Mr. Wm. Kent.—I think it will be observed that this engine belongs to the same type which has been the favorite style of American blast furnace blowing engines for the last twenty years, and that there is no essential difference between this engine and the ordinary blowing engine, except in some details in the valve motion and in the proportioning of parts. The paper states that the details are the result of an effort to design a standard form for such machinery. The attempt, I am sorry to say, has not gone beyond the perfection of details of the common standard—which is a bad form indeed. The last paragraph says that the total weight of this form of engine is 100 tons. Now, I do not believe that in the whole range of steam machinery of any kind there is an engine which weighs 100 tons which develops as little power as the common blast furnace blowing engine. I am not criticising this particular engine, but the whole type of which this blowing engine is a sample. It is a bad type in that it uses enormous weight of engine for the horse-power to be accomplished. It uses a very large steam-cylinder. It runs a slow stroke. It has not an economical valve motion. It has the type of valve which was condemned by the late E. N. Dickerson, in his lecture before the Electric Club, as the very worst valve designed since the time of James Watt. This type of engine has been developed without any regard whatever to economy of fuel, and I do not think that is creditable to blast furnace engineering of the present day. A good blast furnace plant should have not only blast furnaces but a rolling mill alongside of it to utilize the surplus waste steam of the blast furnace. There is no excuse to-day for low economy of steam in blast furnace plants. During the last twenty years there have been marked improvements in air-valves. I suppose, that with Mr. Gordon's experience, these are very good air-valves, although I have not examined them. We are able now to design air-cylinders which can be run fast, and having reached this point it is now time to pay attention to the steam economy of the engine. I will have to quote here the last paragraph of Mr. Howe's paper, because, "though I have tried to treat the subject judicially, allowance should be made for bias on my part owing to direct pecuniary interest"—in a better engine. That is, I had the temerity some years ago to apply for a patent, and I got a patent on what I think is a better type of blowing engine. I used three steam and three air cylinders, with cranks at 120°. When I was over in England two years ago, some one told me that there was just such an engine built at Leeds by Sir James Kitson & Co., and I went there to see it, and they said to me : "We have not got quite so far as that yet ; we are only using two cylinders." Now, I hope that some American builder will go as far and will have the courage to build a three-cylinder blowing engine with cranks at 120° ; and, for fear I may be biassed financially, I will say that I will present to the first man that builds it a free license.

Mr. J. F. Holloway.—I am not now engaged in building blowing engines, and yet from the experience which I have had in that direction, I am not willing to go back on my fellow-members who are still building the American type of blowing engines. While at the Centennial, walking down the broad aisle of the machinery department one day, a gentleman caught me by the arm and said : "Now I have got you two together, and I want to introduce to the long-stroke blower of America the short-stroke blower," and as I turned around I met for the first time a gentleman whom we all highly honor, John Fritz, of Bethlehem. I said to my friend : "Possibly the right thing, then, is half way between these two extremes." Now, we all remember the early blowing engine of this country had a large cylinder and a long stroke. Indeed, if time would permit I might give you a history of some of those engines. Experience has, however, shown that the short-stroke blower has produced a revolution in iron-making in this country. It has produced a machine which in a small space has given much better results than it was possible to obtain by the early machine. There is one feature about a blowing engine which we should remember, and that is that the leakages of a blowing engine are imperceptible. In a slow-moving piston these leakages become a large part of the sum total of the capacity of the cylinder. The air is frittered out through the valves and joints, and in fact, like the wind in the Scripture, "it bloweth where it listeth; thou hearest the sound thereof, but canst not tell whence it cometh or whither it goeth." Now, the quick short-stroke blowing engine eliminates in part the question of time in which the air piston is getting from one end of the cylinder to the other. This to a large extent saves a wastage of the air in the cylinder. Now, I fancy that my friend Mr. Kent is not the first gentleman who has made the mistake of not understanding fully the business he is now criticising. (Laughter.) And I want to defend my brethren who are still in the business by saying that the pumping of air is a very peculiar business. I know of no other machine which is subject to just such trying conditions. You are called upon to pump a large volume of air under a very considerable pressure, and in order to do that you must have a pretty large steam-cylinder; and when you are at the beginning of your stroke, with your immense steam-power in your steam-cylinder, you have no resistance whatever on your air-piston; but when the steam-piston shall have travelled some portion of its stroke the resistance begins to accumulate, and when your steam has expanded itself down to its lowest pressure, it has then the most work to do. That is the reason why there is so much iron put into a blowing engine. The power has got to be transmitted, in the first place, to the fly-wheels. They then store up the power, for which there is no use at that time, and afterward give it out when the steam pressure is diminished and the air resistance is the greatest. The want of economy is a matter which gentlemen who have given much attention to high-speed engines, small clearances, and great expansion, no doubt deem very unfortunate, and, as compared with the engines with which we are familiar and of the class some of which we saw yesterday, it is a wasteful engine in the way of using steam; but it is a matter of necessity that steam should be used as it now is, and I do not know whether there is any patent which will come out hereafter which can make the possibility of cutting our steam short in a single-cylinder blowing engine when, as I say, there is no resistance against it at the beginning of its stroke, and when its greatest resistance is at its termination.

Mr. Daniel Ashworth.—This paper is specially interesting to me. Mr. Holloway has touched upon the vital points about which I intended to speak in regard to the peculiarities of a blowing engine. During the last two years I have been engaged in following this special subject. There is one question I would like to ask Mr. Gordon. In the paper he mentions the fact that the valve motion is such that it maintains a constant lead. In the Carnegie system, and in the blowing engines about Pittsburg, it is the general practice to dispense with the lead in this special line of engine, and the compression is kept down to a minimum, for the reason that we receive the compressive action and the resistance in the air-cylinder. The matter was brought forcibly to my mind during a test made in one of our large steel works, in which the diagram showed an inordinate amount of compression, and in my report I admonished them to have the valves so changed or adjusted, as quickly as possible, that this compression should be reduced to its minimum, to prevent a break-down, which I predicted would have disastrous results. The multitude of duties upon me, taking me from the city very often, was such that in the course of time it was lost sight of by myself, but I was called upon again by the same concern on other matters connected with blowing engines, and they informed me that in this interim of time they had received a very important lesson from the report that I had previously made to them; namely, that this identical engine had broken its cross-head and sheared off its crank-pin by reason of this evil. I am now in the midst of a series of tests in blowing engines, and I trust at some future time I may be able to present a paper before this Society covering this special field. In looking over this paper I recognize some points about this engine which had passed from my memory, but to-day in this field we are looking for more positive action in our air-cylinders. You give way to this enormous resistance, and the troubles which we have with valves adhering to their surfaces, or becoming detached.

Within a day or two before leaving my city, we started at the Edgar Thomson Steel Works a new type of blowing engine, built by the Southwark Company, at Philadelphia. It is of a type of positive action in the air-cylinder governed automatically by the pressure of the air in the receiver, and it promises to give entire satisfaction. I firmly believe that we are on the threshold of important steps in this matter, and I am glad that a paper of this kind has been presented; but I wish again to ask the amount of lead, and also compression, in Mr. Gordon's blowing engine practice.

Mr. Fred. M. Wheeler.—I was hoping that Mr. Kent would bring up some practical points in regard to this special service. I was also disappointed that Mr. Holloway should consider blowing engine matters so seriously, and that they required such remarkable handling. I think, on the contrary, that it is a very simple service, comparatively speaking, because the pressures to deal with are very low. In the compressing of air and gases, especially under higher pressures, there have been very great advances made in the past few years; for instance, in securing an earlier cut-off in the steam-cylinder and consequent better steam economy. In some air-compressing engines they secure this by arranging the steam and air cylinders at different angles. In the DelaVergne machine, the engine cylinder is placed at right angles to the air-cylinder, so that when its piston is giving out its greatest power it is then meeting with the greatest load on the air-piston. There is no reason why another and entirely different mechanical arrangement than that shown should not be adopted for blowing engines if we are to gain in economy of steam. I also do not understand why the writer of this paper did not consider the matter of minimum clearances in air-cylinders, because that is quite an item. I also notice that this engine made but 35 revolutions a minute, which is only about 350 feet of piston travel. I have seen blowing engines (with the design of which I had something to do) making as high as 200 revolutions per minute; while at 100 revolutions a minute they worked with greatest ease and entire satisfaction; I therefore do not see why an engine of the calibre to which this paper refers cannot be run at 100 revolutions.

Mr. Holloway.—Speaking with regard to the very high speed of blowing engines, as recommended by the last speaker, I want to say that there was at the Centennial a blowing engine whose highest recommendation was that it ran at an enormous speed, which was true; but it didn't blow, as the air could not get through its valves in time to fill the air-cylinder.

Mr. Henry M. Howe.—I would like to ask those who are present who are steam engineers whether the fact brought out by Mr. Holloway, that the air offers no resistance at the beginning of the stroke, is not the strongest possible argument for using a compound engine rather than a simple engine?

Mr. Holloway.—Yes, sir.

Mr. Howe.—I believe compound engines are not used very much for blowing engines in this country, but I know that they are used in England with very great success.

Mr. James McBride.—If I understand this matter, the steam for those engines is generated from waste gases. This being so, the attempt to make the engines highly economical in the use of steam reminds me of a friend of mine who had a large planing mill, using the shavings for fuel. He was induced to put in a very economical boiler and engine, and has since been employing a horse and wagon, and two men at a dollar and a half each, to cart away his shavings to the meadows and burn them.

Mr. John F. Wilcox.—In regard to what has been said by Mr. Kent, that if anybody would put in a triple cylinder compound blowing engine he would make them a present of a license under his patent, I would say we now have in process of erection such an engine for the use of the West Superior Iron and Steel Co., designed by our fellow-member, Mr. W. F. Mattes, general manager of that company, and I accept Mr. Kent's offer. I am glad he made it before the meeting, because I have ample witnesses to the fact of offer, and that I now take it up.

*Mr. F. W. Gordon.**—I will say to Mr. Kent that we have no desire to make any essential changes for the sake of change alone. Where a design of engine has stood the test of severe use, we will not depart from it without very thorough reasons. I believe this American type of blast furnace blowing engine to be the best which has ever been made for the purpose, and any departure from it would be a mistake. Many endeavors have been made by eminent engineers radically to change this blowing engine, but these very makers have returned to this general style of engines. The public use of a thing for a series of years, to the exclusion of all other forms, is certainly a very high commendation; not that it proves that it is theoretically correct, but that it is commercially correct; and this reasoning may be applied to the form of steam and exhaust valves which is here employed, and may be a direct contradiction to the views of Mr. E. N. Dickerson, for had these valves been the worst design since the time of Watt, the public would have discarded them long ago, or else we must consider the public an egregious set of fools. For my part I have great faith in the popular decision in such matters. My own experience with poppet valves has been exceptionably favorable. I designed and constructed the steam engines for the steamer *Natchez* in 1869, wherein were placed double poppet valves exactly the same as shown in this drawing. After ten years' use, working a steam pressure of 140 lbs. per square inch, these engines were taken out and placed upon a new steamboat, the valves never having been ground in during the entire time, the best proof of their excellence. As far as my experience goes, a double poppet valve, with a 60°

*Author's Closure.

angle and seating perpendicularly, will keep interminably tight. A flatter seat than this has a tendency to wear from side to side, and will not centre itself with sufficient certainty to maintain a tight joint.

The remarks of Mr. Holloway go far to answer Mr. Kent's objection to the great weight of the engine. Lighter forms of engines are condemned by the users, for they fully well know that they are apt to give way. Note in Mr. Ashworth's remarks the instance where the engine gave way, shearing the crank-pins and breaking the cross-head. I do not know the design of this engine, but assume that the Edgar Thomson Steel Works would buy eminently suitable and especially massive machinery. That by the derangement of the lead of the valves such enormous strains might be brought upon the engine, is a decided proof of the necessity of excessive weight in this class of machinery. As Mr. Holloway justly remarks, it is subject to much more severe strains than ordinary motive-power engines.

Mr. Kent refers to the speed of the engine, which is 40 revolutions per minute, or 400 feet of piston speed. It may surprise him to know that this is an excessively high speed for a blast furnace blowing engine, as perhaps there is not one in a hundred which is doing this work.

Mr. Wheeler states that he did not see why an engine of the calibre to which this paper refers cannot be run at 100 revolutions. I would say that there is nothing in the world to prevent the engine from running at 100 revolutions per minute, except that the air valves are not sufficiently numerous. If it was designed for that speed, the steam and air valves would have to be proportionately increased, but it is not thought desirable for the small amount of steam and the slight saving in the total cost of the blast furnace to run the engine at such a speed, and depend upon that engine for making pig iron for 365 days and nights in the year. The blast furnace blowing engine is not a transatlantic steam-engine which runs a week and has a week for repairs. Neither have you the Saturday afternoons and Sundays of the factory to look into the details and keep the machinery in perfect order. The engine must go every day and every night, Sundays included. Sometimes it does its hardest work when its stoppage would be death to the furnace while out of repair.

It no doubt appears to all steam engineers, who have not had experience with blast furnaces as well as blast furnace blowing engines, that it is a simple proposition to make blowing engines for blast furnaces which would be economical in their steam consumption and approach modern steam practice in every respect. Our firm construct engines with Corliss steam valves, the speed being controlled by the governor regulating the cut-off. This would be a little more economical than this engine, but very little more so, as it would be necessary to wire-draw the steam on a great many occasions, or else to regulate the whole steam pressure of the plant. Ordinarily a blast pressure of 6 or 7 lbs. is sufficient. Occasionally 15 lbs. are a necessity. To meet these occasions where the cost of the engine would be lost in a very short time if the furnace would chill, the proportion of the steam-cylinder has to be abnormally large in proportion to the blast-cylinder, and on these very occasions the fuel, which is the waste gases from the furnace, is likely to entirely fail us, and steam at that time has to be raised with prodigious effort by firing under difficulties. It is then that the engine should be the most economical in its steam consumption, as at other times a greater power would be obtained from the gases than is required for the work.

If the steam is permitted to follow one-quarter of the stroke of this engine, by adding the clearance, the ratio of expansion will be three and sixty-five one-hundredths, and with steam at 75 lbs. above the atmosphere at the point of cut-off, we will have a mean effective pressure of about 40 lbs. per square inch, which, less friction, would be equal to 10 lbs. mean effective force upon the piston of the blowing cylinder. A higher pressure than this is frequently required, and then either a higher steam pressure or a longer follow must be substituted. Blast furnace proprietors are averse to very high steam pressures, and it is very seldom that 100 lbs. is used. Eighty is considered high, and 60 is a very common pressure to work at. It will be seen, then, that instead of the steam-cylinder being too large for the air-cylinder, it is too small to get the best economy by cutting off the steam when the most difficult emergencies require the best economy.

Mr. Wheeler seems to consider that the blowing of air is a very simple service. In this I would agree with him, if the service was constant, but to build an engine which will blow from 5 to 15 lbs. pressure per square inch, and do it efficiently, is perhaps as difficult a problem as was ever presented to engineers. I do not think the question of compounding engines comes within the scope of this paper. This is a single-cylinder high-pressure engine, and will ordinarily give the best economy with about one-quarter follow under its ordinarily heaviest duty, and by the medium of the variable cut-off gear can be arranged to give the best efficiency practicable under the range of duty required. Compound engines may sooner or later be introduced into blast furnace practice. At present they are not in favor. So far as my experience goes with blast furnaces, I would be adverse to anything which would complicate in the least the blast furnace blowing engine. The economy of fuel within the furnace, the absolute command of the furnace, the reliance to be placed upon the engine, made simple and massive, will redound to the profit of the iron-master much more than the saving of the steam by the most efficient engine.

Answering Mr. Ashworth's question: The valves of this engine are set without lead—that is, at the dead-centre; the steam-valves are just about to leave their seat, and as the gear gives a very rapid movement to the valves, they soon have a free opening for the admission of the steam, giving practically boiler pressure at the commencement of the stroke. The exhaust-valves are set to close when the piston is within two inches of the end of the stroke, causing very little compression indeed, and thus meeting the very objections which he found to the engine he describes. It is ordinarily true that the blowing engine has sufficient compression in its blowing cylinder to do away with all compression in the steam-cylinder. The American type of blowing engine is more suitable to take advantage of this than any coupled engine with beams or cranks transmitting the energy through joints. The compression of the blowing cylinder is here transmitted directly to the whole mechanism by the piston rods, without joints, except those of the connecting rods.

What I referred to in the paper was, that the valve gear gave an *invariable* lead and exhaust, and this is more important in the case of a blowing engine, where lead is so objectionable, than where it may be used to considerable extent without injury to the machine. When this engine is placed upon the dead-centre, the whole range of expansion may be passed through by the movement of the lever without the slightest movement of the steam-valves being observed.

I have referred to the positive-acting valves that Mr. Ashworth describes, and gave as my reason for their use that heretofore air-valves of blowing engines working automatically have been made excessively heavy, even with guide stems top and bottom, or a large flat leather valve with leather hinge, and that had a movement in opening of 3 or 4 inches sometimes. The difficulties with these types of construction have led to a desire for positive-moving valves with their necessary complicated mechanism and sliding surfaces.

The valves we employ are automatic, very light, and lift a moderate distance. The action is so quick that no slip of the air or loss is to be seen in the diagrams or by observation of the engine itself. We have operated these engines as high as 85 revolutions per minute, and found no objection in the operation of these valves; and, were they doubled in number, we would not hesitate to guarantee these engines to do efficient and satisfactory work at 100 revolutions per minute. But, as said before, we would not advise the iron-master to rely upon one of these engines to do the work of which this engine is capable at 100 revolutions per minute.

CCCCXLII.

A BELT DYNAMOMETER.

BY S. P. WATT, CINCINNATI, OHIO.
(Junior Member of the Society.)

THE dynamometer here presented (Fig. 221) consists of a set of pulleys mounted on a suitable frame and disposed as follows:

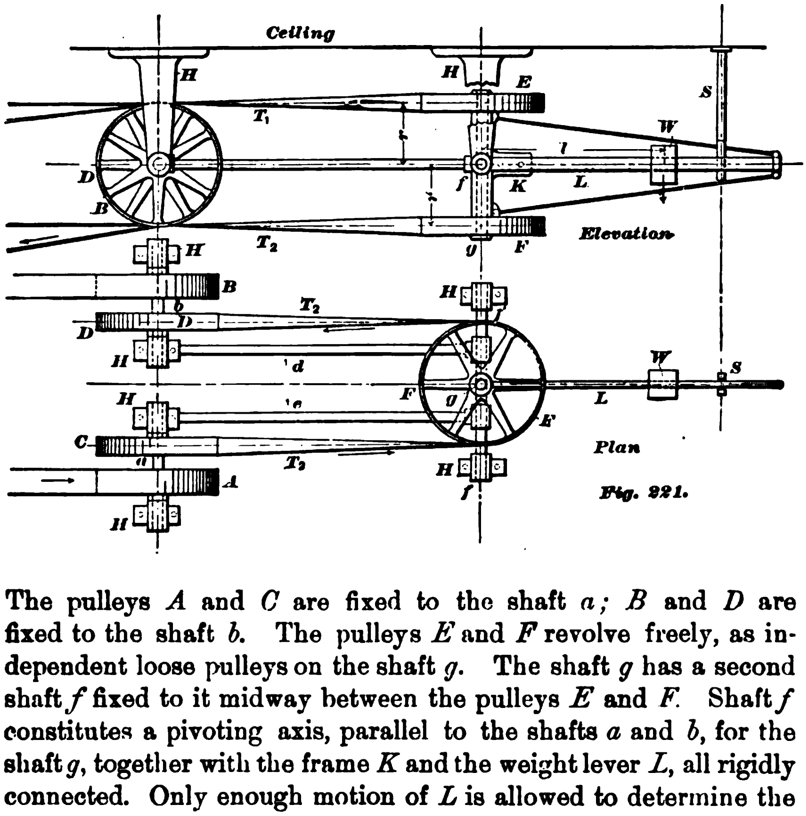


Fig. 221.

The pulleys A and C are fixed to the shaft a; B and D are fixed to the shaft b. The pulleys E and F revolve freely, as independent loose pulleys on the shaft g. The shaft g has a second shaft f fixed to it midway between the pulleys E and F. Shaft f constitutes a pivoting axis, parallel to the shafts a and b, for the shaft g, together with the frame K and the weight lever L, all rigidly connected. Only enough motion of L is allowed to determine the direction of action. Instead of the weight on the lever L, the

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. of the *Transactions.*

end of the lever could be connected to a small platform scale and its tendency to rotate weighed. It might be useful to make the frame between the pulleys adjustable at *d* and *e* in order to vary the tension of the dynamometer belt, or better still it might be so constructed that the absolute tension could be noted at any time whether working or at rest. The working of the apparatus is as follows: a driving belt from the source of power is put on the pulley *A*. The machine to be driven is belted from the pulley *B*. The dynamometer belt passes from the lower side of the pulley *C* to the pulley *F*, around *F* to *D*, around *D* to *E*, around *E* back to *C*. It will be seen that *C* is a driving pulley and *D* a driven pulley. When the system is at rest the four strands of the dynamometer belt have the same tension. If now *C* revolve and drive *D*, the tension T_1 of the belt from *C* around the loose pulley *E* to *D*, will correspond to the tension of the taut side in a simple system of two pulleys, and the tension T_2 of the belt from the lower side of *D* around *F* back to the lower side of *C* will correspond to the slack side.

The difference of tension is the driving pressure *P*, and taking what actually occurs we have

$$\frac{2T_1-2T_2}{2} = T_1 - T_2 = P.$$

Now *P* in pounds multiplied by the speed of the belt is foot pounds developed or consumed, ignoring friction. Let *r* be the radius of the position of pulleys, *E* and *F* from the pivot *f*. Let *l* be the distance of the weight *W* from *f*, to balance the tendency of the frame K to rotate about *f* when working, then

$$Wl = r\,(2T_1 - 2T_2) \text{ or } \frac{Wl}{2r} = T_1 - T_2 = P.$$

It is evident that should a Prony brake be put in place of the pulley *B* the power developed by the motor to *A* could be determined. If a machine be driven from the pulley *B*, the power consumed could also be noted in the speed of belt and the position of the weight from the same formula. In the use of different belts as dynamometer belts, the relative efficiency of such belts can readily be determined by the use of the brake attachment. It will also be seen that but one side of the belt comes in contact with the pulleys.

CCCCXLIII.*

ON THE EXACT SUBDIVISION OF AN INDEX-WHEEL INTO ANY REQUIRED NUMBER OF EQUAL PARTS.

BY W. A. ROGERS, WATERVILLE, ME.
(Member of the Society.)

THE writer had occasion, a few months ago, to subdivide into one thousand equal parts the index-wheel of the screw-cutting engine (Fig. 227) of the paper on a perfect screw-cutting machine. The diameter of the wheel is four feet.

At first, an attempt was made to produce an exact reproduction of a graduated disk having one thousand subdivisions, purchased of a maker of considerable reputation; but an investigation of a provisional transfer indicated the existence of very large errors of a systematic character. While the relative errors of adjacent subdivisions were very small, the cumulative errors were very large. At one point in the revolution, the lines were even thrown out of the field of the microscope. This method of transfer was therefore abandoned.

The method which will now be described was found to be easy of application and entirely adequate. First, a narrow strip of firm paper was wrapped around the disk, and its length was made equal to the circumference. By three foldings it was then divided into four equal parts. Each of the four parts was then further subdivided into twenty-five equal parts by the aid of an auxiliary paper scale.

The paper band was then fastened firmly to the periphery of the wheel, and the divisions were transferred to the disk by means of a small try-square and a sharp cutting tool.

It was found by examination that while the errors of the individual graduations were in some cases quite large, the cumulative errors did not much exceed the accidental errors of subdivision.

The next step consisted in the investigation of the errors of

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. of the *Transactions*.

the one hundred subdivisions thus produced. Two microscopes, fitted with filar micrometers, were set at a distance apart approximately equal to one-tenth part of a revolution. The fixed line of the micrometer of the lower microscope was set in succession upon each subdivision into ten equal parts, and the reading of the micrometer of the upper microscope was taken for coincidence with the lines under this microscope. The microscopes were then set at a distance apart equal to one-hundreth of a revolution, and similar measures were made for each group of ten subdivisions. The method of reducing the observations will appear from the following example. The first column contains the readings of the upper micrometer, after coincidence had been made with the tenth line preceding under the lower microscope. The second column contains the corrections required to reduce the readings for each line to the mean of the ten readings. These relative corrections are referred to

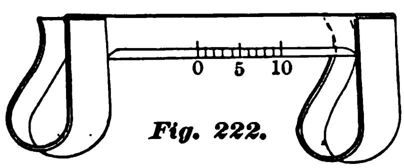


Fig. 222.

the initial line of the series by successive additions, as shown in the third column.

The summed series is indicated by Σ, the usual sign for summation. For example, the space between lines 0 and line 60 is 48.1 divisions of the micrometer, greater than $\frac{6}{10}$ of the entire circumference.

In the present case the value of one division of the micrometer was found to be eight-millionths of an inch, but the reduction to absolute units of length is not required.

The second table contains a similar investigation of the corrections for each line of the groups 0..10, 10..20, 20..30, etc.

In order to economize in space, only the data of the first four groups are given.

The results given in Tables I. and II. are combined in Table III. The first column contains the corrections for each tenth of a revolution, distributed proportionately over each hundredth part of a revolution. This process is not strictly correct. In order to reduce the magnitude of the error thus introduced, it will generally be found necessary to make two and sometimes three repetitions of the investigation with new sets of graduations obtained by applying the corrections already obtained to the micrometer reading of each of the lines of the first set of graduations. It is my custom to trace the graduations on opposite edges of the periphery, so that when a second set is made the first may be polished out.

If the accidental errors of the provisional graduations are large, this method of reduction will introduce rather large systematic errors, which will nearly or quite disappear in the second approximation. In the present case three approximations reduced the errors of the subdivision to a minimum value of about 0.0003 inch. It may be added that the time required for each series of measures was about two hours.

The subdivision of each of the 100 subdivisions into 10 equal parts was accomplished by means of an arrangement shown in Fig. 222.

A strip of band saw steel was prepared with a bevelled edge, upon which were traced two lines whose distance apart was by successive trials made exactly equal to each of the subdivisions ruled upon the index-wheel. This space was then subdivided into 10 exactly equal parts, thus obviating the necessity of applying any corrections.

The process of ruling the lines consisted in bringing each of the 100 lines into coincidence with the micrometer of the fixed microscope, and each line of the auxiliary scale in coincidence with the micrometer of a second microscope, the scale being moved forward into coincidence with a new subdivision as fast as each one-hundreth division was ruled. The time required to make the entire subdivision into 1,000 equal parts was three hours and forty minutes.

TABLE I.

Lines.	Microm. reading.	Corr.	Σ.
1	15.4 div.	+ 16.2	+16.2
2	10.4	+ 21.3	+37.5
3	45.1	− 13.5	+24.0
4	12.0	+ 19.7	+43.7
5	59.0	− 27.4	+16.3
6	96.0	− 64.4	−48.1
7	12.0	+ 19.7	−28.4
8	0.4	+ 31.2	+ 2.8
9	0.1	+ 31.61	−34.4
10	66.0	− 34.3	0.0
	31.64		

TABLE II.

Group 0 to 20. Microm. reading.	Corr.	Σ.	Group 10 to 20. Microm. reading.	Corr.	Σ.	Group 20 to 30. Microm. reading.	Corr.	Σ.	Group 40 to 50. Microm. reading.	Corr.	Σ.
10.6 div.	+ 5.9	+ 5.9	10.0 div.	−0.8	− 0.8	6.2 div.	− 3.6	− 3.6	7.6 div.	− 1.0	− 1.0
20.1	− 3.6	+ 2.3	18.0	+7.2	+ 6.4	8.0	− 1.8	− 5.4	8.0	− 0.6	− 1.6
38.0	−21.5	−19.2	14.2	+3.4	+ 9.8	0.0	− 9.8	−15.2	1.0	− 7.6	− 9.2
4.2	+13.3	− 6.9	17.3	+6.5	+16.3	20.1	+10.3	− 4.9	12.4	+ 3.8	− 5.4
8.1	+ 8.4	+ 1.5	2.4	−8.4	+ 7.9	12.0	+ 2.2	− 2.7	17.1	+ 8.5	+ 3.1
25.1	− 8.6	− 7.1	16.3	+6.0	+13.9	14.0	+ 4.2	+ 2.5	19.0	+10.4	+13.5
0.4	+16.1	+ 9.0	9.2	−1.6	+12.3	17.0	+ 7.2	+ 8.7	1.0	− 7.6	+ 5.9
40.1	−23.6	−14.6	4.8	−6.0	+ 6.3	2.2	− 7.6	+ 1.1	3.8	− 4.8	+ 1.1
10.2	+ 6.3	− 8.3	6.7	−4.1	+ 2.2	10.4	+ 0.6	+ 1.7	5.8	− 2.8	− 1.7
8.3	+ 8.2	0.0	8.6	−2.2	+ 0.0	8.1	− 1.7	+ 0.0	10.3	+ 1.7	0.0
16.51			10.8			9.8			8.6		

TABLE III.

Lines.	I.	II.	Sum.	Lines.	I.	II.	Sum.	Lines.	I.	II.	Sum.	Lines.	I.	II.	Sum.
0	0.0	0.0	0.0 div.	10	+[16.2]	0.0	+[16.2]	20	+[37.5]	+ 0.0	+[37.5]	30	+[24.0]	0.0	+[24.0]
1	+ 1.6	+ 5.9	+ 7.5	11	+18.3	− 0.8	+17.5	21	+36.2	− 3.6	+32.6	31	+26.0	− 1.0	+25.0
2	+ 3.2	+ 2.3	+ 5.5	12	+20.5	+ 6.4	+26.9	22	+34.0	− 5.4	+28.6	32	+28.0	− 1.6	+26.4
3	+ 4.8	−19.2	+14.0	13	+22.5	+ 9.8	+32.3	23	+33.5	−15.2	+18.3	33	+29.9	− 9.2	+20.7
4	+ 6.4	− 6.9	− 0.5	14	+24.8	+16.3	+41.1	24	+32.1	− 4.9	+27.2	34	+31.9	− 0.4	+31.5
5	+ 8.1	+ 1.5	+ 9.6	15	+26.6	+ 7.9	+34.5	25	+30.8	− 2.7	+28.1	35	+33.9	+ 3.1	+37.0
6	+ 9.7	− 7.1	+ 2.6	16	+28.8	+13.9	+42.7	26	+29.4	+ 2.5	+31.9	36	+35.8	+13.5	+49.3
7	+11.3	+ 9.0	+20.3	17	+31.0	+12.3	+43.3	27	+28.0	+ 8.7	+36.7	37	+37.8	+ 5.9	+43.7
8	+12.9	−14.6	− 1.7	18	+33.1	+ 6.3	+39.4	28	+26.7	+ 1.1	+27.8	38	+39.8	+ 1.1	+40.9
9	+14.5	− 8.3	+ 6.2	19	+35.3	+ 2.2	+37.5	29	+28.4	+ 1.7	+30.1	39	+41.7	− 1.7	+40.0
10	+[16.2]	0.0	+16.2	20	+[37.5]	0.0	+[37.5]	30	+[24.0]	+ 0.0	+[24.0]	40	+[43.7]	0.0	+[43.7]

CCCCXLIV.*

A NEW BELT TESTING MACHINE.

BY GEO. I. ALDEN, WORCESTER, MASS.
(Member of the Society.)

A MACHINE for testing belts should meet the following requirements: It should give accurately the sum of the belt tensions; the power transmitted, from which the difference of tensions can be readily found; the number of revolutions of the driving pulley; the number of revolutions of the driven pulley and the number of runs of the belt during the trial, and also during any interval of the trial.

It should be capable of adjustment to a constant load and to any load within the capacity of the belt it is proposed to test; to a change of load without changing the sum of the tensions; to any constant load with the sum of the tensions varied. It should give the arc wrapped by the belt on the driving and driven pulley and should record automatically the revolutions of the driving and driven pulleys and the runs of the belt.

The object of this paper is to describe a belt testing machine, designed for the Engineering Laboratory of the Worcester Polytechnic Institute, and built at the Washburn shops. Much of the design is the work of Mr. Wm. W. Bird, formerly an instructor in the Institute.

The peculiar features of the machine are:

First. The attachment to the driven pulley of an automatic absorption brake, made on the principle of the dynamometer, described on page 958, Vol. XI., of the *Transactions* of the Society.

Second. The method of supporting the driven pulley, by which the weight of the pulley is taken by the tensions of the belt.

Third. The recording apparatus.

The construction of the absorption brake and its attachment to the driven pulley is shown in Figs. 223 and 224. Fig. 223 is

---

* Presented at the Providence meeting of the American Society of Mechanical Engineers (1891), and forming part of Volume XII. of the *Transactions*.

a part sectional plan and Fig. 224 an elevation. The pulley *A* turns freely on the shaft *B*. The hub of this pulley is turned down at *a*, and a cast iron piece *C* is bored out, fitted upon the hub, and bolted to the web of the pulley, as shown at *c c*. This cast iron piece has its face *F* finished smooth. Another cast iron piece *D* carries a copper disk *d*, which makes water-tight joints with the piece *D* by means of driven wrought-iron rings, *R R*. Between the copper disk *d* and the cast iron piece *D* is a space *S S*, to which city water is admitted through a pipe not shown in the drawing.

The forked arms *E E* are bolted to the piece *D* and carry projections *e e*, from which, by means of knife edges and rods, are suspended the pans *F F*, which receive the weights used to balance

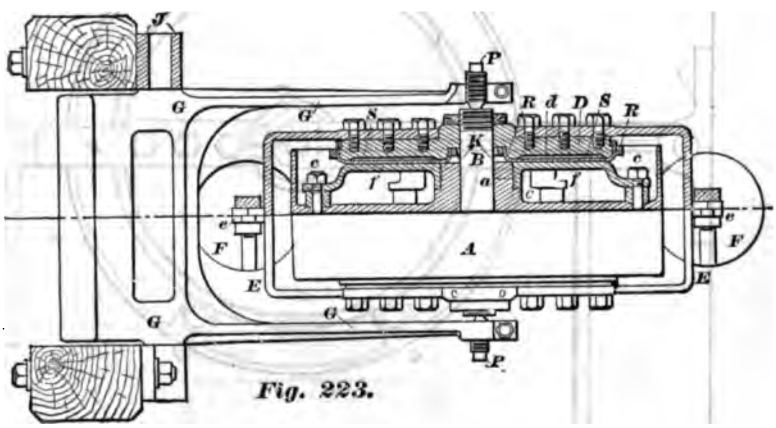


Fig. 223.

the load, this load being the friction between the copper disks *d* and the piece *C*. The pressure which causes this friction is produced and regulated by the admission of city water to the space *S*. The rubbing surfaces *f* are lubricated by the admission of oil to the cavity *K*, the oil being carried slowly over the surfaces from the centre to the circumference by centrifugal force, making the lubrication perfect. The heat generated is carried off by allowing the water, which presses the copper disk against the surface of the revolving piece *C*, to circulate freely to the brake.

The pulley and its connected parts are supported by the belt, and in order to keep the axle of the pulley horizontal, a balancing frame *G* is provided. This frame is hung in adjustable bearings *J*, and its weight is so distributed that its centre of gravity is on the axis of the bearings. Pins *p*, with conical ends, engage with the ends of the shaft on which the pulley revolves, in the manner shown in Fig. 223, so that while the balancing frame does not affect the weight which comes on the belt, it prevents oscillation of the pulley axle, and at the same time allows the pulley to adjust itself to the stretch of the belt.

Fig. 225 is an elevation of the machine in which M is the driving and B the driven pulley. The speed of the driver may be varied in any usual way. S is a shaft which receives a slow positive rotary motion from the shaft of the driver by means of a worm T and a train of gearing.

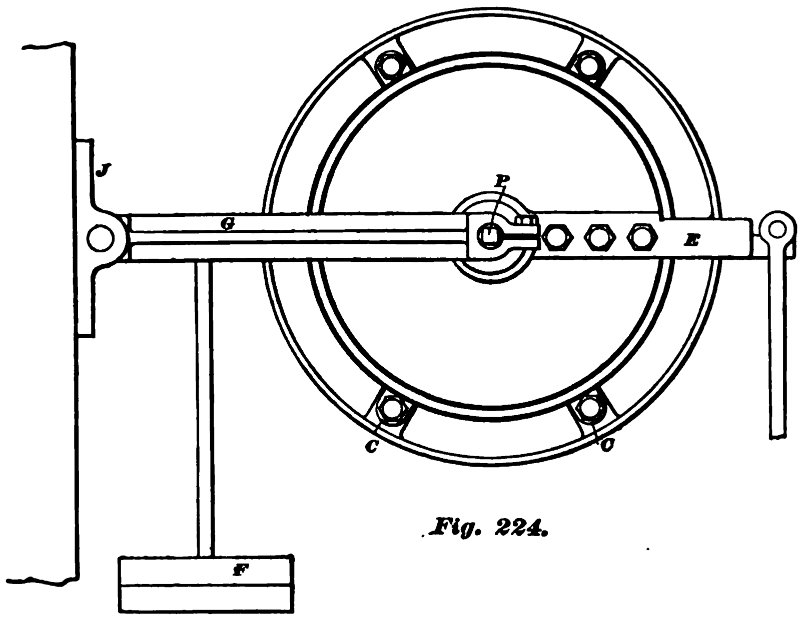


Fig. 224.

The shaft S carries a wheel W, having at uniform distances on its face sharp points, by means of which the revolution of the wheel gives a positive motion to a strip of paper, which is thus transferred from the spool L to another spool just behind L (Fig. 225), the paper being kept at a constant tension by means of small weights, which, acting upon cords wound around the spool axles, actuate the spools. The sharp points in the face of the wheel W leave prick marks upon the paper, the distance between two consecutive marks representing 50 revolutions of the driver. Upon one element of the face of the wheel W is a pair of points which serve to mark on the paper a length equal to the circum-

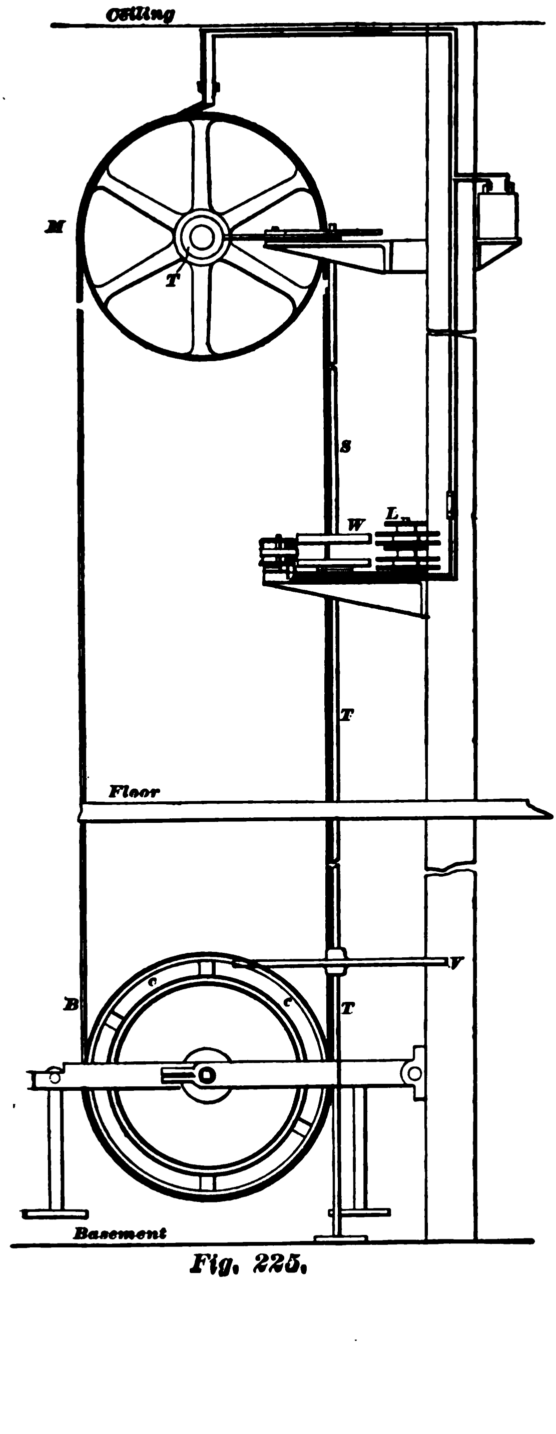


Fig. 225.

ference of the wheel W, this length representing 600 revolutions of the driver.

A pencil P carried by an arm l and pressed by a spring against the paper, as it moves with the face of the wheel W, records a line on the paper. As often as a certain point in the belt passes a given fixed point, the pencil is actuated by an electro-magnet, and records a notch in the line upon the paper.

The number of revolutions of the driven pulley B are recorded in a similar manner, except that as the driven pulley revolves on its axle, rotary motion is communicated to the shaft T by a cam c in the rim of the driven pulley, which cam operates a click which engages with a toothed wheel V upon the shaft T.

The machine is operated as follows:

The belt being adjusted upon the pulley, weights are placed in the pans, which, together with the weight of the machine, constitute the sum of the belt tensions.

These weights are so distributed in the two pans as to give the desired load.

Power is applied to the shaft of the driver and the machine attains its normal speed.

The city water is then admitted to the brake in the driven pulley, and is so regulated either by an automatic valve or by an attendant as to cause the friction of the brake to balance the weights in the pans.

The experimenter, standing near the recording apparatus, holds his watch and, at the instant he desires to begin the test, turns a switch and notes the time. During the test a button may be pressed at regular intervals, and a mark recorded. These marks divide the whole record into any desired number of sub-records, each complete in itself, but representing a shorter run.

The object of making these sub-records is to ascertain whether the total slip for the trial is uniform or intermittent.

The test is closed by throwing out the switch and noting the time. The two records are then examined and the data obtained may be written upon the ends of the ribbons, which are coiled and filed away for future reference.

In summarizing the advantages of the machine it may be noted—

First. That it requires no calibration except the weighing and measuring of the parts.

Second. It has no errors except such as are due to imperfect construction, and these may be reduced to any desired minimum

Third. It does not require during the trial, observations in which the errors or the personal equation of the experimenter are included.

Fourth. The axis of the driver being virtually above that of the driven pulley, the arc wrapped by the belt, on each pulley, can be accurately determined.

Fifth. It furnishes a complete and accurate record of the pulley revolutions and belt runs, from which the slip of the belt on each pulley may be computed.

Sixth. It has but few parts, and is operated with ease and convenience.

The discussion of the results obtained by experiments with the machine is reserved until more abundant data have been obtained.

DISCUSSION.

Prof. R. C. Carpenter.—It may be a matter of interest to know that through the courtesy of the Messrs. Wm. Sellers & Co., of Philadelphia, Sibley College of Cornell University was made the custodian of such parts of the transmission dynamometer that was used by Mr. Wilfred Lewis in the tests of belting and gearing, described in Vol. VII. of the *Transactions*, as were in existence at the opening of the year. The remainder of the machine has been rebuilt, very much the same as in the original form, the changes mainly being suggested by Mr. Lewis, although the details of brake and the slip recording device are essentially different from the form used by Mr. Lewis. This machine, it strikes me, is somewhat more complete than the one described by Prof. Alden, inasmuch as it provides a method of measuring the internal resistance of the machine and the loss of work due to stiffness of the belt, since the power received and delivered are both measured, and the slip can be read directly and to the minute fraction of a revolution, thus avoiding any errors due to computation. The transmission dynamometer is described in full in Vol. VII, page 274, and on page 550 is seen to be a form of a pillow block dynamometer, arranged so that the friction of its own bearings does not affect the determinations. The form of the machine used by Mr. Lewis is shown on page 551, Vol. VII., with weighing scales for the various data required in position.

The modified machine is shown in the drawing (Fig. 311) wit out the weighing scales in position. From this drawing it is se that the power received at P is transmitted through the dy mometer D, the force being weighed by a pair of scales plac at A, and the number of revolutions obtained by a continuo counter attached to the shaft (not shown in the drawing) and stop watch. The belt to be tested is put on the pullies E a F, and can be subjected to any strain required by tighteni the nut on the rod N, by which means the whole after part the machine is moved forward or backward on guides. T amount of this pull on the belt is read directly by a pair scales placed at C, the scale reading being to this pull as t horizontal arm K is to the vertical arm R. The absorpti dynamometer G is a Prony brake, of form designed by Pr Sweet, and much used in the Sibley laboratory. The pow transmitted is measured by a platform scale placed under t arm at B. The brake wheel has an internal flange, which filled with water by a pipe in the form of a crane's neck, and removed from the opposite side by a similar device. Such brake has been found to work very smoothly.

The device for measuring the slip consists of a shaft actuated by a worm gear, and carrying a graduated disk rotated by the shaft on the pulley P. At the other end of th shaft IJ, and near the disk S, is the wheel L, exactly equal the gearing at I, but free to move on the shaft IJ. To the hub the wheel L, a vernier may be clamped by the screw T, at a desired position. As the motion of the shaft IJ is slow, trouble whatever has been experienced in reading the verni when the machine was in motion; the amount of slip is sho by the variation of the vernier from its original position, th graduations being arranged to give the per cent. of slip direct

From these various measurements can be obtained the sum the tensions, difference of tensions, per cent. and amount of sli and arc of contact; and, finally, by computation, the coeffici of friction, by methods explained in full in the paper by M Lewis.*

The machine described by Prof. Alden is, it seems to me, very valuable adjunct to any laboratory, and is especially to

*Transactions American Society Mechanical Engineers, Vol. VII., page 550. Friction and Lubrication. By Thurston. New York: Wiley & Sons. Machinery and Mill Work: RANKINE.

The modified machine is shown in the drawing (Fig. 311) wit out the weighing scales in position. From this drawing it is se that the power received at P is transmitted through the dyn mometer D, the force being weighed by a pair of scales plac at A, and the number of revolutions obtained by a continuo counter attached to the shaft (not shown in the drawing) and stop watch. The belt to be tested is put on the pullies E a F, and can be subjected to any strain required by tighteni the nut on the rod N, by which means the whole after part the machine is moved forward or backward on guides. T amount of this pull on the belt is read directly by a pair scales placed at C, the scale reading being to this pull as t horizontal arm K is to the vertical arm R. The absorpti dynamometer G is a Prony brake, of form designed by Pr Sweet, and much used in the Sibley laboratory. The pow transmitted is measured by a platform scale placed under t arm at B. The brake wheel has an internal flange, which filled with water by a pipe in the form of a crane's neck, and removed from the opposite side by a similar device. Such brake has been found to work very smoothly.

The device for measuring the slip consists of a shaft actuated by a worm gear, and carrying a graduated disk rotated by the shaft on the pulley P. At the other end of th shaft IJ, and near the disk S, is the wheel L, exactly equal the gearing at I, but free to move on the shaft IJ. To the hub the wheel L, a vernier may be clamped by the screw T, at a desired position. As the motion of the shaft IJ is slow, trouble whatever has been experienced in reading the verni when the machine was in motion; the amount of slip is sho by the variation of the vernier from its original position, th graduations being arranged to give the per cent. of slip direct

From these various measurements can be obtained the sum the tensions, difference of tensions, per cent. and amount of sli and arc of contact; and, finally, by computation, the coeffici of friction, by methods explained in full in the paper by M Lewis.*

The machine described by Prof. Alden is, it seems to me, very valuable adjunct to any laboratory, and is especially to

* *Transactions* American Society Mechanical Engineers, Vol. VII., page 550. Friction and Lubrication. By Thurston. New York: Wiley & Sons. Machinery and Mill Work: RANKINE.

commended for its principal features, the automatic recording device and the absorption dynamometer.

This dynamometer, or, as termed here, the Alden brake, has been in almost constant use in the Laboratory of Sibley College the past year, and we have found it automatic in its action, cleanly, and so delicate as to be almost an instrument of precision.*

Mr. Wm. Kent.—I think the best part of this paper is the last paragraph: "The discussion of the results obtained by experiments with the machine is reserved until more abundant data have been obtained." When they have been obtained there will be something to discuss. There is no use in attempting to criticise the machine. It is probably all right. While the other paper on belts is not under discussion, I hope the author of it will soon give us some results of that dynamometer. I am especially interested in the result of dynamometers just now, for the reason that I have been engaged in a law suit in which it has been sworn to by a former member of this society—and I am glad to say he is not now a member of it—that dynamometers and Prony brakes are of no use, and that the proper way to measure the horse power was to measure the belt. I hope no other member will ever get on the witness stand and say such a thing as that. So, that for this reason, I am very anxious to get on record actual records of belt dynamometers. We have had discussions of this matter in the society before, and the chief thing we had to bring in opposition to this expert (in addition to our dynamometer tests) was to show that those whom we recognized as experts on belts, and who have read papers before this society on the subject of belts, were unable themselves to judge of the horse power of the belt without an actual test.

The particular belt in question was said by him to transmit 21 H P., as he knew from his experience and observation that he could estimate within 10% of the truth. After we had shown that that man had made a mistake in measuring the driving pulley, saying it was 36 instead of 30 inches, the other side brought another so-called expert, who, from his knowledge and experience, testified that that belt was transmitting 18 H.P., and he knew he was right, because his estimates never varied

*Transactions of the American Society Mechanical Engineers, Vol. XI., page 959.

from actual tests more than ten per cent. The next day after his testimony we put a Prony brake on the driven pulley, and we found that that belt could not transmit over 7 H P. Two years before it had been tested and found to be transmitting 7¼ H.P. So that the difference between testing by a Prony brake and guessing runs all the way from 7 to 21 H.P.

*Prof. G. I. Alden.**—I appreciate Mr. Kent's remarks, and in reply will say, that when I commenced the preparation of the paper, I hoped to have enough results to present in connection with the description of the machine.

I am glad Prof. Carpenter has referred to the belt testing machine of Messrs. Wm. Sellers & Co., and I think Sibley College fortunate in having secured so valuable a piece of apparatus. I am sure we are all much indebted to Mr. Wilfred Lewis for the extensive and careful experiments made with the machine referred to, and published in Vol VII. of our Transactions. It seems desirable that the different engineering laboratories of the country should work independently upon various problems in which data of practical value may be obtained, not only for the purpose of comparing results to establish truth or reveal error, but also for the purpose of making the students in these laboratories familiar with results, and also with the experimental work by means of which the results are reached.

It was the need in the laboratory of the Worcester Polytechnic Institute of some apparatus for testing belts that led to the construction of the machine I have described. With reference to measuring internal resistances and loss of work due to stiffness of belt, I suppose a transmission dynamometer would be required for that purpose with any machine, though that would be likely to introduce some unmeasured resistances, such as axle friction, which it would be difficult to determine separate from the work due to stiffness of belt.

The machine which I have described could readily be driven through a transmission dynamometer. As used for ordinary testing, the total work done at the face of the driven pulley is measured, including axle friction.

The use of the absorption dynamometer in the driven wheel enables the machine to be made very simple, both in construction and operation. The measurement of the slip on both the

*Author's Closure.

driving and the driven pulleys is a feature of interest in the results.

The elimination of scale beams with dashpots is also desirable, as it removes the necessity of constantly taking weighings from a vibrating beam.

CCCCXLV.*

STEEL CASTINGS.

H. L. GANTT, PHILADELPHIA, PA.
(Member of the Society.)

During the past few years steel castings have come extensively into use in the construction of machinery. The great power and size of many machines now being built demand a metal stronger than cast iron, and steel seems to be the most suitable substitute. Steel castings, however, have not given universal satisfaction by any means.

One man complains that they are not true to pattern; he has been using iron castings and expects those of steel to be as accurate as the iron ones. A little consideration will show that such expectation is unreasonable. Steel castings not only shrink much more than iron ones, but with less regularity. If iron castings are made from approximately the same grade of metal, they will be very nearly uniform in size. With steel castings the amount of shrinkage varies with the composition and the heat of the metal: the hotter the metal, the greater the shrinkage; and, as we get smoother castings from hot metal, it is better to make allowance for large shrinkage and pour the metal as hot as possible. Then all castings should be thoroughly annealed, and in doing this a heavy scale is formed, leaving the casting pitted; this is very marked if the metal has been a little cool in pouring. We must not therefore expect to find steel castings as true to pattern as iron ones. The safest thing to do is to allow three-sixteenths or one-quarter inch per foot in length for shrinkage, except in very heavy castings, where one-eighth inch per foot will generally be sufficient and one-quarter inch for finish on all machined surfaces except such as are cast "up." Cope surfaces which are to be machined should, in large or hard castings, have an allowance of from three-eighths to one-half inch for finish, as a large mass of metal slowly rising in a mould is apt to become crusty on the

* Presented at the Providence Meeting of the American Society of Mechanical Engineers (1891), and forming part of Volume XII. of the *Transactions*.

surface, and such a crust is sure to be full of imperfections. On small, soft castings one-eighth inch on drag side and one-quarter inch on cope side will be sufficient. No core should have less than one-quarter inch finish on a side, and very large ones should have as much as one-half inch on a side. This seems a great deal, but will be found economical in the long run.

Another man will say that so many castings have "blow holes" in them. This is undoubtedly true; but unless the holes are so large as to make the appearance of the casting very bad, or occur on a wearing surface, they should not give any anxiety, as such metal is always tougher than solid metal of the same hardness. Blow holes can be entirely prevented in castings by the addition of manganese and silicon in sufficient quantities, but both of these cause brittleness, and it is the object of the conscientious steel maker to put no more manganese and silicon in his steel than is just sufficient to make it solid. Working with this object in view, he occasionally fails to get quite enough in to make his castings entirely free from blow holes, while one who is simply aiming at solid castings regardless of quality, gives his metal a dose of silicon which is sure to make it solid; and if there should already be a little more silicon in the metal than he is counting upon, his castings will be beautifully smooth and solid, but brittle and far inferior to those with the blow holes in them. If such castings are to replace iron ones which have broken, they will, as a rule, be good enough; but if they are for new work and good tough castings have been counted upon, they will be very apt to fail.

Another complaint is that many of the heavy sections on being cut into are found to contain large shrinkage holes. This is true, but most people do not consider the excessive shrinkage of steel in designing their castings, but make them just as if they were to be cast of iron. An iron founder can always get hot metal from his cupola to feed any casting as long as it continues to shrink. The steel founder taps all the metal from his furnace at once and it is necessary that the casting shall be so designed and the sinking heads so placed that it shall need no more feeding. In order to get such a casting solid it must cool from the bottom up, and the sinking heads must be the last portions to chill or set. In other words, every portion of the casting must have a fluid connection with the sinking head until it is solid. The necessity, therefore, for designing steel castings in such a way that the heavy portions may be cast "up" with sinking heads on them is much greater than in cast iron, for many a section which in cast iron would have scarcely any shrinkage hole in it, would in steel be so badly shrunk as to be unfit for use. It will follow also that the section of the sinking head must be greater than any portion of the casting to enable it to retain its heat until the whole of the casting is set.

The best results are arrived at when all portions of the castings are of a uniform thickness, or very nearly so. In such a case there is but little danger of shrinkage holes, and the even cooling will, as a rule, cause the casting to retain its shape and be true to pattern; while those in which the sections vary very much in thickness are apt to depart a good deal from the pattern by warping from unequal cooling.

Another and very serious trouble is the tendency which castings have to crack in cooling. These cracks, or "pulls" as they are generally known, take place just after the steel has passed to the solid state and when it is still greatly lacking in tenacity. The contraction of the casting begins very soon after it has become solid, and unless every portion is free to shrink, there may not be strength enough in the metal to overcome the resistance offered by the sand in the mould and to pull the different parts together, and a crack will open in the hottest or softest part. If there is much variation in the thickness of the metal, it is almost impossible to prevent the formation of cracks where heavy and light sections join. If, on the other hand, the casting is of uniform thickness throughout, such a tendency is reduced to a minimum, as there are no very soft places for cracks to start. All inside corners should have large fillets in them, as such corners when sharp almost always crack open. The tendency of castings to "pull" makes it very desirable that they should be as simple in construction as possible. For instance, it is better to bolt several small castings to one large one than to complicate the large one by casting small projections on it. In case of the built-up structure we know exactly what we have, but when a number of brackets and knobs are cast on, we may never know the cracks or defects until something breaks off.

The principal cause of the difficulties in filling specifications on castings for the United States Navy Department is that all designs are made, not with special reference to avoid difficulties in casting but with the purpose of getting the strength needed with the least possible weight of metal. The result is that the castings are often very complicated, and, as in making such castings each new shape is necessarily an experiment, they are very expensive. In many cases, also, there are no doubt shrinkage holes in the centres of the castings, but usually such holes are not suspected until the casting breaks. In the majority of such cases the steel founder has done all he can to prevent shrinkage, but the design of the casting is such as to cut off fluid connection from the sinking head to some portions of the castings before such portions are set. Steel founders should decline to make such castings as will surely contain serious shrinkage holes, or at least warn their customers that such will be the case, and make an effort to have the design modified. It is a good plan for those not practically familiar with the manufacture of steel castings to submit to those who are, designs of any large or important castings before making the patterns, as in many cases unimportant changes may be made in the design and much better castings secured. Work will often be much expedited by consulting the founder about the way of making the pattern, as it is often necessary to pour a steel casting in a position very different from that in which an iron one would be cast.

In an earlier part of this paper it was stated that the fracture of an unannealed steel casting very much resembled that of burnt steel. The behavior of an unannealed steel casting also resembles that of an overheated forging. Its chief characteristic is its brittleness when subjected to shock. Hard castings have this property to such a marked degree that sinking heads are often broken off by the shock of chipping off the runner, in which case they nearly always carry a large piece of the casting with them. Such castings, unless containing large amounts of manganese and silicon, become tough and reliable when annealed, and show a fine even grain upon fracture. Another effect of annealing is to equalize the strains of cooling. These strains are sometimes so great as to cause the casting to crack open with a loud report. This happens not only in hard steel, but also in steel low in carbon. As an illustration of this the cracking of a large gear-wheel may be cited weighing 8,000 lbs. (carbon .34%). The wheel had been cast several days and was about half-cleaned. It was left in a perfect condition one evening and found next morning with a crack halfway across one of the arms.

The following table will illustrate the effect of annealing on tensile strength and elongation:

Carbon.	UNANNEALED.		ANNEALED.	
	Tensile strength.	Elongation.	Tensile strength.	Elongation.
.23%	68,738	22.40%	67,210	31.40%
.37%	85,540	8.20%	82,228	21.80%
.48%	90,121	2.85%	106,415	9 80%

In the above table the unannealed and annealed specimens having the same carbon were in each case taken from the same runner of a casting and very close to each other. On considering this table the value of annealing becomes immediately evident, as does the absolute necessity for it in case of hard castings. The proper annealing of large castings takes nearly a week, and when we remember that all steel castings are made in dry sand moulds, and have sinking heads on them which in most cases have to be cut off in the machine shop, it will be evident that they should be ordered three or four weeks before they are needed.

One more complaint of frequent occurrence is that the metal is too hard or too soft, no specifications having been given except that the castings shall be of steel. In such a case the manufacturer has probably put in the casting the grade of metal which he considered best for the work it was to do, but his judgment will often differ from that of the purchaser. Our experience with steel gears, and especially with roll pinions, is that they do not break, as a rule, but wear out. Consequently we make them hard, pinions being made hardest. The result of this has been that some have complained that they were so hard they could not be machined. For people who have their machines speeded to cut cast iron only, they are of course too hard, but if it is desired to get good wear it is much better to slow down the machine and use hard steel for such castings. One objection to hard steel is the tendency which it has to form blow holes, and consequently the strong inducement to add large quantities of silicon and manganese to the metal to make it solid. It is in high carbon steels that silicon is specially objectionable, for while annealing will greatly diminish brittleness due to carbon, it does not appear to have much effect on that due to silicon. Another essential in the production of solid castings of high carbon steel is that the metal shall be very hot. Consequently small castings and those having very thin sections will often contain blow holes, while a thick, heavy casting made from the same metal will be perfectly solid. As large and small castings are usually made from the same ladle of metal, it is considered preferable to run the risk of having a few blow holes in small castings rather than have the large ones unnecessarily brittle. Such blow holes are usually very close to the surface and may in general be gotten rid of by allowing a little extra metal for machining. In the case of small gearing they show up in the teeth, but our experience has been that such gears wear far better than softer ones, which can easily be made solid.

The proper steel for roll pinions, hammer dies, etc., seems to be that containing about .60% of carbon. Such castings properly annealed have worn well and seldom broken. Miscellaneous gearing should contain carbon .40% to .60%, gears large in diameter being softest. General machinery castings should, as a rule, contain less than .40% of carbon, those exposed to great shocks containing as low as .20% of carbon. Such castings will give a tensile strength of from 60,000 to 80,000 lbs. per square inch and at least 15% extension in a two-inch long specimen.

Machinery and hull castings for the war vessels now being built for the United States Navy, as well as carriages for naval guns, contain from .20% to .30% of carbon. The smaller gun carriages are subjected to a ballistic test, in which the casting is pierced by two shots from a three-pounder Hotchkiss rapid firing gun, the shots striking not less than four diameters apart. If the metal shows a tendency to crack seriously, the casting is condemned, as are all others made from the same heat. Castings for the hull and machinery of the ships are subjected to a bending test, in which an inch square bar is bent at right angles around a radius of 1½ inches. If the bar shows any serious cracks the casting is condemned.

In designing machinery the question often arises as to which will give the desired result most economically, a steel or an iron casting, and is sometimes quite difficult to settle without making both designs and getting prices. If a few facts about the relative size of steel and iron castings of the same strength be borne in mind they will greatly help us in this matter.

With reference to strength, it has been noted that good castings for general machinery work will give 60,000 to 80,000 lbs. tensile strength and 15% extension. Where strength and not stiffness is required, a structure of such material will resist a strain of 15,000 lbs. per square inch with greater safety than one of cast iron will resist 5,000 lbs.; especially if subjected to shocks. It follows, then, that if the castings are large, a steel casting to stand the same strain as an iron one need only be about one-third as heavy. In the case of small castings the use of steel will generally allow us to make them somewhat lighter, but the same proportion will not hold good, as the difficulty of making a perfect steel casting increases very rapidly as the sections become very thin. In general small steel castings will be found to cost proportionately more and to be less satisfactory than large ones. They come into use where iron castings will not answer, while large steel castings will often take the place of iron ones on the score of cheapness alone.

The following is a partial list of castings in which steel seems to be rapidly taking the place of iron: hydraulic cylinders, crossheads and pistons for large engines, roughing rolls, rolling mill spindles, coupling boxes, roll pinions, gearing, hammer heads and dies, riveter stakes, castings for ships, car couplers, etc.

Perhaps the one class of castings which is of most general interest is that of gearing. A few years ago steel castings were so rough that it was necessary to cut all steel gears; but now this difficulty has been practically overcome, and while they will never be as smooth as good iron ones, uncut steel gears, even of small sizes, are giving perfect satisfaction in many places, and are now used by the Midvale Steel Company almost to the entire exclusion of iron ones.

DISCUSSION.

Mr. Wm. Webber.—Mr. Gantt has not mentioned the most prolific objection to steel castings, and that is in the using of steel castings in general manufacturing business, the steel foundry promises a steel casting in three weeks, and if you get it in three months, you are doing quite well.

Mr. Oberlin Smith.—I have used a good many steel castings, and my experience is that they are apt to be either good or bad, and you either do or do not know which. On the whole, I have had very good luck with them. Really, "luck" is the proper word; for the makers do not seem to be playing a game of skill. In shafts for presses and other percussive machinery, I have subjected them to severe tests and have rarely had one break. When you get them of an old maker with whom you have dealt a long time, and at *long* intervals, as Mr. Webber suggests, you generally get pretty good results. As before stated, I have had very few breaks, and most of the blow holes have been on the surface. Of course, when I find 500 or 600 on a shaft 2 feet long and 3 inches in diameter, as I have done, I throw it out. But shafts of that kind I usually get from new men, who have not been in the business long. A peculiarity about this business is that when you get pretty well established and are running along nicely, some new man will come to you with a story so plausible that you will change and buy of him; and then in the course of three or four months you will change back again, and you would change in three weeks if you could. But, seriously, they do answer an excellent purpose for cranks and many other kinds of work. I think there is a decided improvement in them, noticeable of late years—provided you always avoid the new men.

Mr. Jos. Leon Gobeille.—The fault with many steel castings lies largely with the man who designs the shape. A complicated form will not come out satisfactory in one case out of five. This fact makes a very great difference in foundry practice for iron and for steel, and the man who has charge of the moulding has to put in much more of his time thinking how he can get out what is furnished to him. I think the manufacturers are working against themselves in designing complicated forms.

Mr. E. F. C. Davis.—I think it is only due to some manufacturers to say that there are some people in this country who can make steel castings with some degree of certainty. In getting steel castings for the engines of the *U. S. S. Texas*, we have sent back very few castings, and some of them were very large and very complicated. They have had very few blow holes, and those that were in them were so slight that they did not do any harm. The quality of the castings has been very good indeed. The patterns were changed slightly from the original design, but I think the result has been very satisfactory, and shows that the manufacture of steel castings has arrived at a degree of efficiency which is not generally believed. We have had a great deal more difficulty in getting steel plates for the *Texas* boilers than we have had for steel castings.

Mr. Wm. Kent.—I would like to ask Mr. Gantt if it is now the custom to immure steel castings in oxide, and why it takes a week?

Mr. Wm. H. Weighsman.—The Westinghouse Brake Company are using cast-steel fittings throughout, and have been for three years.

Mr. C. H. Parker.—I am glad to hear a good word spoken in defence of blow holes in steel castings. I have recently passed through rather a serious experience with steel castings on some machines which we are making for the Government, intended for the United States cruiser *Monterey*. The appearance of the castings was not very satisfactory. The steel castings were to withstand a working steam pressure of 160 lbs. to the square inch, and the specifications called for a test pressure of 240 lbs. to the square inch, and finally they had to pass the government inspector's approval. In order to be sure, I built the whole machine and put it together, and tested it at 300 lbs. to the square inch, and of course where there were blow holes, they manifested themselves, but not to such a degree that they could not have been stopped by very small plugs, or some of them even by the hammer. I got the work in a condition which I thought would meet the government inspector's approval, and he came and examined the work, saw the test of 300 lbs. to the inch. Of course, I had not taken any measures to hide the blow holes. I left them in their natural condition. There was no trouble in maintaining the pressure with a small pump of about half an inch plunger. The cubical contents of the vessel filled was about 12 feet. He expressed his satisfaction with everything, so far as the test went, but he said that the Government would not under any circumstances accept castings with blow holes, and that settled the whole thing, and the work was condemned at a loss of some $500 or $600. Now, those blow holes were very small in magnitude, and could have been easily stopped ; and, if a blow hole is an indication that a casting is low in silicon, I cannot see why the requirement that there should be no blow holes should condemn a piece of work which is otherwise perfectly satisfactory. The castings were rough in appearance, but that could have been easily remedied. My object was to lay out no more labor upon them than was absolutely necessary in order to have the government inspector accept them.

Mr. Oberlin Smith.—One feature about steel castings which is particularly good is their adaptability for shafts with cams on them, running against rollers. In many cases where I could not make a shaft of good hard cast-iron stand at all, steel castings under the same conditions have lasted for years without a scratch or a mark, the rollers and cams both getting all the time burnished down to a harder and smoother surface. Cast-iron in the same circumstances became crushed upon the surface, and quite rough. I have also found that steel casting shafts wear a great deal better, running in cast-iron bearings, than steel forgings of any kind, under the same conditions. In the case where a steel forging (of all the way from .30% to .50% carbon) would "cut" under the pressure, a steel casting would remain perfectly smooth. It seems to have more of a self-lubricating nature than either cast-iron or steel forging.

Mr. H. M. Howe.—Mr. Gantt speaks of the shrinkage varying with the composition of the metal for given casting temperature. Is that an accurately observed fact, or simply a matter of belief? It seems to me rather surprising that any variation of composition within usual limits should induce a change in the dilatation so great as to be easily seen in the manufacture. Then I notice that he also says in the paper that blow holes are almost necessary if we are going to get a tough casting. I do not think that is quite true. No doubt the presence of blow holes would be an indication that the casting was probably tougher than if the casting was perfectly solid. I meant to bring here some shells which are perfectly solid and free from blow holes and at the same time pretty tough—a shell so tough that when it is fired under full charge against earth work without breaking, of course must be a pretty tough casting. Those are made in enormous quantities and tested in that way with few if any failures. Those castings are perfectly solid. Then I have photographs of some of Hadfield's cast shells which have been neither hammered nor rolled, though they have undergone a little mechanical work. One is of a thirteen-inch shell which has penetrated two feet into wrought iron. That, of course, must have been pretty tough, and if it was tough after only undergoing this very slight amount of work, it is reasonable to suppose that it was also pretty tough before and when it was simply a casting.

I think that the making of steel castings is a matter of trick and knack a great deal, and one of the most important of these is the one which is dwelt upon so much—providing plenty of feeding for the pipe. There is one thing which is not mentioned in this paper which I think is often a very good plan, and that is to have hot tops to your moulds. It is an old trick, but one which has not been put in operation as much as it might be. At the top of your mould place a cap which has been previously heated to a white heat. That is done in Sheffield, England, even in casting ingots, and is very much more applicable to castings. Favor the feeding by putting at the top of the mould a mouthpiece which is previously very highly heated, so that the metal there will keep perfectly hot, and feed after the metal below has become solid. Another well known trick for preventing cracks where the section changes abruptly, *e. g.*, where the spoke of a wheel joins the rim, is to cast in a fin, which, cooling early, will become hard and strong when the rest of the metal is still weak, and liable to pull apart owing to unequal contraction—a bracket, it is called technically. This fin is readily chipped out later.

Mr. Jno. T. Hawkins.—I have no desire to blow any more holes in steel castings than are usually found there, so I will not touch upon that part of the question, but I desire to say that in my opinion the engineer who will succeed in correcting the erratic character of steel castings as to shrinkage, will have done a good thing. Mr. Gantt's suggestion of leaving a greater amount of metal on fitting surfaces to correct the irregular shrinkage, is very well on such character of work as will permit of it, but on a very large variety of work that is not permissible. I had occasion quite recently to attempt to adopt steel castings in a very important part of a machine which required that the castings should be of uniform size. We experimented in order to get our patterns just right, and after a good many trials, carrying out the injunctions of the steel casting people, we thought that we had got the thing down very nicely, and were satisfied that we had arrived at a point where we were justified in ordering a pretty large invoice of these castings, when to our surprise they were all too small,—the shrinkage had gone back to where the original one was—and we were obliged to abandon it, and the profit on these particular machines shrunk in proportion. There are very many places in machinery where steel castings may be made to serve a very valuable purpose if that unfortunate peculiarity in them can be eliminated, where they are now prohibited because of it.

Mr. T. R. Morgan, Sr.—We have had considerable experience with steel castings at our works from the inception of the business in this country to the present time. Castings have come to us from all the leading steel foundries both of the lightest and heaviest grades, the plainest and the most complicated: from the Pittsburgh Steel Casting Co.; Mackintosh, Hemphill

& Co.; Carnegie, Phipps & Co., Pittsburgh; Cambria Iron Co., Johnstown; Sharon Steel Casting Co., and Wellman Iron & Steel Co.; Thurlow, Midvale Steel Co., Nicetown, Philadelphia, Pa.; The Otis Iron & Steel Co., Cleveland, and the Solid Steel Co., Alliance, Ohio.

From one and all we have had many excellent castings in material and solidity. The difficulties in making good steel castings are, in my opinion, more in our mind than they are in fact, as verified by our experience.

We know the making of steel castings is a new and high class branch of business which requires chemical and practical care combined, the practice varying considerably in many cases from that of iron founding, since the high temperature of steel when casting, with its quick cooling and great contraction at the same time, compel better judgment to obtain similar comparative results than iron founding requires. This settles itself when the steel mixture is in proper condition to no other than a good industrial organization governed by good leaders, a good sober class of workmen with good common sense, and records well kept both practical and theoretical. Steel castings can and have been as well made as the best iron castings.

For some classes of steel castings special care and good judgment are required in designs and patterns, as well as in feeding, much more so than in iron castings; apart from this, many iron castings of a complicated character, when made in green sand, are more difficult to make than moulds for steel castings of the same kind which are all dried to a cake before casting.

We have had some plain steel castings made which were excellent and solid throughout; a number following from the same pattern would not compare with the first lot, some lacking solidity and workmanship both.

Believing, as I do, that like has produced like, and always will to the end of time, I must and will always feel that when such comparative conditions adverse to the new industry show themselves, the fault is not in the business itself, but in the lack of constant conditions, which have not been obtained in any business of a high character until a high organization of management and workmen has been obtained, each carrying out their respective parts from long and co-operative experience.

While in all lines of business I believe there is room for improvement, I believe America has done wonderfully well in the steel casting line, and our people are continually making great inroads into the cast-iron casting business and will continue to do so in the future.

The United States Government has received and is continuing to receive some of the best, most perfect, and complicated of steel castings, guaranteed to high tests, and prices accordingly, but the government rates are such as are entirely too high for commercial purposes, as the laws of demand and competition will not admit of them.

Iron flasks for moulds for steel castings are needed, which has added to the cost of such castings. The time will come, and that before long, I believe, when this country will correct these conditions as it has in other departments of manufacture, through having accumulated ample facilities and proper organization, so that steel castings will be produced in the best manner at much more reasonable rates than at present; especially plain castings requiring only plain labor and its conditions. When that is done steel castings will make continuous and rapid inroads into the iron casting business, as steel castings in tenacity and endurance average about four to one of the cast-iron; so I feel thoroughly warranted in saying that any difficulties which may seem to come in the way of obtaining good sound steel castings are only similar to difficulties which have always existed in any other business which has failed at first to produce first-class work. Some of the difficulties are in the science and the management, but many are in the practical and commercial side of the business, but these will all be corrected in time in all first-class establishments.

Mr. H. H. Suplee.—I think Mr. Morgan's remark ought to be followed up by the definition of engineering, which was made by the late Mr. James Nasmyth: "Engineering is common sense applied to the use of materials."

Mr. O. C. Woolson.—I have had some experience with steel castings, and I am inclined to emphasize one or two points which Mr. Gantt has mentioned in his paper. One is the great necessity of ample sink-heads—usually much larger than required for gray iron castings. And he mentions another thing: he says that the castings required by the United States Navy are very difficult to make, because they are so complicated, and he advises the bolting on of such projections, lugs, etc., as can be, to simplify the main casting. That is all right. The complicated requirements that the government draughtsman sometimes specifies, remind me of an experiment which I had with some government work on one of their light-houses. I had great difficulty in erecting it at our works, inasmuch as in numerous places it required 3 inch steel pins in 2½ inch holes, and I had the greatest difficulty in filling such requirements.

Mr. Robert W. Hunt.—I think we all recognize that steel castings have come to stay, and the subject is of great interest. It has been shown that steel casting has passed the experimental stage. Of course, there are still many difficulties to be overcome. It is quite true, that when you are getting an order duplicated from the manufacturer, it is natural that you should have less difficulty with the goods furnished than if you go to one that has not had the same experience. Do you recall how it was with steel rails? A steel rail broke, and all steel rails were condemned They forgot that where one steel rail broke, a hundred iron ones had failed. Do we not have some trouble with iron castings?

Mr. Gantt.—I think Mr. Morgan is just about right. I did not mean to say that those complicated castings could not be made. They can be made, but every one of them is an experiment, and somebody has to pay for that experiment, and the ordinary manufacturer won't do it. Mr. Davis said he had been getting good castings for the *U. S. S. Texas*, and had sent but few back for defects. He did not tell us, however, how many of them were condemned by the inspector at the works where they were made. I happen to know in some cases how that was, and so I won't ask him to tell. Now, about blow holes. There is no more difficulty in getting your steel casting perfectly sound than there is in an iron casting. If poured quickly with hot metal containing the proper amount of silicon and manganese, they will nearly always be solid, and a careful consideration of the paper will show that Mr. Howe has misunderstood me; I did not mean to say that castings could not be made solid, that I preferred a casting containing blow holes to one containing an excess of manganese and silicon. With regard to annealing, I did not say anything about that before, but Mr. Kent has asked a question which has brought it up, why it takes a week? Well, it does not take a week to anneal one casting, unless it is a very large one. There is no use in running any risks on castings. They must be heated slowly. They nearly all have a scale on them when they come out of the furnace, which has to be taken off, and this requires a certain amount of time. With regard to deliveries on castings, that is, of course, a matter with every manufacturer. I do not think that the deliveries on steel castings are much worse than they are on other things. Mr. Howe's suggestion of a hot-top for the mould is possibly a good thing. But suppose you have fifty moulds to pour from a ladle of steel; it would be a difficult job to have a different top for each one of those moulds. Then the question of fins to keep castings from cracking and pulling—everybody does that; not only that, but we put a big fillet in every sharp inside corner. I have seen a skin of the casting perfectly solid, but when it was cut through, a hole showed underneath. Now, you can get good castings if you make the design right. With regard to the strength, I would say that steel casting, when constructed right, will do three times the work of a cast-iron casting. I don't believe there is any luck in it at all. Now, as to the question of records. That is a thing which requires a higher intelligence than you can find in the ordinary moulder. The only way to do is, to keep a record of how every difficult casting comes out, and when you get a good one keep that record and follow the same plan afterwards.

CCCCXLVI.*

AN ADDITIONAL CONTRIBUTION TO THE PERFECT SCREW PROBLEM.

BY W. A. ROGERS, WATERVILLE, ME.

(Member of the Society.)

THE writer presented a paper at the Pittsburgh meeting under the title, "A Practical Solution of the Perfect Screw Problem."† This title is in some respects a misnomer. The substitution of the word impracticable for practical would more nearly represent the process described. While it is possible to make a precision screw by this method, under certain well defined conditions, of which the screw of the dividing engine made for Cornell University may be taken as an example, the requisite conditions are not easily fulfilled in an ordinary work-shop. It is not surprising, therefore, that the manufacture of screws by this method was not a commercial success, especially as the writer was unable to exercise personal supervision of the work.

About three years since the writer returned to the attack of this fascinating problem. The result of these later investigations will appear in the screw-cutting engine shown in Fig. 227. This machine was built by Messrs. Webber & Philbrick, of Waterville, Me., and the plans were drawn by Mr. F. B. Philbrick of this firm. It has the following dimensions:

1. Length of bed....................................11 feet.
2. Width of bed....................................18 inches.
3. Depth of bed....................................14 inches.
4. Thickness of walls..............................$2\frac{3}{4}$ inches.
5. Width of horizontal ways........................$1\frac{1}{4}$ inches.
6. Diameter of spindle.............................4 inches.
7. Diameter of aperture in spindle.................3 inches.
8. Length of spindle in head-stock.................38 inches.
9. Length of spindle in tail-stock.................27 inches.
10. Length of screw................................8 feet 4 inches.

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. of the *Transactions*.

† *Transactions Amer. Soc. Mech. Engineers*, Vol. V., p. 146.

11.	Diameter of screw	3 inches.
12.	Pitch of screw	½ inch.
13.	Length of tool-carriage	30 inches.
14.	Width of tool-carriage	12 inches.
15.	Weight of engine	4,000 lbs.
16.	Weight of carriage	500 lbs.

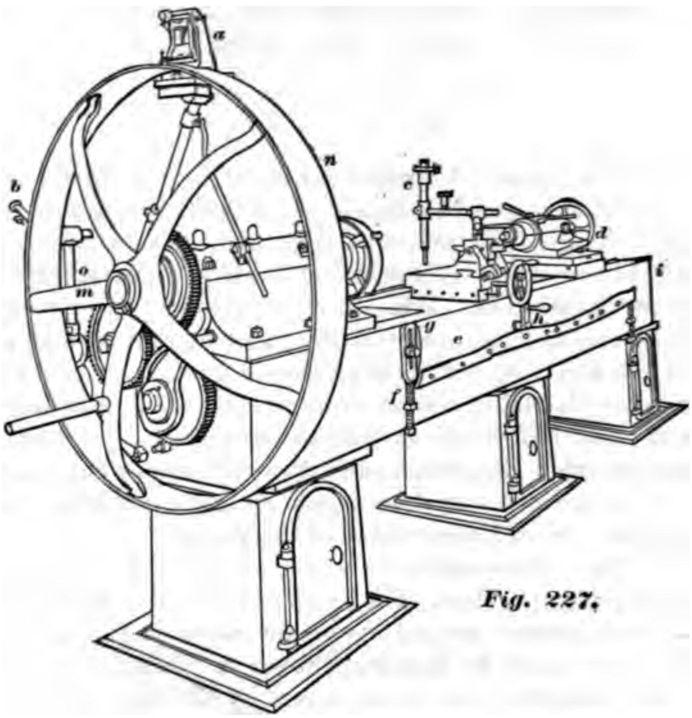


Fig. 227.

An attempt has been made in the construction of this engine to fulfil the following conditions:

(*a*) The ways upon which the tool-carriage moves must lie in a horizontal plane.

(*b*) The vertical wall against which the tool-carriage is pressed, by means of bolts backed by spiral springs, must lie in a vertical plane, which is at right angles to the horizontal ways at every point.

(*c*) The line between the centres must be parallel to both sets of ways.

(*d*) The movement of the tool-carriage must be parallel with the line between the centres at every point.

(*e*) The systematic errors of the screw, which are a function of the single revolution, must be eliminated independently of any method by which the correction of the linear errors of the screw is secured.

(*f*) The correction of the linear errors, including the reduction to 62° Fahr., must be made by a method which admits taking into account the wear of the screw.

In the fulfilment of condition (*a*) a short spirit level, made upon a special order by Fauth & Co., of Washington, was used. The value of one division of this level is about .9 of a second of arc, and the lower face of the base upon which it is mounted is a plane surface. The condition of the surface of the ways was further tested by comparing it with a surface of mercury 8 feet in length.

The fulfilment of condition (*b*) was found to offer the most serious obstacle in the construction of the machine. It is ordinarily assumed that the mean of two nearly parallel lines ruled by a motion of the tool-carriage and with reversed portions of the bar will be a straight line. This, however, is only true when the two lines ruled follow well-defined curves. Moreover, it was found very difficult to prevent the shear of the cutting-tool. The form of a comparator, constructed by the writer, admits of the construction of a line 40 inches in length, which is practically a straight line. With the aid of such a line traced upon a horizontal surface, it was found easy to test the plane of the vertical way. At present this way has a very slight curvature at two points, the maximum deviation from the vertical plane being .0002 inch and .0004 inch respectively.

Condition (*c*) is fulfilled by giving to the frame which carries the centre in the tail-stock a universal adjustment.

The writer will not take time to describe the method by which the systematic errors of a single revolution of the screw were eliminated. It may be said, however, that the method is applicable to screws of almost any degree of badness. In the present case the errors of this class, which existed in the screw of a lathe of well-known make, which is the basis of the present screw, were very large. They have been reduced in amount to such an extent that they are now barely measurable under the microscope.

The linear errors were eliminated by the template shown at the point *g*, Fig. 227. The reduction to 62° is accomplished by a fall of the template amounting to two inches in about eight feet. It was the original intention to replace the present screw with a new one in which the existing errors would be eliminated, but the ease and certainty with which a transcript of the corrected screw can be made, has so far rendered this unnecessary.

The test of the screw is made in the following way: A space of 40 inches has been subdivided to single inches upon the plane surface of a bar of Jessop's steel. By repeated approximations the total length has been made correct at 62° Fahr., and the errors of subdivision have been so far reduced that they can be neglected. This bar is placed upon the bed-plate of the machine and so adjusted that every line is in sharp focus under a microscope attached to the tool-carriage. Making a coincidence of the micrometer line of the microscope with the initial line of the bar, every fifth revolution of the screw should bring coincidences with the successive lines. The amount of the deviation is measured with the filar micrometer. At no point within the limits of 6 feet does this deviation exceed one-six-thousandth of an inch.

The reading microscope is shown at *b*. At *a* will be seen the marking apparatus with which the index-wheel was graduated into 1,000 equal parts. *c* is a microscope attached to the tool-carriage. The template is shown at *g*. The back thrust of the screw is against a plane surface at *i*.

CCCCXLVII.*

STEAM-ENGINE EFFICIENCIES: THE IDEAL ENGINE COMPARED WITH THE REAL ENGINE.

BY R. H. THURSTON, ITHACA, N. Y.
(Member of the Society and Past President.)

THE writer has been much interested of late in making a number of comparisons of the efficiencies computed for the Ideal Thermodynamic Case of the Steam-engine, as by the exact methods of Rankine or of Clausius, with those obtained when the conditions which distinguish the actual from the hypothetical engine are introduced. The ideal engine, as the term is here employed, is that which is treated of in all purely thermodynamic studies of the engine as free from those wastes of heat and of power which distinguish the operation of the real engine as a consequence of the existence of friction and the action of conduction and radiation in the dispersing, in useless ways, of heat which should be applied with best effect in thermodynamic transformations. The ideal engine is one which is assumed to have a non-conducting cylinder and frictionless rubbing parts. Its wastes are simply those which are unavoidable and characteristic in all thermodynamic operations. The real engine, on the other hand, is not only subject to those wastes by unavoidable rejection of heat at the inferior limit of adiabatic expansion, but loses heat by conduction and radiation to surrounding objects in considerable amount, and by internal transfers without transformation in very large amount, often, and also wastes a serious percentage of the power actually obtained, by transformation out of heat energy, in the work of overcoming the friction of its rubbing parts, and of forcibly rejecting the working fluid against an artificially produced back-pressure.

In the time of Rankine and Clausius, only the thermodynamic or ideal case was considered in the theory of the engine, the

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. of the *Transactions*.

discussion of these mechanical and physical changes and wastes of energy only having been introduced, to form a complete theory of the actual case, since their time. The theory of the real engine is thus a mixed theory, involving thermodynamics, physics of heat, and the dynamics of friction. The following interesting computations, made for the writer by Mr. Kuehmsted, illustrate the sometimes great differences between the purely thermodynamic efficiency of the engine and the actual efficiencies as modified by these later extensions of the theory to adapt it to actual practice. The actual figures presented are not so much of importance, since they may or may not accord with any given cases in practice, as the conditions here assumed may approximate more or less closely to those observed in the actual case. The main interest attaches to their illustration of the general effect of such modification in the development of the modern theory of the engine, and the exhibition of the method of variation of the efficiency and economy of the engine, with changing conditions as to extent of expansion of the steam and the power of engine.

The cases to be investigated in this provisional and approximate manner are those which may possibly represent a later practice than is yet reached, the steam-pressure being what is now regarded as above the usually practicable limit for our time. They include the study of the non-condensing engine worked at a pressure of 250 lbs. per square inch above a vacuum, with ratios of expansion ranging from 5 to 20, the back pressure being assumed at 16 lbs., and the temperature of feed-water at 203° Fahr., a comparison being made between the ideal thermodynamic case and that of the real engine, subject to a moderate amount of waste. They also include the case of the condensing engine working under similar initial pressures, and at ratios of expansion ranging from 10 to 50, and with feed-water at 104° Fahr., similarly compared with a case representative of practical working conditions. The back-pressure is assumed to be 5 lbs. above a vacuum.

In making these comparisons, the methods and formulas of Rankine are adopted as most convenient and simple, and as giving approximations which, so far as inaccurate, are probably on the right side. The cases thus computed would in no way differ from those illustrated in the examples given in the works of that author, except as differences of data give difference of numerical results.* An interesting collection of data and results may be be found in Rankine's works, in all cases illustrating the ideal case. In the cases here assumed the jacketed engine is studied, and the conditions of action of the jacket are taken as assumed by Rankine, *i. e.*, that the jacket is so effective as to keep the initially dry steam in the dry and saturated state quite up to the end of the expansion period. It is further assumed that the wastes of heat by conduction and radiation, internally and externally, mainly in the exhaust period, may be represented approximately by an expression of the form deduced by the writer from earlier observation and especially from the Sandy Hook experiments, as a factor, constant for any given engine, multiplied by the square root of the ratio of expansion.† This constant is here taken as .15 for a simple jacketed engine and as .075 for the jacketed compound engine, which figures are sufficiently accurate for the purpose here in view. It is only necessary to assume that the engines here to be considered are of substantially the size and construction of those the trials of which give similar figures for wastes. The Sandy Hook engine may stand for the simple engine, for example. The real case is taken to be that of a compound jacketed engine; for the ideal case there is, of course, no distinction between the simple and compound.

Taking up the first step in the series of problems, we determine the thermodynamic efficiency of the steam, dry and saturated, under the prescribed conditions of the ideal case. In assuming a relation between the weights of steam and of fuel demanded, it is considered that the efficiency of the boiler should be such as to give an evaporation of at least 9 lbs. from 104° Fahr. and of 10 lbs. from 203° Fahr. per pound of coal, which is equivalent to about 10.5 from and at the boiling point under atmospheric pressure. It is assumed that the jacket-water drains completely and automatically back to the boiler, as always should be the case; the neglect of this precaution often making a sensible reduction of the efficiency of the system. The results of these

* It is unnecessary to reproduce the equations. They may be found in Rankine's "Steam-Engine," pages 375–411, and are so well known that it would be simply pedantic to introduce either these expressions or the details of the work here, to encumber our pages.

† Haton de la Goupilliere coincides with Siniglia, who says that this function was first proposed by the present writer, and subsequently confirmed by direct experiment at Sandy Hook and elsewhere.—*Cours des Machines*, Vol. II.

computations are given in tabular form in the annexed general table of figures, data and results being presented so fully that comment is hardly necessary. It will be seen that the efficiencies range from 16.7% to 18.2%, in the case of the non-condensing, and from 16.9% to 22% for the condensing engine; the maximum being found at a ratio of expansion, in the first case of about 10 and in the second of about 30. Beyond these ratios the terminal pressure falls below the back-pressure, and a waste follows instead of gain by further expansion.

These results are still better exhibited by the curves (Figs. 228 and 229) plotted from the numerical values; the ideal case in both sets being represented by dotted lines, and the real engine giving the widely different curves in full lines. The great difference between the condensing and the non-condensing engine, for the ideal case, is well shown, not only as to consumption of fuel at a similar ratio of expansion, but also as affected by changing values of that ratio. The gain by expansion in the former case continues far beyond that at which the latter finds a limit; while the point of maximum effect is far more sharply defined with the non-condensing engine. Variation from the best ratio for the latter causes much more serious loss than with the condensing engine. The numerical values obtained are presumably those which we should obtain if we were to find a way of building engines with working cylinders having non-conducting inner surfaces, as sought to be secured by the various expedients adopted or proposed by Smeaton, Emery, the writer, and others. The points of maximum efficiency and those for minimum consumption of steam and of fuel are coincident in these cases, and also that for minimum supply of feed-water. As will be seen presently, this last is not the case for jacketed engines, in either the ideal or the real case, in consequence of the fact that a part of the working fluid circulates continuously between jacket and boiler and makes no demand upon the source of supply for replenishment. At each cycle it gives up its latent heat of evaporation; that measurable as sensible heat remaining constant throughout its period of action.

COMPOUND STEAM-ENGINE. JACKETED.

NON-CONDENSING.

Ideal Case.																					Real Case.			
r	v_1	$v_2 = r\,v_1$.	p_2	p_3	$v_2\,(p_2-p_3)$.	U_1	U_2	U_1-U_2.	$U' = U_1-U_2+v_2\,(p_2-p_3)$.	H_2	(Feed = 208°) h_4.	$h = U_1-U_2+H_2-h_4$.	H_1-h_4.	Q Heat Supplied by Jacket pr. lbs. Steam. $h-(H_1-h_4)$.	Effic. $= \frac{U'}{h}$.	S Lbs. H_2O worked in Cyl. pr. I. H. P. pr. hr. $\frac{1980000}{U_1}$	m (10) Coal to Evap. S $\frac{S}{10\ \&\ 9}$	n Coal to Supply Heat Supplied Jacket. $\frac{Q\,m}{H_1-h_4}$	Total Fuel. $m+n$.	$1 + 0.075\sqrt{r}$.	Efficiency.	Water per I. H. P. pr. hr.	Coal per I. H. P. pr. hr.
5	1.84	9.2	6494	2304	38548	424280	312633	107647	146195	899854	132360	875141	797251	77890	.1671	13.54	1.354	.1354	1.489	1.168	.1430	15.815	1.739
8	"	14.72	3931	"	23949	"	283189	137091	161040	892916	"	897647	"	100396	.1794	12.29	1.229	.1508	1.389	1.212	.1480	14.895	1.683
10	"	18.40	3102	"	14683	"	269523	150757	165440	889960	"	908557	"	111106	.1821	11.96	1.196	.1674	1.363	1.237	.1472	14.795	1.686
15	"	27.60	1987	"	—8749	"	245228	175032	166303	884604	"	927296	"	130045	.1798	11.91	1.191	.1906	1.382	1.290	.1392	15.864	1.788
20	"	36.80	1483	"	—30213	"	228319	191961	161748	881093	"	940694	"	143443	.1720	12.24	1.224	.2208	1.444	1.335	.1290	16.340	1.928
CONDENSING.																								
5	18.4	9.20	64.94	720	53121	420280	312633	107647	160768	899854	55612	951889	873999	77890	.1689	12.315	(9) 1.368	.122	1.490	1.168	14.46	14.384	1.740
10	"	18.40	3102	"	43829	"	269523	150757	194586	889960	"	985105	"	111106	.1975	10.170	1.180	.142	1.272	1.237	.1597	12.580	1.573
20	"	36.8	1483	"	28078	"	228319	191961	220039	881093	"	1017442	"	143443	.2163	8.995	.999	.172	1.171	1.335	.1620	12.008	1.563
30	"	55.2	961.	"	13303	"	204058	215322	228625	876434	"	1036144	"	162145	.2207	8.660	.962	.1732	1.135	1.411	.1564	12.360	1.601
40	"	73.6	709.9	"	—743	"	189023	231257	230514	873373	"	1049018	"	175019	.2197	8.590	.954	.189	1.143	1.474	.1491	12.662	1.685
50	"	92.0	559.9	"	—14729	"	170665	243625	228896	871067	"	1059080	"	185081	.2161	8.650	.961	.208	1.169	1.520	.1410	13.235	1.789

$p_1 = 36000$ lbs. per sq. ft. $= 250$ lbs. per sq. in.

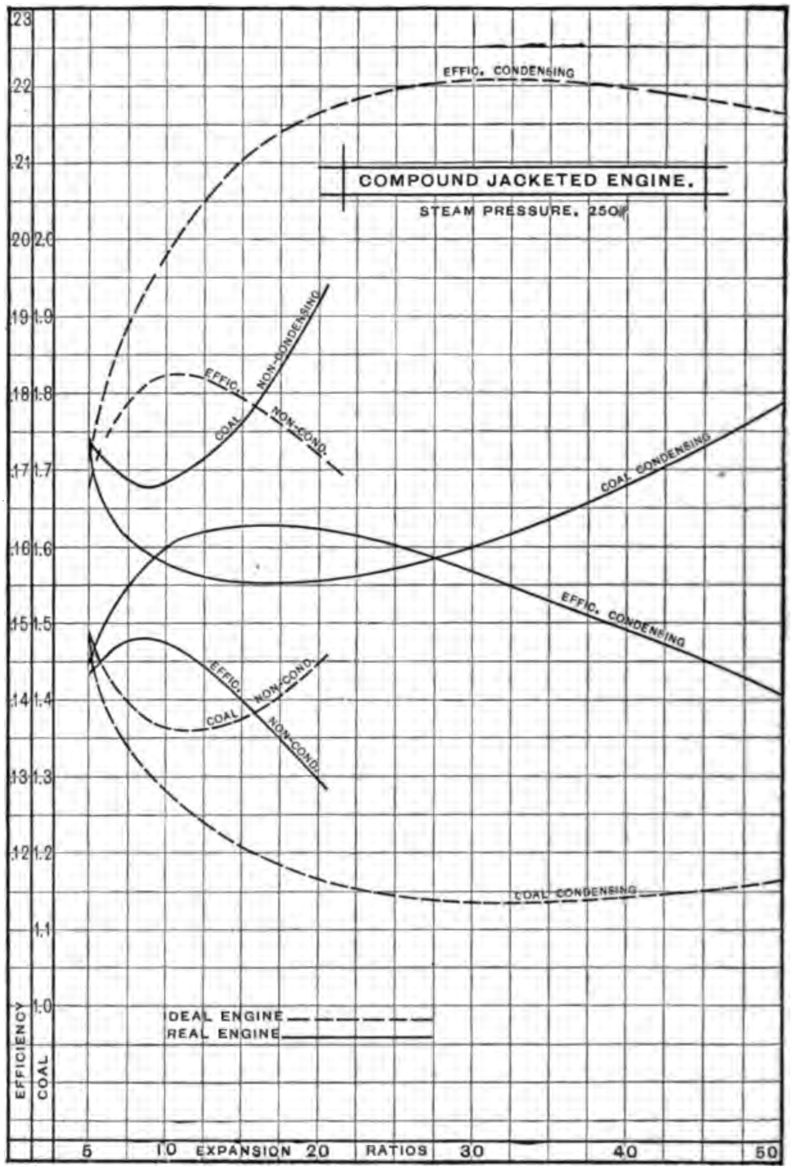


FIG. 228.

Taking up the real engine, as modified by the assumed conditions of operation, and the computed probable wastes, we find that we must add to the heat-supply a portion measured by the assumed heat-wastes, and which we here compute by multiply-

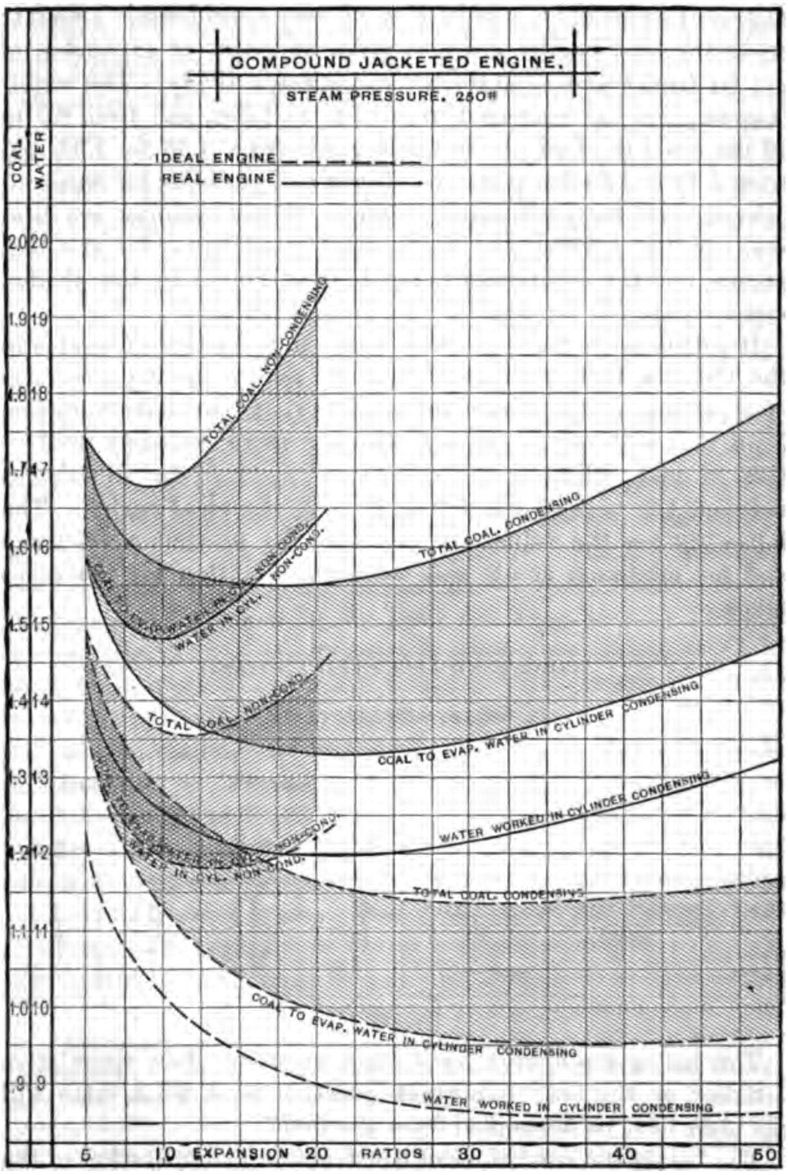


FIG. 229.

ing the amount for the thermodynamic case by the factor, $1 + 0.075 \sqrt{r}$, that for one of the two cylinders of the compound engine under the prescribed conditions. The efficiencies now range from 13% to about 15%, and from 14% to 16%, in the two engines respectively, a reduction of very considerable amount; while the best results are now given at ratios of expansion of not far from 8 and 20 in the two cases, respectively. The water-consumption has increased from 12 to 14.8 lbs., and from 8.5 to 12 lbs.; and the fuel account has risen from 1.36 to 1.68, and from 1.13 to 1.55 lbs. per horse-power and per hour for non-condensing and for condensing engines. These changes are best seen on the curves; the lower sets being those for the real engine, and the differences being best exhibited by the shaded areas separating the pairs on the second plate.

It will be seen that the effect of this introduction of wastes in the ideal as in the real engine, is to reduce greatly the ratio of expansion giving maximum efficiency, and to make variation from that ratio of maximum efficiency more seriously productive of loss; while at the same time making the differences between the several cases less than in the ideal engine. The following are the values of the ratios for maximum efficiency and for minimum steam and water consumption for the cases taken:

COMPOUND JACKETED ENGINE.

$p_1 = 250$; $p_2 = 5$ and 16; r variable.

Case.	Non-Condensing.		Condensing.	
	Ideal.	Real.	Ideal.	Real.
r for Maximum Efficiency	11	8.5	32	17
" Minimum Fuel	11	8.5	32	17
" " Water	13	9.5	38	21
Water-rate	12	14.75	8.5	12
Fuel-rate	1.35	1.68	1.1	1.55

The last given figures for r have no other than speculative interest, as the only important question is at what ratio will the least heat be demanded from the fuel?

The better the conditions of construction and operation of the engine, and the less the difference between the two sets of cases, the nearer will the value of the ratio of maximum efficiency and minimum expenditure of steam and of fuel approach a common value—that for the ideal case. The wastes assumed in the above computation are probably capable of being reduced, by careful drying or moderate superheating of the steam, and by adopting high speed of engine, to one-half these proportions. The figures here computed have been actually approximated by some of our best constructors when using steam pressures one-half those here assumed. It ought to prove practicable, one would think, to bring the water-consumption by the adoption of these high-pressures, and by further dividing the wastes by the use of three or four cylinders in series, and by properly supplying dry steam to the engine, down to within 10% of the best figures here obtained for the ideal case. This is, however, pure speculation, and the main interest attaching to these results comes of their clear illustration of the method by which efficiency varies with variation of the quantities entering these computations.

In computing probable wastes by the real engine, either the methods here adopted as first proposed by the writer, or that lately suggested by Professor Cotterill, or that adopted by Professor Marks, may be used. Either, if the method of use and the constants employed are rightly taken by reference to experiment, will give substantially similar, and probably satisfactory results. The method employed by Mr. Buel and approved by Professor Smith, also, for the usual case, will serve conveniently and fairly well for applying corrections to the thermodynamic problem in the attempt to adapt it to the real case.*

The real measure of the useful power of an engine is not the indicated power, however, but the dynamometric power, as measured by the brake, or at the point at which the engine delivers its energy to the machinery of transmission. A well-built non-condensing engine should have an efficiency of machine at least as high as 92.5%, and engines have been constructed doing better than this. An equally well-built condensing engine should approximate 90% efficiency of machine, though 85 is a much more

* For these quantities and equations, see papers in earlier numbers of the *Transactions of the A. S. M. E.*, the late new edition of Professor Cotterill's admirable treatise on the steam-engine, the papers of Mr. R. H. Buel in the *American Machinist*, and those of experimentalists referred to by these several writers, and especially as summarized by Professor Cotterill. The constants here taken are obtained by assuming the single cylinder to have the general proportions and speed of piston of the Sandy Hook engine, the gain by jacketing to be about as found by Emery and a number of other investigators, including the writer, and the loss in multiple-cylinder engines to be substantially that of one cylinder. Engineers familiar with a different practice may assume a correspondingly different set of constants.

ımon figure. Taking these maximum values for the two ,chines respectively, the last table becomes modified thus:

COMPOUND JACKETED ENGINE.

Data as above.

Case.	Non-Condensing.		Condensing.	
	Ideal.	Real.	Ideal.	Real.
r for Maximum Efficiency	11	8.5	32	17
Water per D. H. P	18	16	8.5	13.5
Fuel " "	1.5	1.8	1.2	1.7

It is apparently doubtful whether a vacuum will be found desirable, with its concomitant costs for the air-pump system, when we come to the utilization of these high pressures, unless we can at the same time find a way to reduce its wastes of power and the tax thus imposed upon the engine.

DISCUSSION.

Mr. Geo. H. Barrus.—I notice in Prof. Thurston's valuable paper that the tables which he gives are based upon estimates of cylinder condensation in which he uses the constant .15 for a simple jacketed engine, in the formula $c = .15 \sqrt{r}$, which is his formula for working out the wastes by cylinder condensation. This constant is the same as the professor used in a previous paper on "Philosophy of the Multi-Cylinder, or Compound Engine," which was read at the New York meeting in November, 1889. In a criticism of the paper which I presented, and which may be found in Vol. XI. of the Transactions, 1890, I drew attention to the results of a large number of tests which I had made on the class of engines to which Prof. Thurston referred, and I pointed out that in an average of these tests, the constant, which is applicable to the real performance of engines, was very different from the one which Prof. Thurston had taken, and I threw some doubt upon the applicability of the Sandy Hook experiments, by which this constant was obtained, as representing ordinary good practice. In view of this criticism, which, in the professor's closing reply, he seems to have accepted as being a tenable one, it is greatly to be regretted that he did not use some other constant in working up the tables of the present paper than the one referred to. I am aware that in a foot-note he states that other engineers can employ any constant which they see fit, and therefore adapt the tables to their own practice; but, in spite of this, it would have been of much greater interest and value if the professor had done this in his paper. More so, perhaps, because the constant which he uses appears to be erroneous, and no reliable conclusions can be drawn from it. I should be glad on my part if Prof. Thurston would review his paper, and give the results of computations based on some different constant more in accord with what seems to be better practice, and incorporate the results with those which he has already given in the paper. I dare say that a different constant would make a considerable difference in the relative showing of the different cases which he works out.

CCCCXLVIII.*

APPLICATION OF HIRN'S ANALYSIS TO MULTIPLE-EXPANSION ENGINES.

BY C. H. PEABODY, BOSTON, MASS.

(Member of the Society.)

THE application to simple engines of Hirn's theory, or, more properly, of Hirn's analysis of the interchange of heat between the steam and the walls of the cylinder, is now familiar to the engineering profession, through the labors of Hirn, Hallauer, Dwelshauvers-Dery, Mair, and others. But the writer has not been able to find a clear and correct statement of the analysis for compound and multiple-expansion engines. Applications of the analysis to compound engines are given by Hallauer in the form of numerical calculations only, and he contents himself with finding the heat rejected from the walls of the cylinder during exhaust. Among the very important tests on large engines reported by Mair † are several on compound engines, but he has united in one term the algebraic sum of the quantity of heat absorbed by the low-pressure cylinder walls during admission, and that yielded during expansion, a quantity which in his tests is sometimes positive and sometimes negative, and which does not appear to have any physical meaning.

The statement of the analysis given here is a development of those given by Dwelshauvers-Dery ‡ and Mair, making use of the latter's method of representing the heat equivalent of the intrinsic energy of all the fluid in the cylinder by one character. An application is given to experiments made on the triple-expansion engine in the engineering laboratories of the Massachusetts Institute of Technology.

* Presented at the Providence Meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. of the *Transactions*.

† *Proc. of the Inst. Civ. Engs.*, Vol. LXX., p. 313; Vol. LXXIX., p. 323.

‡ *Revue Universelle des Mines*, Vol. VIII., p. 362; 1880, also *Proc. Am. Mech. Soc.*

SIMPLE ENGINES.

Suppose that M pounds of steam are admitted to the cylinder of a simple engine per stroke, having in the supply pipe the pressure p pounds per square inch absolute, and the quality x; *i. e.*, each pound is x part steam mingled with $1-x$ part water. The heat brought into the cylinder per stroke, reckoned from freezing point, is

$$Q = M(q + x\,r) \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

in which q is the heat of the liquid and r is the latent heat of vaporization. Should the steam be superheated in the supply pipe to the temperature t_s, then

$$Q = M[q + r + c_p(t_s - t)] \quad . \quad . \quad . \quad . \quad (2)$$

in which t is the temperature of saturated steam at the pressure p, and $c_p = 0.4805$ is the specific heat of superheated steam at constant pressure.

Let the heat equivalent of the intrinsic energy of all the water and steam in the cylinder at any instant be represented by I. Let M_0 be the weight of steam caught in the cylinder at compression. Then we shall have

at admission $\quad I_0 = M(q_0 + x_0\,\rho_0) \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$

at cut-off $\quad I_1 = (M + M_0)(q_1 + x_1\,\rho_1). \quad . \quad . \quad . \quad (4)$

at release $\quad I_2 = (M + M_0)(q_2 + x_2\,\rho_2). \quad . \quad . \quad . \quad (5)$

at compression $\quad I_3 = M_0(q_3 + x_3\,\rho_3). \quad . \quad . \quad . \quad . \quad . \quad (6)$

in which, q_0, q_1, q_2, q_3, and ρ_0, ρ_1, ρ_2, and ρ_3 are respectively the heats of the liquid, and the internal latent heats, or heat equivalents of the internal work, at the absolute pressures p_0, p_1, p_2, and p_3 measured on the indicator diagram, at admission, cut-off, release, and compression.

If the work under any line of the indicator diagram, reckoned from a perfect vacuum, is represented by W, and if A is the reciprocal of the mechanical equivalent of heat (equal to $\frac{1}{778}$), then the heat equivalents of the external work done by or on the piston are

during admission	$A\ W$
during expansion	$A\ W_b$
during exhaust	$A\ W_c$
during expansion	$A\ W_d$

Usually heat is absorbed by the walls of the cylinder during admission of steam and up to the point of cut-off, heat is yielded during expansion and during exhaust, while the interchange during compression is uncertain. It is possible that the walls may receive heat during expansion, and it is conceivable that they should give heat to the steam during the entire cycle; consequently it is convenient to affect all of the interchanges of heat by the same sign and to assume that heat absorbed by the wall is positive; in the solution of examples a negative sign will then mean that heat is yielded by the walls.

The interchanges of heat may be expressed as follows:

during admission $\quad Q_a = Q + I_0 - I_1 - A\ W_a \quad . \quad . \quad . \quad . \quad (7)$

during expansion $\quad Q_b = I_1 - I_2 - A\ W_b \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$

during exhaust $\quad Q_c = I_2 - I_3 - M_{q4} - G\,(q_k - q_l) + A\ W_c \ (9)$

during compression $\quad Q_d = I_3 - I_0 + A\ W_d \quad . \quad . \quad . \quad . \quad . \quad . \ (10)$

In equation (9) the expression Mq_4 represents the heat carried away by the condensed steam flowing away from a surface condenser at the temperature t_4, and $G\ (q_k - q_0)$ represents the heat carried away by G pounds of cooling water, which enters with the temperature t_l and leaves with the temperature t_k.

Let the volume of 1 lb. of water in cubic feet be σ, and let the increase of volume due to vaporization be u, so that the volume of 1 lb. of dry steam is $u + \sigma$, while the volume of 1 lb. of moist steam is $xu + \sigma$. Let the volume of the clearance of the engine be V_0 cubic feet, and let the volume developed by the piston at cut-off and release be V_1 and V_2, while the volume remaining at compression is $V_3 + V_0$. The volumes of the mixed fluid in the cylinder are:

At admission $\quad V_0 = M_0\,(x_0\,u_0 + \sigma) \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$

At cut-off $\quad V_0 + V_1 = (M + M_0)\,(x_1\,u_1 + \sigma) \quad . \quad . \quad (12)$

At release $\quad V_0 + V_2 = (M + M_0)\,(x_2\,u_2 + \sigma) \quad . \quad (13)$

At compression $V_0 + V_3 = M_0\ (x_3\,u_3 + \sigma) \quad . \quad . \quad . \quad . \quad (14)$

An examination of the three groups of equations (3) to (6), (7) to (10), and (11) to (14), will show that all that is requisite for an entire solution of our problem is the knowledge of the quality of the steam at one of the four events of the cycle. It is customary to assume the steam at compression to be dry and saturated, or that x_3 is unity. Such an assumption gives for the weight of steam at compression,

$$M_0 = \frac{V_0 + V_3}{u_3 + \sigma} = (V_0 + V_3)\,\gamma_3 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

in which γ_3 is the density or weight of one cubic foot of saturated steam at the absolute pressure at the beginning of compression.

Applying this result to equations (11) to (13), gives

$$x_0 = \frac{V_0}{M_0 u_0} - \frac{\sigma}{u_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

$$x_1 = \frac{V_0 + V_1}{(M + M_0)\,u^1} - \frac{\sigma}{u} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

$$x_2 = \frac{V_0 + V_2}{(M + M_0)\,u_2} - \frac{\sigma}{u_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

Having the quality of the steam at admission, cut-off, and release, the several values of I at these points and at compression may be found from equations (3) to (6), and finally the interchanges of heat may be calculated by aid of equations (7) to (10). It may happen that the assumption of dry steam at compression may make the steam in the cylinder at admission appear to be superheated, *i. e.*, x_0 will appear to be larger than unity; in such case it will be convenient to assume the steam to be saturated at admission also. The superheating is commonly small, and the error from the assumption of dry steam at compression and at admission is believed to be insignificant. In some of the tests to be quoted later, the steam in the intermediate and low-pressure cylinders is superheated at release by the action of jackets supplied with steam at the boiler pressure. In such case the intrinsic energy of one pound of superheated steam may be calculated by aid of the equation.*

$$E = 3\,pv + 778 \times 857.2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

* *Thermo-dynamics of the Steam Engine*, p. 128, by the author.

in which v is the specific volume in cubic feet, and p is the pressure in pounds on the cubic foot. Which will give at release

$$I_2 = (V_0 + V_2)\frac{3 \times 144 p_2}{778} + 857.2 \quad . \quad . \quad . \quad . \quad (20)$$

in which p_2 is the absolute pressure per square inch at release.

Let the heat lost by radiation, condensation, etc., be Q_e, and let Q_i be the heat supplied by the steam jacket when one is used. If the jacket is supplied by a separate pipe, or if in any way the water condensed therein can be withdrawn and measured, the value of Q_i is readily determined for any test; Q_e may be determined by collecting the condensation in the jackets while the engine is at rest.

Of the heat supplied to the cylinder per stroke, a part is changed into work, a part is carried away by the condensed steam and the cooling water, and the remainder is lost by radiation; therefore

$$Q_e = Q + Q_i - Mq_4 - G(q_k - q_i) - AW \quad . \quad . \quad (21)$$

in which W is the work represented by the area of the indicator diagram or

$$W = W_a + W_b + W_c + W_d \quad . \quad . \quad . \quad . \quad (22)$$

Equation (21) may be used to calculate Q_e when it cannot be determined directly, but such a calculation is vitiated by all the errors of determining the several quantities in that equation. When Q_e is determined directly, equation (21) may be considered a check on the accuracy of the determination of all the quantities appearing in it; it is not a check on the determination of the interchanges by equations (7) to (10).

If a test is made on a non-condensing engine, or if for any reason the terms depending on the condenser cannot be determined, then these terms may be eliminated from equation (9) by aid of equation (21), giving

$$Q_c = I_2 - I_3 - Q - Q_i + Q_e + A(W + W_c) \quad . \quad (23)$$

It may be of interest to note that this equation is equivalentto Hirn's direct determination of the exhaust waste, and that equation (9) is equivalent to his second determination or check. It is hardly necessary to call attention to the fact that such a check appears much more striking in an arithmetical calculation than in the algebraic statement of the analysis. Exception must be taken to the idea that the exhaust waste can be considered as a test of the efficiency of an engine While it is true that a large exhaust waste causes a poor economy, the extinction of the exhaust waste, by lavish use of steam jackets or otherwise, may be dearly bought. This is well shown by Mair's tests.

COMPOUND ENGINES.

The equations (7), (8), and (10), with equation (23), may be applied directly to the high-pressure cylinder of a compound engine, giving

$$Q_a = Q + I_o - I_1 - A\,W_a \quad . \; . \; . \; . \; . \; . \; . \quad (24)$$

$$Q_b = I_1 - I_2 - A\,W_b \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (25)$$

$$Q_c = I_2 - I_3 - Q - Q_i + Q_e + A\,(W + W_c) . \quad (26)$$

$$Q_d = I_3 - I_0 + A\,W_d \quad . \; . \; . \; . \; . \; . \; . \; . \quad (27)$$

By aid of equation (21) the term Q may be eliminated from the equation for the interchange of heat during admission; the other equations, (8) to (10), can be used unchanged in form. The interchanges of heat in the low-pressure cylinder can then be found by the equations

$$Q'_a = Q'_e - Q'_i + Mq_1 + G\,(q_k - q_i) + I_0 - I_1 + A\,(W' - W'_a)\ (28)$$

$$Q'_b = I'_1 - I'_2 - A\,W'_b \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (29)$$

$$Q'_c = I'_2 - I'_3 - Mq - G\,(q_k - q_i) + A\,W'_c \quad . \; . \; . \; . \; . \; . \; . \quad (30)$$

$$Q'_d = I'_3 - I'_0 + A\,W'_d \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (31)$$

These equations may be applied to any compound engine, whether it has a receiver or not, and without taking account of the loss or gain of heat experienced by the steam passing from the high to the low-pressure cylinder. Thus it may be applied when there is a reheater between the cylinders, even though the heat thus imparted is not determined. It is apparent that this method cannot be extended to a triple-expansion engine.

MULTIPLE-EXPANSION ENGINES.

For multiple-expansion engines the following method may be used. The heat rejected by the high-pressure cylinder during exhaust is

$$Q + Q_i - A\,W - Q_e \quad . \; . \; . \; . \; . \; . \; . \; . \quad (32)$$

This heat passes into the first intermediate cylinder, and from there, with the gain or loss experienced, proceeds to the next cylinder. The sum, or difference, may be taken for Q'_1 the heat brought into the next cylinder per stroke. The same operation may be applied to each successive cylinder, and the final result may be checked by aid of the data depending on the condenser.

Compound and multiple-expansion engines, having various arrangements of steam-jackets and reheaters, will require special modifications of the form of the analysis. If two or more steam-jackets have a common drain or if steam is passed in succession through several jackets, then the distribution of the heat among the several jackets must be determined or estimated.

The equations for the interchanges of heat in the triple engine of the Massachusetts Institute of Technology are as follows, the terms Q_{eR} and Q_{jR} representing the heat radiated from the surface of an intermediate receiver and the heat supplied by condensation of steam in the jacket of such a receiver:

High-pressure cylinder,

$$Q = M(xv + q) \quad \ldots \quad (33)$$

$$Q_a = Q + I_0 - I_1 - AW_a \quad \ldots \quad (34)$$

$$Q_b = I_1 - I_2 - AW_b \quad \ldots \quad (35)$$

$$Q_c = I_2 - I_3 - Q - Q_j + Q_e + A(W + W_c) \quad \ldots \quad (36)$$

$$Q_d = I_3 - I_0 + AW_d \quad \ldots \quad (37)$$

Intermediate cylinder,

$$Q' = Q + Q_j - AW - Q_e + Q_{jR} - Q_{eR} \quad \ldots \quad (38)$$

$$Q_a = Q' + I'_0 - I'_1 - AW'_a \quad \ldots \quad (39)$$

$$Q_b = I'_1 - I'_2 - AW'_b \quad \ldots \quad (40)$$

$$Q_c = I'_2 - I'_3 - Q' - Q'_j + Q'_e + A(W' + W'_c) \quad \ldots \quad (41)$$

$$Q_d = I'_3 - I'_0 + AW'_d \quad \ldots \quad (42)$$

Low-pressure cylinder,

$$Q'' = Q' + Q'_j - AW' - Q'_e + Q'_{jR} - Q'_{eR} \quad \ldots \quad (43)$$

$$Q''_a = Q'' + I''_0 - I''_1 - AW''_a \quad \ldots \quad (44)$$

$$Q''_b = I''_1 - I''_2 - AW''_b \quad \ldots \quad (45)$$

$$Q''_c = I''_2 - I''_3 - Q'' - Q''_j + Q''_e + A(W'' + W''_c) \quad (46)$$

$$\text{or } Q''_c = I''_2 - I''_3 - M_{q4} - G\,(q_k - q_i) + A\,W''_d \quad . \quad . \quad . \quad (47)$$

$$Q''_d = I''_b - I''_0 + A\,W''_d \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (48)$$

Finally,

$$\Sigma Q_e = Q_e + Q_{eR} + Q'_e + Q'_{eR} + Q''_e = Q + Q_j + Q_{jR} + Q'_j + Q'_{jR} + Q''_j - M_{q4} - G\,(q_k - q_i) - A\,(W + W' + W'') \ (49)$$

APPLICATION OF ANALYSIS TO INSTITUTE OF TECHNOLOGY ENGINE.

The experimental engine in the engineering laboratories of the Massachusetts Institute of Technology is a three-crank horizontal triple-expansion engine with two intermediate receivers, built by E. P. Allis & Co., of Milwaukee. The receivers and the cylinders are thoroughly steam-jacketed by separate pipes from the main steam-pipe, and drain individually into five receptacles for gathering and measuring the condensation. The jackets on the heads and barrels of the cylinders can be applied separately or together. Reducing valves are arranged to supply the jackets of the second receiver and the intermediate and low-pressure cylinder with steam at less than boiler pressure. The cranks are usually set 120° apart, the high-pressure crank leading, but the angle between the cranks can be varied. Steam is supplied to the engine through a 6-inch pipe carried under ground about 1,000 feet from the Rogers Building. Condensation is withdrawn from the pipe when it enters the building, and for further precaution the steam is passed through a Stratton separator on the way to the engine. The quality of the steam is determined near the throttle-valve by a throttling calorimeter. The exhaust from the engine is led to a surface condenser, beneath which is a Blake direct-acting air pump which discharges the condensed steam in weighing tanks. The automatic governor is arranged to control the cut-off of any or all of the cylinders; or, when the automatic gear is disconnected, the cut-off may be varied by hand. During all of the tests the governor controlled the cut-off of the high-pressure cylinder, and the cut-off on the other cylinders was adjusted to give expansion down to the back pressure line in the high-pressure and intermediate cylinders. The low-pressure cylinder has two wrist plates, so that the cut-off may be made longer than half stroke. The piping of the engine is arranged so that it may be run in several combinations; before making a test or series of tests all valves are arranged and tested to guard against leakage.

The main dimensions of the engine are as follows:

Diameter of the high-pressure cylinder	9	inches.
Diameter of the intermediate cylinder	16	"
Diameter of the low-pressure cylinder	24	"
Diameter of the piston-rods	$2\frac{3}{16}$	"
Stroke	30	"

Clearance in per cent. of the piston displacements:

High-pressure cylinder,	head end,	8.88;	crank end,	9.76
Intermediate	"	10.4	"	10.9
Low-pressure	"	11.25	"	8.84

The following table gives the data and results of form tests made for the application of Hirn's analysis, by the class graduating this year:

	I.	II.	III.	IV.
Duration of test, minutes	60	60	60	60
Total number of revolutions	5299	5228	5178	5148
Revolutions per minute	88.3	87.1	86.2	85.8
Steam consumption during test, pounds:				
Passing through cylinders	1193	1157	1234	1305
Condensation in h. p. jacket	57	50	29	30
" in 1st receiver jacket	61	64	69	72
" in inter. jacket	85	92	97	105
" in 2d receiver jacket	53	50	52	51
" in l. p. jacket	89	76	90	87
Total	1538	1489	1571	1650
Condensing water for test, pounds	22847	22186	20244	20252
Priming, by calorimeter	0.013	0.012	0.011	0.012
Temperatures, Fahrenheit:				
Condensed steam	95.4	92.1	102.4	105.3
Condensing water, cold	41.9	42.1	43.0	42.8
Condensing water, hot	96.1	96.6	106.3	109.6
Pressure of the atmosphere, by the barometer, pounds per square inch	14.8	14.8	14.7	14.7
Boiler pressure, pounds per square inch, absolute	155.3	155.5	156.9	157.7
Vacuum in condenser, inches of mercury	25.0	25 1	24.1	23.9
Events of the stroke:				
High-pressure cylinder—				
Cut-off, crank-end	0.192	0.194	0.245	0.283
head-end	0.215	0.205	0.271	0.305
Release, both ends	1.00	1.00	1.00	1.00
Compression, crank-end	0.05	0.05	0.04	0.04
head-end	0.05	0.05	0.05	0.05
Intermediate cylinder—				
Cut-off, both ends	0.29	0.29	0.29	0.29
Release, both ends	1.00	1.00	1.00	1.00
Compression, crank-end	0.03	0.03	0.03	0.03
head-end	0.04	0.04	0.04	0.04
Low-pressure cylinder—				
Cut-off, crank-end	0.38	0.38	0.38	0.38
head-end	0.39	0.39	0.39	0.39
Release, both ends	1.00	1.00	1.00	1.00

	I.	II.	III.	IV.
Absolute pressures in the cylinder, pounds per square inch:				
High-pressure cylinder—				
Cut-off, crank-end	145.9	145.9	138.8	138.8
head-end	143.2	143.1	140.8	140.6
Release, crank-end	41.3	41.5	44.7	48.4
head-end	41.5	40.5	45.7	49.8
Compression, crank-end	43.7	45.8	48.5	58.2
head-end	48.7	47.9	54.5	62.0
Admission, crank-end	64.5	68.8	72.2	81.2
head-end	75.8	74.8	86.7	97.8
Intermediate cylinder—				
Cut-off, crank-end	37.2	37.6	38.6	40.9
head-end	35.0	35.3	39.6	42.6
Release, crank-end	13.6	14.2	14.7	16.0
head-end	13.4	13.8	14.9	16.0
Compression, crank-end	16.8	17.8	18.2	19.0
head-end	17.9	18.8	20.8	22.4
Admission, crank-end	20.4	20.8	22.2	23.1
head-end	21.1	22.8	24.2	26.7
Low-pressure cylinder—				
Cut-off, crank-end	12.1	12.6	12.4	13.2
head-end	12.0	12.4	13.1	14.0
Release, crank-end	5.6	5.8	5.1	5.7
head-end	5.4	5.8	5.9	6.4
Compression and Admission, crank-end	3.7	3.8	4.1	4.2
head-end	4.3	4.5	4.6	4.7
Heat equivalents of external work, B. T. U., from areas on indicator diagram to line of absolute vacuum:				
High-pressure cylinder—				
During admission, $A\ W_a$, crank-end	5.71	5.78	7.00	8.19
head-end	6.61	6.37	8.42	9.50
During expansion, $A\ W_b$, crank-end	10.65	10.76	10.40	10.25
head-end	10.81	11.04	11.22	11.09
During exhaust, $A\ W_c$, crank-end	7.78	7.89	8.44	9.02
head-end	8.08	8.15	9.04	9.66
During compres'n, $A\ W_d$, crank-end	0.48	0.60	0.49	0.73
head-end	0.62	0.64	0.73	0.81
Intermediate cylinder—				
During admission, $A\ W_a$, crank-end	7.58	7.57	7.98	8.64
head-end	7.48	7.55	8.46	9 10
During expansion, $A\ W_b$, crank-end	9.54	9.54	9.91	10.64
head-end	9.22	9.81	10.37	11.14
During exhaust, $A\ W$, crank-end	9.27	9.47	9.64	10.54
head-end	9.27	9.47	10.18	10.84
During compres'n, $A\ W_d$, crank-end	0.39	0.48	0.57	0.46
head-end	0.60	0.70	0.78	0.84
Low-pressure cylinder—				
During admission, $A\ W_a$, crank-end	7.75	7.95	8.38	8.97
head-end	7.99	8.10	8.66	9.39
During expansion, $A\ W_b$, crank-end	6.83	7.10	6.86	7.45
head-end	6.87	7.12	7.34	7.87
During exhaust, $A\ W_c$, crank-end	5.08	5.08	4.62	5.09
head-end	5.08	5.16	4.81	5.00
During compres'n, $A\ W_d$, crank-end	0.00	0.00	0.00	0.00
head-end	0.00	0.00	0.00	0.00

	I.	II.	III.	IV.
Quality of the steam in the cylinder: At admission and at compression the steam was assumed to be dry and saturated:				
High-pressure cylinder—				
At cut-off.....................x_1..	0.785	0.784	0.848	0.875
At release.....................x_2..	0.899	0.903	0.920	0.931
Intermediate cylinder—				
At cut-off.....................x_1..	0.899	0.912	0.906	0.908
At release.....................x_2..	0.994	super-heated.	super-heated.	super-heated.
Low-pressure cylinder—				
At cut-off.....................x_1..	0.978	super-heated.	0.970	0.974
At release.....................x_2..	super-heated.	super-heated.	0.970	0.974
Interchanges of heat between the steam and the walls of the cylinders, in B. T. U. Quantities affected by the positive sign are absorbed by the cylinder walls; quantities affected by the negative sign are yielded by the walls.				
High-pressure cylinder—				
Brought in by steam.........Q ..	132.92	130.77	141.11	149.84
During admission............Q_a ..	23.54	23.48	17.49	14.93
During expansion............Q_b ..	−18.69	−19.28	−15.33	−14.03
During exhaust..............Q_c ..	−8.36	−7.22	−3.50	−2.38
During compression..........Q_d ..	0.45	0.51	0.49	0.52
Supplied by jacket..........Q_j ..	4.56	4.08	2.39	2.50
Lost by radiation...........Q_e ..	1.50	1.52	1.54	1.54
First intermediate receiver—				
Supplied by jacket..........Q_{jR}..	4.92	5.20	5.67	5.95
Lost by radiation...........Q_{eR}..	0.58	0.58	0.59	0.59
Intermediate cylinder—				
Brought in by steam.........Q' ..	131.89	129.61	137.87	146.64
During admission............Q_a' ..	13.62	11.74	11.83	11.75
During expansion............Q_b' ..	−18.65	−18.84	−20.30	−21.88
During exhaust..............Q_c' ..	0.22	1.57	2.88	3.41
During compression..........Q_d ..	0.44	0.51	0.62	0.59
Supplied by jacket..........Q_j' ..	6.82	7.50	7.97	8.64
Lost by radiation...........Q_e' ..	2.45	2.48	2.50	2.51
Second intermediate receiver—				
Supplied by jacket..........Q_{jR}..	4.20	4.04	4.27	4.22
Lost by radiation...........Q_{eR}..	1.20	1.22	1.28	1.24
Low-pressure cylinder—				
Brought in by steam.........Q'' ..	132.14	130.50	138.61	147.33
During admission............Q_a''..	5.85	3.05	5.57	5.29
During expansion............Q_b''..	−9.51	−7.09	−8.65	−10.13
During exhaust..............Q_c''..	2.53	2.23	−1.44	−0.11
During compression..........Q_d''..	0.00	0.00	0.00	0.00
Supplied by jacket..........Q''..	7.08	6.20	7.41	7.14
Lost by radiation...........Q''..	4.34	4.40	4.45	4.47
Total loss by radiation:				
By preliminary test.........ΣQ_e ..	10.07	10.20	10.31	10.35
By equation (49)..................	11.68	10.19	8.75	8.07

	I.	II.	III.	IV.
Power and economy:				
Heat equivalents of works per stroke:				
High-pressure cylinder.... AW ..	8.44	8.34	9.17	9.52
Intermediate cylinder ..... AW' ..	7.12	6.95	7.77	8.42
Low-pressure cylinder..... AW''..	9.64	10.06	10.87	11.79
Totals	25.20	25.35	27.81	29.73
Total heat furnished by jackets......	27.58	27.02	27.71	28.45
Distribution of work:				
High-pressure cylinder............	1.00	1.00	1.00	1.00
Intermediate cylinder.............	0.84	0.83	0.85	0.88
Low-pressure cylinder............	1.14	1.21	1.19	1.24
Horse-power	104.9	104.2	113.1	120.3
Steam per horse-power per hour.....	14.65	14.31	13.90	13.73
B. T. U. per horse-power per minute.	258.3	252.8	244.6	241.1

An investigation of the results in the table shows that the difference in economy depends on the cut-off of the high-pressure cylinder, which is at about $\frac{1}{5}$ stroke for the first two, and at about $\frac{1}{4}$ stroke for the last two. The quantities that are most notably affected by the change of cut-off are the heat furnished by the high-pressure jacket and the condition of the steam, and interchanges of heat in the high-pressure cylinder. Thus the heat supplied by the high-pressure jacket for the tests, with the cut-off at $\frac{1}{4}$ stroke, is about half that furnished in the tests at $\frac{1}{5}$ stroke; which finds explanation or confirmation in the facts, that the steam is much dryer at cut-off with the longer cut-off, and that both the heat absorbed and yielded by the walls of the cylinder is notably less with that cut-off. The general accuracy and correctness of the tests is shown by the comparison of the total radiation, as found by collecting and weighing the jacket condensation when the engine was at rest, and by calculating the same by aid of the terms depending on the condenser. As has already been pointed out, this does not serve as a check on the interchanges of heat, but the regularity of the results for the intermediate and low-pressure cylinders must be considered to be very satisfactory, especially when it is remembered that any error made in work on the high-pressure and intermediate cylinder is carried on through all the remainder of the test. It may not be out of place to call attention to the fact that the four tests were made by four divisions of the class, and to say that carelessness or inaccuracy of a single observer might readily derange all the results of the test.

It is notable that the steam becomes dryer in its course through the engine, under the influence of the thorough steam-jacketing, with steam at boiler pressure, and that it is practically dry at release, in both the intermediate and low-pressure cylinders. In some of the tests it appears to be superheated at this point, and in one, even at cut-off in the low-pressure cylinder. The superheating is in no case large, and as it is accompanied by a small positive value for Q_c, the exhaust-waste, it may be that the steam was really dry and saturated at those points, though superheating by jackets filled with steam at so high a pressure is not impossible. Since a positive value for Q_c indicates that the walls absorbed heat during exhaust, such a result, if large, would be absurd; but in these tests it may properly be looked upon as the error, being in no case larger than about two per cent. of the value of Q; if the steam at release is either dry or superheated, the value of Q_c must be very small or zero.

It is most remarkable that the heat furnished by the jackets is nearly as large in amount as that changed into work in the tests with the cut-off at $\frac{1}{4}$ stroke, and is actually larger for the tests with the cut-off at $\frac{1}{8}$ stroke. Also, that the heat lost by radiation is one-third as much as the heat changed into work, if not more; the engine being arranged for convenient experimental work, probably has an unusually large radiatión, though it is thoroughly wrapped and lagged on all hot surfaces.

DISCUSSION.

Prof. R. H. Thurston.—I think it will be found, on investigation, that Prof. Peabody has here given the most complete analysis ever yet made of the operation of the steam engine, considered as a thermodynamic machine. We were already indebted to him for his presentation, for the first time, in English, with satisfactory completeness, of the principles of the analysis of Hirn and Dwelshauvers-Dery. His book and this bit of work will stand as landmarks in the history of steam-engineering on this side the Atlantic. He deserves our heartiest thanks. I am glad to see that, in turn, he does not forget our indebtedness to M. Dwelshauvers, who developed the ideas of our old friend Hirn in algebraic language.

The paper is too long and too important to permit an intelligent criticism until time can be taken for careful and deliberate study. But we can at once see that its author, taking advantage of the opportunities now so abundantly offered in the large schools of engineering, to secure the help of intelligent, well instructed and reliable computers, has given us a set of figures which will throw more light on the internal thermodynamic operations of the modern engine than could be obtained from all the rest of our literature on the subject taken together. In this he is fortunate in being able to avail himself of all the advantages of a well-designed and beautifully made "experimental engine," on which he may ring all the changes desirable in the pursuit of his researches. I am especially pleased to get these results at this time; as I have felt some anxiety as to the real economic value of the engine, of which a duplicate, as to general dimensions, and in many details, is under construction, under my order and contract, for Sibley College. I hope we shall be able to pull a little ahead of these figures, but am not at all confident; the less so since I have seen these details of heat-transfer and transformation. To do better than this, means a great deal.

I note that the best work is done at the lower grade of expansion; and this fact makes it very desirable to carry out an investigation determining the ratios of expansion giving maxima for the several efficiencies. The large loss by radiation is a surprise to me, and indicates the desirability in this class of engine of providing with exceptional care for its reduction by thoroughly clothing the cylinders—and I should say steam-pipes as well.

The heat supplied by the jackets is a very large proportion of that passing through the engine, and this would seem to show the desirability of even here securing moderate superheating, in order that the re-evaporation may become complete still earlier, and thus insure that the jackets will not give up too much heat during the expansion period, and especially during its later stages and during exhaust. Here I think, Dwelshauvers to the contrary, notwithstanding, that some superheating would give advantages which would be superposed on those of the jacketing. I should judge, however, that the ratio of expansion here adopted and the power developed are, respectively, too large and too small for the best results, for this engine, even thermodynamically, and as a matter of finance considerably removed from those which would be adopted by the designer, if familiar with the principles involved. We shall all be interested in later reports, should Prof. Peabody be able to present them, showing the working of the engine at twice this power and at intermediate powers. The machine will, I have no doubt, be made to give us much and most valuable information on all the new and obscure points of steam-engine thermodynamic operations. We have come to a point in this field of research at which only such *stethoscopic* methods—if that term is allowable—can give us much new light.

Prof. Peabody.—In answer to Prof. Thurston's very complimentary comments of this paper, I will say that some tests made with greater power developed show a somewhat better economy, and that we hope to do better still; he may well expect to "pull a little ahead" of our results given here.

No one who is familiar with the work of Hirn and his school can fail to appreciate its importance and value. We should not, however, lose sight of the fact that the most important question concerning a steam-engine is its commercial economy, which can be stated in simplest terms, *i. e.*, the cost in pounds of steam per horse-power per hour, or better, in thermal units per horse-power per minute. *That* is finally the test of the performance and value of a steam-engine; and while Hirn's masterly analysis of the transformations of steam and heat in passing through the engine, gives us light, where before much was blind groping, it has not, and perhaps cannot, give us a direct solution of the best design for a given combination.

CCCCXLIX.*

THE PREMIUM PLAN OF PAYING FOR LABOR.

BY F. A. HALSEY, SHERBROOKE, P. Q., CANADA.
(Member of the Society.)

THIS plan has been devised in order to overcome the objections inherent in the other plans in general use. It accomplishes this purpose without introducing corresponding objections of its own. Its merits are best shown by contrasting it with the other plans in common use, and it will be discussed with them in the following order:

I. The day's-work plan.
II. The piece-work plan.
III. The profit-sharing plan.
IV. The premium plan.

I. THE DAY'S-WORK PLAN.

Under this method the workman is paid for and in proportion to the time spent upon his work. The objections to the plan are well known. Analyzed to their final cause, they spring from the fact that any increase of effort by the workman redounds solely to the benefit of the employer, the workman having no share in the consequent increase of production. He has consequently no inducement to exert himself and does not exert himself. Under this system, especially in a manufacturing business, matters naturally settle down to an easy-going pace, in which the workmen have little interest in their work, and the employer pays extravagantly for his product.

II. THE PIECE-WORK PLAN.

Under this plan the workman is paid for and in proportion to the amount of work done. It is a natural attempt to overcome the objections to the day's-work plan. It has the appearance of being

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. of the *Transactions*.

just and of being based upon correct principles. Nevertheless, extended inquiry has convinced the writer that it seldom works smoothly, and never produces the results which it should.

An employer who has become dissatisfied with the results of the day's-work plan, and decides to adopt piece work, usually reasons that work which is costing in wages, say one dollar per piece, could, with some extra effort, be produced on the existing scale of wages for about eighty cents; and desiring to give the workman some inducement offers him ninety cents per piece, thereby dividing the expected saving with him. The trouble begins at once. The workman does not believe that he can "make wages" at the rate offered, and objects. He is, however, finally induced or compelled to try it, and immediately proceeds to astonish himself and all others by increasing his output far beyond the expected 25%. His earnings increase with startling rapidity, *but the cost of the work remains where set, at ninety cents per piece*, and the employer soon finds that instead of a substantially equal division of the savings he is getting but little, and the workman practically all of it. He accordingly proceeds to cut the piece price, and the fatal defect of the system appears. This cut is in appearance and in fact an announcement to the workman that his earnings will not be allowed to exceed a certain amount, and that should he push them above that amount he will be met with another cut. Cutting the piece price is simply killing the goose that lays the golden egg. Nevertheless, the goose must be killed. Without it the employer will continue to pay extravagantly for his work; with it he will stifle the rising ambition of his men. The difficulties of the day's-work and piece-work plans are thus seen to be the exact antitheses of one another. Analyzed to their final cause, the difficulties with the piece-work plan spring from the fact that the piece price once set, any increase of effort by the workman redounds to *his own* benefit alone—the employer having no share in the consequent saving of time. To obtain a share he cuts the piece price, with the consequences stated. Under this system matters gradually settle down as before to an easy-going pace in which the workmen approach the limit of wages as nearly as they consider prudent. Their earnings are somewhat more and the cost of the work is somewhat less than under the day's-work plan, but there is no more spirit of progress than under the older method. The employer is constantly on the lookout for a chance to cut the piece prices, that being his only method of reducing cost; and the men are constantly on the lookout to defeat the employer's well understood plan, knowing, as they do, that any one who is so unwise or so unfortunate as to do an increased amount of work will be in effect punished for it by having his piece price cut and himself thereby compelled to work harder in the future for the old amount of income. The system makes the interests of the employer and employee antagonistic, and hence of concerted effort toward a progressive reduction of cost there is none. This I believe to be the usual and natural history of the piece-work plan. I know it to represent the situation in some of the foremost machine shops of the country. An additional objection to the plan grows out of the fact that it requires a knowledge and record of the cost of each piece of a complicated machine, and oftentimes of each operation on each piece. This limits its range of application to products which are produced in considerable quantities.

III. THE PROFIT-SHARING PLAN.

This plan was originally devised in the effort to avoid the objections to the two former plans. Under it, in addition to regular wages, the employees are offered a certain percentage of the final profits of the business. It thus divides the savings due to increased production between employer and employee, and at first sight appears to meet the difficulties of the plans thus far discussed; but, nevertheless, on analysis, will be found to be as defective as they, both in principle and application. The leading objections to the plan are the following:

First. The workmen are given a share in what they do not earn. Increased profits may arise from more systematic shop management, decreased expenses of the sales department, or many other causes with which the workmen have nothing to do. Anything given them from such sources becomes simply a gift, the result of which is wholly pernicious—in fact the entire system savors of patronage and paternalism.

Second. The workmen share, regardless of individual deserts. An active, energetic workman cannot have the same incentive to increased exertion under a system which divides the results of his efforts among a dozen lazy fellows at his side that he would have under one in which his earnings depend on himself alone; on the other hand, a lazy workman would naturally consider it much easier to take his portion of the earnings of his fellows than to exert himself and then divide the results with all the others of the force.

Third. The promised rewards are remote. The incentive cannot be as great under a system which computes and divides the savings once or twice a year as under one which pays out the extra earnings week by week.

Fourth. The plan makes no provision for bad years. We hear much of profit sharing, but nothing of loss sharing. And yet the workman cannot expect to share the profits while others assume the losses; and, *per contra*, those who assume the risk of loss cannot be expected to share the profits with those who have nothing at stake.

Fifth. The workmen have no means of knowing if the agreement is carried out. With their exaggerated ideas of the profits of business, the results must be in many cases disappointingly small, and they will doubt the honesty of the division. What is to be done in such a case? Invite the workmen to appoint a committee to examine the books, and report? Most employers will demur at this, and yet without it the employees can have no assurance of good faith; and were it done, what good could result? How many workmen's committees are there who are sufficiently versed in modern accounts to form any idea of the proceeds of the year's business from an examination of the books? In this light the profit-sharing plan is seen to be an agreement between two parties, the first of whom has every temptation and opportunity to cheat the second, while the second has no means of knowing if he has been cheated, and no redress in any case. In the present state of human nature this cannot be expected to be satisfactory to the second party. The fact that the plan has worked with apparent success in some instances and for considerable periods of time proves nothing. The most disastrous boiler explosions and bridge failures have been preceded by long periods of apparent safety. Even the Conemaugh dam held water for many years. It is a truism that the most rickety and unsafe devices often serve their purpose for long periods. At the beginning the workmen look on the amount received at the annual division as a bonus, and anything is better than nothing; but later on they will look on it as theirs by right of having earned it, and the above situation is certain to arise. The fact is, that the profit-sharing plan is wrong in principle, and cannot be in any large sense a solution of the wages problem.

IV. THE PREMIUM PLAN.

Taking up now the subject proper of this paper, it aims at a division of the savings due to increased production between the employer and employee, but by a direct method instead of the circuitous one of the profit-sharing plan. The plan assumes two slightly different forms, according to the nature of the work; one form being suited to work produced in such quantities as to be reducible to a strictly manufacturing basis, and the other form to the more limited production of average practice. In both forms the essential principle is the same, as follows: The time required to do a given piece of work is determined from previous experience, and the workman, in addition to his usual daily wages, is offered a premium for every hour by which he reduces that time on future work, the amount of the premium being less than his rate of wages. Making the hourly premium less than the hourly wages is the foundation stone on which rest all the merits of the system, since by it if an hour is saved on a given product the cost of the work is less and the earnings of the workman are greater than if the hour is not saved, the workman being in effect paid for saving time. Assume a case in detail: Under the old plan a piece of work requires ten hours for its production, and the wages paid is thirty cents per hour. Under the new plan a premium of ten cents is offered the workman for each hour which he saves over the ten previously required. If the time be reduced successively to five hours the results will be as follows:

1	2	3	4	5
Time consumed.	Wages per piece.	Premium.	Total cost of work = col. 2 + col. 3.	Workman's earnings per hour = col. 4 ÷ col. 1.
Hours.	$	$	$	$
10	3.00	0	3.00	.30
9	2.70	.10	2.80	.311
8	2.40	.20	2.60	.325
7	2.10	.30	2.40	.343
6	1.80	.40	2.20	.366
5	1.50	.50	2.00	.40

This table illustrates the manner in which the cost of the work diminishes and the workman's earnings increase together until, to cite the extreme case of the last line, if the output be doubled, the wages paid per piece will be reduced 33⅓%, but the workman's earnings per hour will be increased 33⅓%. Were the premium less than ten cents per hour, the reduction in cost for each hour saved would be greater, and the workman's earnings less. On the other hand, the workman would have a smaller incentive, and the time would not be reduced so much. The output would be less, and the net result might be worse for both employer and employee. This raises the inevitable question: What should be the rate of the premium? Nothing but good sense and judgment can decide in any case. In certain classes of work an increase of production is accompanied with a proportionate increase of muscular exertion, and if the work is already laborious a liberal premium will be required to produce results. In other classes of work increased production requires only increased attention to speeds and feeds with an increase of manual dexterity and an avoidance of lost time. In such cases a more moderate premium will suffice. Any attempt, however, on the part of the employer to be greedy and squeeze the lemon too dry will defeat its own object, since if a trifling premium be offered, the workman will not consider it worth while to exert himself for so small a reward, and the expected increase of output will not take place. On the other hand, if the premium offered be too high, the employer will simply pay more than necessary for his work, though less than he has been paying. If the rate of premium is decided upon judiciously, it may and should be made permanent. No cutting down of the rate should ever be made unless, indeed, improved processes destroy the significance of the first time base. Every increase of earnings is necessarily accompanied by a corresponding decrease of cost, and if the premium be such as to give these a satisfactory relation, the workman may be assured that there will be no limit set to his earnings; that the greater they are the more satisfactory they will be to the employer. The importance of this cannot be too strongly insisted upon. If the premiums be cut the workmen will rightly understand it to mean, as under the piece-work plan, that their earnings are not to be permitted to pass a certain limit, and that too much exertion is unsafe. The very purpose of the plan is to avoid this by so dividing the savings between employer and employee as to remove the necessity for cutting the rate, and hence enable the workman's earnings to be limited only by his own ability and activity. The baneful feature of the piece-work plan is thus completely obviated, and instead of periodical cuts with their resulting ill-feeling, the premiums lead the workman to greater and greater effort, resulting in a constant increase of output, decrease of cost, and increase of earnings.

The broad-minded employer will not fail to recognize that his own gain from the system comes largely from the increased production from a given plant, since not only does the system reduce the wages cost of the piece of work in hand, but in so doing it increases the capacity of the plant for other work to follow. The advantages from this source are so great as to render unnecessary any refined hair splitting as to the rate of the premium.

Such is the premium plan, and the writer confidently predicts that the more it is studied the more perfect will appear its adaptation to the requirements of industrial enterprise and human nature. Surely, a system which increases output, decreases cost, and increases workman's earnings simultaneously, without friction, and by the silent force of its appeal to every man's desire for a larger income, is worthy of attention. In addition to the commanding features noted it has others of lesser note. The transition to it from the day's-work plan is easy and natural. It does not involve a reorganization of the system of bookkeeping, but only an addition, and a small one, to the existing system. No opposition to it, organized or otherwise, is possible, since there is nothing compulsory about it, and nothing tangible to oppose. It is simply an offer to gratify one of the strongest passions of human nature, and the difficulty often found in introducing piece work cannot occur with this.

In carrying out the plan in connection with work which has been reduced to a manufacturing basis, the writer finds the following form of time ticket convenient:

Time Ticket.

Name of Part .... **No. of Pieces** ....

Operation .... **Workman** ....

HOURS.

Date.	7	7½	8	8½	9	9½	10	10½	11	11½	12	12½	1	1½	2	2½	3	3½	4	4½	5	5½	6	Over time.

Machine set by .... *Time* .... *hours.*

This ticket is issued by the foreman, the blanks at the top being filled up by him. If desired as a check he punches a hole on the line, indicating the hour when the work is given out, repeating the same when the work and ticket are returned. The record of the time is kept by drawing a line between the various hour marks, an operation which the most illiterate can perform.* The ticket provides for several days' work, and is not returned until the work is completed, when it contains the record of the entire job.† On the back of the ticket is printed the following:

"According to previous experience this work should require ... hours. If completed in less time than that a premium of ... cents will be paid for each hour saved."

When the ticket is returned, a comparison of the back with the front shows the premium earned. This is entered opposite the workman's name, in a book kept for the purpose, which is a companion to the usual time book or pay roll. On pay day the accrued premiums are paid to each workman along with the regular wages. The cost book is written up from the ticket in the usual way, except that as the ticket usually contains the record of several days' work, the labor of keeping the cost book is much abridged.

On work which, while produced as a regular product, is still not produced in sufficient quantity to justify recording the cost of each part, the premium offer is made to the group of men who carry out the work. The proposition is made as a posted notice, or otherwise in the following form:

"According to previous experience this work should require ... hours. If completed in less time than that a premium of ... cents per hour saved will be divided among those working on the machine, division to be in proportion to time spent on the work."

In this form the system loses the advantage of dealing directly with the individual, and the second objection to the profit-sharing plan is introduced, though in a modified degree, as a small group

* Attention was called to this form of time ticket by Professor Hutton in Vol. IX. of the *Transactions*, page 386.

† This rule holds, even when the job after being partly finished is interrupted by something more pressing. In such a case the ticket is taken up by the foreman in order to insure that the entries have been made for the completed work. He issues the ticket again when the work is resumed, and when all is completed this ticket goes to the office, where a single entry in the cost book records what, under the usual method, might require a half dozen or even more entries.

of men is dealt with instead of the entire force. The remaining objections to the profit-sharing plan are not introduced, and on such work the plan proposed is distinctly superior, though lacking theoretical perfection. The piece-work plan does not apply to work of this kind, and hence there can be no comparison between it and the plan under discussion.

On contract work undertaken for the first time the method is the same, except that the premium is based on the *estimated* time for the execution of the work.

The system is thus applicable to all classes of machine-shop work except "jobbing" or work done by the hour, and there is no very vociferous demand from the shops for a method of reducing the time on that class of work.

The writer believes that, judiciously administered, the plan proposed will produce a larger output and cheaper work, and at the same time pay higher wages than any other whatsoever.

DISCUSSION.

Mr. John T. Hawkins.—In this paper we have still another attempt to protect—to be plain—unfair workers and employers against encroachments upon each other's rights. It is of a piece with about all former attempts to substitute for the proper spirit of fair dealing between man and man in the two capacities, something which shall force both to observe what the whole history of this question shows cannot be so controlled.

If men would deal justly and fairly with one another in this matter, it is self-evident that the day's-work, or better, the hour's-work plan, is the ideal one; but human cupidity on both sides interferes with its proper working, and the only remedy, in the writer's opinion, is to be found in some system of educating employers and workmen alike up to the fact that their common interests lie solely in each doing to the other what he would have the other do to him. But with such an observance of the golden rule would come the millennium; and there is no very imminent prospect of great possibilities in this direction; nor will there be more than is brought about by the gradual but sure development of the race to higher ethical planes, unless some organized effort is made to school both sides up toward this ideal.

The writer fails to see in the proposed plan any surer escape from the evils of the time in the relations of employer and workman, than in the older ones which the author of the paper mentions. In point of fact, it is far below the piece-work plan, which has shown itself to be the only ameliorating system where practicable.

The author assumes a case in detail, and also assumes it possible in this case, to reduce the time required to produce a given price of work by 50%. If we apply this to an average manufacture, in which the labor constitutes one-half and material and expenses the other half of the total cost, with 10% profit to the employer to make up his selling price, we have the unexistable conditions that the employer will thereafter realize about 25% profit and the workman obtain 33% advance of wages On its face this looks very well and would seem to deal quite fairly between the two; but the true state of the case is that no manufacturer has any right to such a profit, and if the outside world would stand it, a much larger part of the saving should accrue to the workman, whose labor produces the given article; this is of course providing we are prepared to condone in the workman the fact that he was capable by a fair day's work of doing double what he actually performed under an agreed day's wages.

But there are others deeply interested in such a result: What of the consumer? He would say that these people were getting rich at his expense and altogether too rapidly, that both were in a fair way to become those terrors of the times, "robber barons;" and he looks about him for means of putting a little of this undue enrichment, of this—as he would probably term it—monopolistic combination of workman and employer, into his own pocket; then, appealing to the cupidity common to employer and employee, and taking counsel of his own, he proceeds to procure the starting up of rival concerns, who will be satisfied with more legitimately low profits, and who would obtain workmen at lower pay than that gotten by the workmen in the aforesaid combination, and thus the initial wages rate upon which 10 cents for each tenth of the time saved is advanced on the rate of premium, and the 25% profit of the manufacturer both succumb to the inexorable laws of trade and manufacture.

The author, in discussing the piece-work system, shows that when the workman has "astonished himself and all others by increasing his output far beyond the expected 25%," and "his earnings increase with startling rapidity," "while the cost of the work remains the same," and says that under these conditions the employer, in cutting the prices, practically announces to the workman that "his earnings will not be allowed to exceed a certain amount," concludes that this cutting amounts to "killing the goose which lays the golden egg." Nevertheless this goose must be killed in any event. If the employer doesn't kill it, the consumer, through his competitors, will not only kill it, but him. The fact is that no such golden eggs are to be had in industrial pursuits, and it would be most unrighteous that they should.

As the writer has had occasion to contend in a previous discussion of this subject: If a workman at piece work succeeded in producing a piece of work in one-half the length of time he previously employed himself upon at day wages, there is no escape from the fact that, when working by the day he was not only robbing his employer, but the consumer of his product as well; and it is to the credit of the piece-work system that it furnishes the means of discovering such a state of things.

The piece-work system, in the writer's opinion, furnishes the only relief from the unsatisfactory relations between employer and workman which so largely obtain under the day's wages plan; while the older and still prevalent plan of paying a given sum for a given time, with a *quid pro quo* rendered, is that which will continue to prevail and improve—as it, of all the proposed systems only, is capable of improvement—as men themselves improve. It does not lie either in profit sharing or premium paying to ameliorate the regrettable features of this older plan, nor can any such amelioration come from any system which overlooks the iron laws of supply and demand.

In all the systems proposed, and in the newer more than in the original day-wages plan, human rapacity or cupidity defeats itself and with it the most promising of these new systems, whether it exists in the workman or employer or both; and when workmen learn that their interests lie in doing the best they can for an agreed compensation by the day or hour, and the employer similarly is persuaded that his success in the long run depends upon his paying the best wages that the market for his product will permit, commensurate with a fair profit to himself; paying always the highest wages to the best and most efficient workman, as a proper and legitimate incentive to increased effort on his part, we will have as nearly the ideal conditions as are possible in any industrial system.

Mr. E. F. C. Davis.—I have known several instances where the bonus plan was adopted, by which a certain amount of time was arrived at as being customary for certain work, and then cards were given out stating that half of the time saved would revert to the mechanic and half to the shop; and that established what might be considered a minimum amount of time which each job would be likely to require with the appliances then in hand. Those shops which are doing this have arrived at what may be considered fair piece-work prices, and, after using the premium plan a little while, have dropped it as being too cumbersome, and come down to the simple piece-work plan. But by starting out with the bonus plan first, they avoided the necessity of such extensive cutting and have gotten at fair piece-work prices by the bonus system, which was too cumbersome to keep up; but I have never known anybody who carried out Mr. Halsey's plan for any great length of time.

Mr. William O. Webber.—I think Mr. Davis has rather hit the nail on the head. Success in using the piece-work system results largely from making your prices right in the first place, and I think that can be easily done by a manager who thoroughly understands his business. On the other hand, I think that Mr. Hawkins has given one of the strongest points about the whole wage problem, and that is the willingness between both the employer and the employee to be absolutely fair with each other. We have found in the Erie City Iron Works, at Erie, Pennsylvania, that treating our men in that way has resulted not only to their advantage, but to ours. We have even had men come to us and say that the piece-work price for a certain piece of work was too much, comparing it with similar pieces of a different size. Now, we take that as a pointer, and if we find that, as in many cases, the prices seem high for a certain piece, we do not see anything unfair or unbusiness-like in going to the workman who is doing that work and suggesting a revision of those prices. Sometimes they can be revised with good reason. We recently had a workman in our works come and say this. We were paying him a certain price for planing cylinders. He said if we would give him a more modern planer, he would plane those pieces for 20% less. That was certainly a fair proposition on his part, and we immediately took it up. I think that the piece-work system is the only one which will ever be successful in any way, and to make it a success, there must be a complete fairness between the employer and the employee, and the making of prices right in the first place, which any man who understands the business ought to know how to do.

Mr. William Kent.—I regret that Mr. Halsey has not given us actual data in his paper. I regret still more that Mr. Hawkins has presented an argument which is altogether *a priori* without any facts, except those drawn from experience with other systems than the premium system. Mr. Halsey some three or four years ago proposed to me this premium plan, and I fortunately was then in a position to put it in practice at once, and it has been in use in the shops of the Springer Torsion Balance Company now for three years, with satisfaction to both employer and employed. I heartily endorse the plan as admirable in every respect. It has given no trouble at all. I may admit with one of the other speakers that this plan may result finally in the piece-work plan pure and simple. Whenever experience has gone so far that you cannot improve the method of manufacture, and the workman has got into a rut, then the amount which he gets under the premium plan will be a certain amount per piece. You might just as well in that case pay him by the piece; but as long as there is any chance of improvement, and men have not reached the utmost limit of their skill or inventive powers, so long will the premium plan be a good incentive to the workmen. In practice under the working of this premium plan, we have perhaps a small piece of machine work to do, and we have no previous experience in making it, but we put a boy to work upon it, and find that by his ordinary skill in attending a machine, doing as he is told, he turns out say 100 pieces in a day. We tell him that there may be some quicker way of doing that work if he can find it out, and we tell him we will give him a quarter of a cent premium upon every piece he makes over 100 a day. Now, we have had the number of pieces turned out jump up to three hundred a day, and that simply by some little knack that the boy discovered, which he was under no obligation to discover, and which he had no incentive (except the premium) to discover; he was not cheating his employer by not inventing or discovering that method before, but when we gave him an incentive to try and discover something, why, he went to work and he succeeded and added 50% to his wages. This is an actual case, and not a hypothetical one. So I close as I began, by saying that I heartily endorse Mr. Halsey's plan, and I hope that gentlemen will not condemn the plan unheard, or until they have more data as to its actual working.

Mr. Frank H. Ball.—I am very glad indeed to hear what Mr. Kent has said on this subject. I recall—and I presume other members here recall—a paper entitled "*Gain-Sharing,*" that was read at the Erie meeting by our ex-president, Mr. Towne, describing a method very similar to the one which Mr. Halsey proposes. The difference between the two plans is simply this: In the plan of gain-sharing which Mr. Towne described, the business was divided into departments, and the men in each of these departments were given a share in whatever they would save over certain fixed prices. For instance, in a foundry they found that it cost a certain amount per ton for labor, and his plan was to divide with the men what they would save over this fixed cost. Mr. Halsey goes one step further and brings it down to the men individually. The argument which Mr. Towne made for his method as against the profit-sharing plan, was substantially the same as Mr. Halsey's argument, and he gave us to understand that the results were very satisfactory. Mr. Halsey proposes to deal directly with each man, and I think the idea is an excellent one. It seems to me that it has advantages over any other system that has been proposed. If any system other than day wages will work well, it seems to me this system will.

Mr. H. H. Suplee.—The gain-sharing plan has been in use at the works of the Yale & Towne Manufacturing Company, Stamford, and both parties are thoroughly satisfied with its operation. In discussion of this paper with Mr. Towne—and as I think he would have discussed it himself if he were present—he at once noticed the similarity of the method as being the gain-sharing method reduced to a smaller difference of subdivision—the same in principle, but only different in application.

Mr. Henry L. Gantt.—I think Mr. Kent's remarks are subject to another interpretation. If that boy using the same tools could by a little knack increase the product 300%, it seems to me that either the foreman was careless in giving him directions, or was lacking in knowledge; at all events, something was wrong to start with, and the problem resolves itself into the original question: What is a day's work?

Mr. Thos. R. Almond.—I wish to call to the attention of the members something that was stated soon after Mr. Towne's paper at one of the meetings in New York. A gentleman from Altoona stated a case which came under his immediate observation. There were two men who were requested to do their work by the piece, and during the three or four days that they were doing the work by the piece one of them went to the foreman and said, "I cannot make day pay this way," while the other man said, "I am making twice as much as I was by the day." Now, here was a case that you could not account for in any way excepting as stated by the foreman in the answer he made to the man: "The trouble is not between yourself and me; you must go to the Creator."

Mr. Hawkins.—I am prepared to admit the *a priori* character of the discussion that I presented (and I wrote it very hastily and with a view of simply eliciting discussion); at the same time, while the arguments of Mr. Kent and Mr. Suplee seem to be conclusive, from the fact that up to the present time the two systems adopted by the Springer Balance Company and the Yale & Towne Company "have been successful," they give us no measure of this supposed success, and I contend that the time is entirely too short within which they have been put into practice to settle anything in the question. The time will come with both those concerns when they will be ready to drop their new plans, or it will revert practically into the piece-work or day-work system, if it is one of those kinds of manufacture in which the piece-work plan is practicable. Unfortunately, in the machine business it is not practicable in many places. While I have no data, I venture to say that if data could be obtained from piece-work establishments, the working of that system as against the profit-sharing or the premium-paying plan would be shown to have been very much more successful. I can point to an establishment as long ago as from 1850 to 1860 where they established the piece-work system and expected to carry it out thoroughly in all departments, but they have practically dropped even this plan except in those particular special parts of the machine to which it could be most practically applied. It requires a long time to settle this question or any of these new ideas in connection with labor, and I venture to say that they will all finally come back to that one idea: that they must learn to treat one another right.

Mr. R. Van A. Norris.—This seems to be very much like the Pennsylvania Railroad's method of paying the engineers and firemen on their line a premium on coal saved. They allow a certain quantity of coal for the run, and then they pay a premium on all the coal that is saved. A plan resulting very similarly was in use in one of the copper mines in Michigan some years ago, where the mining was found to be very expensive and the powder bill was extremely high. The amount of powder used per cubic fathom of rock by each miner was posted on a bulletin board each month and a prize given to the man using the least. That resulted in a reduction of about one-third in the powder bill, and, as all work was done by contract, the result was a reduction of the contract prices and an increase in the amount of money made by the men.

Mr. E. F. C. Davis.—I think most of the objections raised against piece work are by people who have never had a good o portunity to observe how it works. It is very automatic and self-regulating in its workings. If one man is making too much on a job, and, by any particular appliance is making very much more than his employer thinks he ought to make, there are ten chances to one that somebody else in the shop will notice the money he is making and will come forward and make a bid to get that work at what he considers a fair profit for himself. So that the thing works all the time to the benefit of both parties. Whenever any new appliance is put to work, particularly by the foreman or the employer, no reasonable mechanic ever objects to having his prices changed in proportion. I think that the fact of the matter is that both the premium plan and the piece-work plan come down to very much the same thing. So far as my experience has gone, I think that people who have tried the premium plan have generally abandoned it and adopted the piece-work plan, as being a simpler way of getting about the same thing. The piece-work plan makes a little less bookkeeping.

Mr. James McBride.—The remarks this evening have proceeded entirely upon skilled labor. I want to make a few remarks about unskilled labor. The New York Dye Wood Extract and Chemical Company, of which I am superintendent, employs very largely unskilled labor. About two years ago the firm decided that they would set aside a portion of the earnings of the concern each year, to be divided among their employees

—not committing themselves in any way to their employees, but simply giving to them a certain amount of money, in January each year, when the laboring man needs it more than at any other season, all in a lump sum. That plan was adopted two years ago. The result, while not altogether satisfactory, has been a great improvement to the old system of simply paying them day's wages. We find among the skilled mechanics that those who can reason to a conclusion are very much in favor of it. Among the unskilled laborers, while a good many of them do not exactly comprehend it, they think a good deal of it, so much so that when they get a man in among them who was born tired they make it so hot for him that he is glad to get out. Our pay roll has decreased in some departments very materially from the fact that a good man refuses to work alongside of a poor man. I am in favor of some method by which a portion of the earnings can be divided among the employees. I do not know what is the best plan. We adopted this plan, as we thought it the best for our purposes, and so far as we have gone with it we are satisfied it has been an improvement. This disturbed condition of labor points to the conclusion that manufacturers in the future, in order to pacify their workmen and keep them quiet, will have to devise some means by which the workman can become a sharer in some way with his employer.

Prof. G. I. Alden.—Will the gentleman be willing to state what sum unskilled labor received, and what sum skilled labor received of this division of profit each year?

Mr. McBride.—The first year we paid the common laborers 5% upon the amount of money which they had received during the year, and we made it obligatory that the men should be prompt at their work; if they lost twenty days in the year, this per cent. was either kept from them, or it was reduced. We gave the skilled laborers 10% upon the amount which they had received during the year, and those men higher up received a little more. The first year every man received a percentage, but the second year we were obliged to make a discrimination, and those who were tardy in arriving at their work, instead of getting 5% got only 2½%, and the skilled men got 5% instead of 10%. The result of this discrimination has been that those men who have been docked have since been very regular in attendance; they have become the laughing-stock, as it were, of their fellow workmen, and they have tried to make a better record.

Mr. William H. Weightman.—I think that we might strike at the employers on this question. They are as ambitious for a profit as the employees. My dealings have been more with those operating shops, and I have found that I can save 50% by calling for contract prices, over what it would cost by arranging to pay "the actual cost" with a certain percentage of profit. I had the same party do two similar pieces of work. The first one he did by contract, and he admitted that he made 30% profit. Some six months after that I had a similar machine made, where there was a possibility of having to alter it, and, sooner than have any trouble in regard to these alterations after the contract was made, I told the party to keep track of the time and allow his own profit afterward. This one cost us three times as much as the other, in spite of the fact that we found there was no necessity for the alteration. So that of the two machines, one cost $80, the other cost $150; and while in the first instance he stated (he had forgotten the admission he had made) that he had made 30% at $80, in the other he stated that he made the exact amount that the 30% on $80 was. Thus while the employer and employee can hardly trust each other, sometimes the employer is a little doubtful himself and "the golden rule," a factor all around.

Mr. Hawkins.—In 1877 I took charge of a machinery establishment in the city of Brooklyn, and carried out the old plan. I did not know a single one of the workmen in the whole concern, but I called them all together and I talked to them just as an employer or superintendent should talk to a number of men, expecting them to do the best they could. I gave them to understand that if they did the best they could, due consideration would be given them, and reward to those who did the best in the way of wages; and a dismissal or reduction of pay of those who did the worst. In the course of two or three weeks I found it necessary to discharge half a dozen men, from the fact that I could of my own observation see that the work which they were doing was being "nursed." I discharged them as examples, and I found my action resulted in great improvement on the part of the rest of the men. I also took the other course and raised the wages of the men who were doing the best, and that made a still greater improvement. I carried that system through in that shop so long as the shop was carried on, and I think with a great deal of success; and I will place this success against that of Mr. Kent and Mr. Suplee, and I venture to say that if I were to start a shop to-morrow, I would adopt that course, keeping in proper touch with the men, and I would undersell every one of you who pay these premiums or adopt these other methods. (Laughter and applause.)

Having undersold you, the law of supply and demand steps in and forces you to modify or abandon your scheme in self-defence.

The great trouble with all these proposed systems is, that they only partially, if at all, discriminate between a man's ability and his willingness to do. The old system, properly administered, enables an employer to reward superior ability, with as good a means of discriminating against idleness or unwillingness; in which, of course, the employer is called upon to act fairly. The latter rewards a man in proportion to his ability, while the former too largely offers incentives merely to perform what was the man's obvious duty in any case. "Capital and labor could get on well enough together if there were not so many men trying to get capital without labor."

Mr. Daniel Ashworth.—I have listened very attentively to these remarks. The difficulty with so many fine-spun theories and Utopian ideas such as we have heard, is, that they are trying to get rid of one important factor, and that is the force of circumstances from the commercial side. Mr. Hawkins touched upon a very important point, when he spoke about carrying out a certain scheme to undersell his competitors. Now, mark you, this is one of the turning points. I know it from teachings of experience among many branches of industry in the United States. Systems have been developed upon this question, upon which a manufacturing concern would step forward in the market by some process of distribution of the pay, in the shape of premiums and piece work, and the gentlemen on the other side would immediately figure it out how much cheaper they could sell these goods, and eventually they would invent some plan worse than before, because these competitors would be stirred up. There is the weak point. Cupidity dominates trade, and it is a constant attempt of one to over-reach the other. There is the figuring in the office very elaborately, and it is presented as an ultimatum to the employees, "that we propose to carry out this system," and it is carried out. As has been repeatedly said, it is just like the old story of the naval distribution of prize money: you sift it on a ladder, and that which remains on the rungs is for the men, and that which falls through is for the officers.

I can point you to industries in the Ohio Valley, which for the last twenty years have had a species of competition beyond a parallel in the history of any industries in the world, and they have been going right along. They have been indulging in deception upon the great mass of their intelligent employees; they said there was no money in the business at all, that ruin was staring them in the face, and yet those establishments have been increased from one to four furnaces, and they have branched out in the different valleys where the natural fuel would reach them. With all our boasted civilization, we ignore the fact that these workmen are becoming more and more intelligent every day. When you will take the intelligent working masses and say to them, "Send your representative to us whom you see fit to appoint to discuss these questions," then you will have a starting point, and then you will have the data which have been so much talked about to-night. And right there I wish to say, as has been said by Mr. Webber, start right. When, in God's name, ever was the time that the manufacturer had the right figure? Why, he would change it in six months. (Laughter.)

I stand here to-night to champion the other side of this question, because I know I am in the minority, and yet I know I am right. We might talk until the crack of doom, and it would be one man trying to over-reach the others. Intelligent men, such as our mechanics are, should be, and they are, able to agree with their employers upon an equitable basis.

Mr. Thomas R. Almond.—A case has come under my notice to-day which illustrates what Mr. Ashworth says. This morning I went to see a prominent business man of this city, who said: "I have never met you before, sir. I had an impression that you were a tall, thin, dried-up kind of a fellow whom I didn't wish to see; your letters to me were so peculiar, your prices were always so stiff, that I formed an opinion of you that was far from pleasant; but, sir, I have learned to like your way of doing business." And he added: "It is because we are all cutting each other's throats, and trying to get each other's business that our profits are 10% instead of 20% or 25%. If manufacturers would all keep their prices so that we could rely upon them, we could all maintain our prices." I take it, sir, that Mr. Ashworth is decidedly correct. I said to my foreman four years ago: "I want you to make those goods and fix the prices yourself, and if these are satisfactory to me, I will not cut them as long as you work for me, and the market prices can be maintained." Since then I have increased his prices. I do not wish to laud my own actions in any sense, but I do believe Mr. Ashworth is correct every time when he says that the manufacturers and the dealers are more at fault than the workmen.

Mr. Weightman.—These experiences always result in four or five different concerns reducing their prices until they get to the point of cutting each other's throats, when they conclude to form a trust, binding each other to maintain a certain fixed price, by means of which they are safe against having their business ruined. Instances can readily be cited where concerns agree between themselves that they had been cutting the prices down too low on the strength of this style of competition, using employees as assistants in reducing the prices, until the concerns get in such a condition that they are not doing a safe business. They immediately conclude to form some plan by which they can control each other and keep from going down too low. A combination is formed, an agreement signed, and they work along smoothly with moderate safety. The employees still get their profits. The result is a slight advance, so that the companies as well as the employees are for the time being satisfied. If you can get employer and employee to trust each other and to believe in each other there will be some sort of an understanding in, though hardly a definite conclusion to, the labor problem.

Mr. Robert W. Hunt.—During a somewhat busy life I have been constantly thrown in contact with labor questions. In the great development of the Bessemer business we had constantly to meet new conditions, which caused all of us—both the men and the employers—considerable trouble. The product of last year would be a ridiculous amount to be satisfied with this year, and yet the same number of men would execute the work. This because inventive genius had devised, and much money had been expended in providing mechanical appliances which rendered it possible. These conditions had to be met in some way. Certainly under such conditions the men ought to be willing to work for less pay per ton, while still earning more per day. I do not wish to detain you by giving reminiscences, and will close by saying that in my judgment any step which is taken in good faith, looking toward the solution of this labor trouble, is worthy of respectful consideration; and before its trial it is dangerous for us to say that it will not work. If any gentleman at the head of an establishment is willing to make the experiment, in the name of humanity let us bid him God-speed. Let him try it, and then let him give us his results, and thus help his neighbor in his efforts to better humanity, and his own interests. Undoubtedly the foundation for the solution of these difficulties is the relations existing between the employer and the employee. If a manager is an able and fair man, he will succeed much more easily than if he is incompetent and unfair. Unfortunately there is another difficulty which has to be constantly encountered, and that is, that some man who does not work for you will come in and be supported by your men in his right to assert his authority to discuss with you your relations with your employees. I think that this is the most serious difficulty any fair-minded employer has to meet.

*Mr. Fred. A. Halsey.**—It is all very well to say, as Mr. Hawkins and others have done, that the true solution of the wages problem is for men to deal justly with one another, which in this connection means for the employee to exert himself to the utmost, and for the employer to pay as high wages as he can afford. Unfortunately, however, this is not in human nature; and, whatever the millennium may bring, the problem before the employers of to-day is to deal with human nature as it is. It is a truism that the average man will not work for others as for himself, and the average employer desires to get his product as cheaply as possible. This may seem culpable to Mr. Hawkins, but to the writer it seems natural, just, and proper. At all events it is inevitable, and it is the purpose of the premium plan to provide for, and make use of, these features of human nature by giving the employee an opportunity to work in a measure for himself, and by giving the employer the assurance of a more than proportionate increase of product from a given increase of wages.

Mr. Hawkins's objection that the plan involves the "unexistable" condition of a profit of 25%, and that the manufacturer has no right to such a profit, seems to me irrelevant. Such arguments do not prevail against facts, and while, of course, it is not to be

* Author's Closure.

expected that the system will produce a general increase of output of 100%,* it is nevertheless true that in individual operations that figure has often been exceeded, and that too when such foremanship talent as was available had not discovered anything wrong with previous records.

Mr. Hawkins's third objection, that rival concerns would spring up, and by avoiding the premiums, produce the work that much cheaper, is based upon the assumption that the premiums are a charge upon production, *i.e.*, that the same production can be had without as with them. This is the kernel of the question, and here I believe Mr. Hawkins to be wholly wrong. He here, as, indeed, all along, confounds plans based on the profit-sharing principle, viz.: the division of a percentage of the profits among the employees in the general faith that the promise of such division will buy their good will, and thereby lead them to increase their product, at least enough to make up the amount divided among them, *but which amount is to be divided whether the increase really takes place or not*, with the present system in which the premiums are paid only in case of actual increase. With the first system the increase must take place, or the cost of production is raised, but with the second system this is impossible. There could be no better place to try the question to a conclusion than such a shop as Mr. Hawkins proposes, where the day's-work method is employed in the most enlightened manner. Mr. Hawkins may find a few men (I have never found one) who will produce as much on the day's-wages method as when offered the inducement of the premium plan; but that is not, and never will be, the case with an entire force, and if a single man of the force is susceptible to such inducement the plan applied to him will cheapen the product. *If it does not reduce the cost there will be no premiums to pay.* The fact is, that the average employer, foreman, or workman who has seen machine tools operated only on the day's-work system, does not know what they are capable of producing; and to those who have seen them worked under systems analogous to the one proposed, to claim that as good results can be obtained under day's-work methods, seems absurd.

* The table on page 759 is given merely as an illustration to explain the working of the system. There is nothing about it, or in connection with it, to justify the implication that the last line of the table represents an average result to be expected from the system.

Turning now to Mr. Davis's remarks, I have no shadow of doubt that his workmen could, and to one possessing their confidence would, draw a very different picture from his as to the "automatic" feature of piece work. Mr. Davis seems to have succeeded in getting his men to compete with one another. Few have succeeded in this, the usual reduction in the piece prices being brought about by a direct cut by the employer. However, the results are the same. Each workman knows that if he push his earnings above a more or less clearly defined limit, some other workman will underbid him, and he would be foolish indeed were he too active. When men know, as under piece work all do know, that their earnings will not be allowed to exceed a certain limit, it is idle to expect them to exert themselves to pass that limit, and right here is the essential difference between the piece-work plan and the one under discussion, which some seem unable to see. The piece-work plan necessarily sets such a limit, while the present one does not. As to the system furnishing a mere stepping-stone to the piece-work plan, its case is exactly analogous to a mathematical limit. When improvement can go no further, the system may be turned into piece work; but that, like a mathematical limit, is a condition which we forever approach but can never reach.

As the system proposed has excited considerable interest I add a few practical hints to any who may feel disposed to give it a trial. The first of these is a caution against expecting the workmen to receive it with any enthusiasm; on the contrary, they will in most cases look upon it as piece work in disguise, and that system is so excessively and justly unpopular, that they will at first regard the present plan with suspicion. A little patience on the part of the employer, and a little experience on the part of the employee, will correct this, as the workmen find that the premiums are not a myth.

It has been pointed out in the body of the paper, that the system enables an employer to deal liberally with his men. At the same time there are two directions which this liberality can take. One method—and the wrong one—is to take the best record obtainable as a base, and then offer a liberal premium for its reduction. No force can be composed entirely of "stars," and if the above plan is followed it will be but a question of a short time when an inferior man will be put upon that job, who will soon find that he cannot equal the base rate, and he will cease trying. At the same time, a really superior man on the same job might push his earnings so high, in consequence of the large premium, as to tempt the employer to the fatal step of cutting down the premium. To meet this condition, I have adopted the settled policy of being liberal with the time rate rather than the premium rate, thus giving all a chance, and keeping the premiums within satisfactory limits.

Whatever may be the final policy in this particular, the system should be inaugurated at least with moderate premiums, since if its experimental premiums are made too low, no one can object to their being raised, whereas, should they be too high, it is another matter to lower them. In this connection, the table on page 759 was intended solely as an illustration of the principle of the system. As a matter of fact, a premium rate such as is there given would be altogether too high for the ordinary run of light machine-shop work, while for other and more laborious work, it might be altogether too low. As a matter of fact, I have produced excellent results with premiums as low as three cents an hour.

Finally, it is but just to himself that the author should add that this method has been in no way suggested by Mr. Towne's excellent gain-sharing system. In point of fact, the present plan was clearly formulated before the publication of Mr. Towne's paper. Its publication has been deferred in order that it might first receive practical trial, and be presented with a corresponding measure of assurance. It has now been tried in three establishments, and in each case with results such as have been described.

CCCCL.*

SOME EXPERIMENTS WITH A SCREW BOLT.

BY JAMES MCBRIDE, BROOKLYN, N. Y.

(Member of the Society.)

IT goes without saying that the screw is one of the most useful mechanical powers, and among its many good qualities is that of not "overhauling" when loaded. A large part of its efficiency as a mechanical power is sacrificed to obtain this valuable property, but aside from the necessary loss to accomplish this result, I doubt if it is generally known how much of the total power applied to turn the nut on an ordinary screw bolt is lost in friction. I am quite sure that I did not, and a number from whom I asked for the information, and who I supposed ought to know, apparently knew as little about it as myself; and it was to get some practical knowledge on the matter that these experiments were made.

The experiments were made with an ordinary two-inch screw bolt, such as can be bought anywhere in the market, and it was not specially prepared for the occasion. The pitch of the thread was $\frac{22}{100}''$ (four and one-half threads to the inch), and of the standard *V* shape; the nut was not faced and had the flat side to the washer, which was of malleable iron, not faced. The surfaces of the nut and washer in contact, as well as the threads of nut and bolt were well lubricated with lard-oil; the nut was a good fit, and when not loaded was easily run up and down the bolt with the fingers.

The nut was placed four feet six inches (4', 6") from the floor, about breast high, and rested on a cross beam, supported on wooden horses, the bolt passed through the floor and was loaded at its lower end with about 7,500 lbs. dead weight. To ascertain how much a man could pull on a nut thus placed, a nut was placed four feet six inches (4', 6") from the floor on a bolt fixed to the side of a vertical plank, and one end of a soft and very

---

* Presented at the Providence meeting of the American Society of Mechanical Engineers (1891), and forming part of Volume XII. of the *Transactions*.

pliable rope was attached to the handle of a wrench and passed over a pulley nine inches (9″) in diameter, the top of which was also on a level with the nut and immediately in front of the operator. To the other end of the rope were attached weights which rested on the floor. The operator was required to grasp the wrench with one hand on each side of the rope, where it was fastened to the wrench, and turn the nut on the bolt, which he could only do by raising the weight from the floor. This apparatus was designed to compel the operator to keep the wrench on a level with the nut or nearly so, thus making the conditions, as nearly as possible, the same as when he applied his power to turn the nut on the loaded bolt.

It was found after many trials, by different men, that they could by this device lift weights varying from 182 to 267 lbs., the latter weight being raised just clear off the floor by a very powerful man.

Following are the results of the trials with the weights:

Clark	267 lbs.
Sweeney	227 lbs.
Fay	182 lbs.
Rogers	208 lbs.
Harward	182 lbs.

When the wrench was applied to the nut on the loaded bolt, it was found that the men could just move the nut with the following length of wrench:

Clark	8 inches =	Radius,	Cir.	= 50.26 inches.
Sweeney	13 inches =	"	"	= 81.68 inches.
Fay	14½ inches =	"	"	= 91.10 inches.
Rogers	13 inches =	"	"	= 81.68 inches.
Harward	14¼ inches =	"	"	= 89.53 inches.

The length of wrench was measured from the centre of the nut to a point between the hands, and was taken as the shortest length of wrench with which the nut could be moved; and the operator was allowed to jerk or surge as he liked, with both feet on the floor, and keeping the wrench as nearly horizontal as possible.

The theoretical value of a two-inch screw bolt with a .22-inch pitch, and under the foregoing conditions is as follows:

				lbs.	lbs.		
Clark....	.22 inches	: 50.26 inches	::	267	: 60,997	7500	12.29%
Sweeney..	.22 inches	: 81.63 inches	::	227	: 84,279		8.899%
Fay......	.22 inches	: 91.10 inches	::	182	: 75,364		9.95%
Rogers...	.22 inches	: 81.68 inches	::	208	: 77,224		9.71%
Harward.	.22 inches	: 89.58 inches	::	182	: 74,065		10.12%
							5) 50.96%
						Mean,	10.19%

It will be seen from this that the practical value was only a mean of 10.19% of the theoretical, and that 89.81% of the power applied to the wrench was absorbed by the friction of nut on the washer and on the threads of the bolt.

While these tests, made as they were in a crude way, are not absolutely correct, they go to show that only about 10% of the power applied is converted into strain on the bolt.

Since beginning this paper two cases have come under my observation, which go to show how little is really known of the strains that are put upon screw bolts under various conditions.

One was that of some $\frac{7}{8}$-inch bolts, $\frac{11}{100}$-inch pitch (9 threads to the inch), the nuts of which were being screwed up with a wrench 28 inches long, operated by two men. Leaving out the element of friction, and assuming the combined force applied by the men to be only 100 lbs. (they being in a sitting position and not able to exert their full force), they should have put a strain of 159,927 lbs. on the bolt, which is about $5\frac{1}{4}$ times more than its tensile strength.

It is very evident that no such tensile strain was put directly upon the bolt, although a number of them were broken, the friction of the threads alone being sufficient so to lock the two parts together that the bolt was twisted off just below the nut.

The other was that of a $3\frac{1}{8}$ inches diameter bolt, .25 inch pitch (4 threads to the inch). The nut and its bearing were faced and well oiled, the wrench was 3 feet 6 inches long, having an eye at one end to which was attached a double and triple pulley-block tackle, operated by four men. Four men with such a tackle can just raise 1,700 lbs., which, applied to the end of the 3 feet 6 inches wrench to turn the nut on the bolt, should raise 1,794,452 lbs., which is about 4.5 times the tensile strength of the bolt. As in the former case no such strain ever reached the bolt, but assuming that 10% of it did, it will be seen that this bolt was strained so as to only leave a factor of safety of 2.1 to 1.

The text-books are painfully silent on this subject, and leave the reader in a very uncertain frame of mind how much to allow for this element of resistance. I have found but few experiments recorded even on apparatus specially designed for the purpose, and none at all on the every-day cases which arise in practice. Following are what some of the authors say:

Tomlinson, after figuring out the theoretical value giving a certain result, says: "This is of course omitting the friction, which in the case of the screw is *very great.*"

Oliver Evans, after discussing the theory and giving an example, says: "But this is supposing the screw to have no friction, of which it has a *great deal.*"

Overman says: "For calculating the effect of a screw we employ the formula to the inclined plane, *making due allowance for friction*," but doesn't say how much.

Prof. Ball says: "Experiments showed in two cases respectively about $\frac{2}{3}$ and $\frac{3}{4}$ of the power was lost."

Trautwine says: "In practice the friction of the screw (which under heavy loads becomes very great) *makes the theoretical calculations of but little value.*"

Weisbach says: "The efficiency is from 19 to 30%"

DISCUSSION.

Mr. Wilfred Lewis.—Mr. McBride's experiments confirm my own experience in the same direction, and I think he is right in the opinion that the actual loss in friction is not generally known or appreciated by engineers.

About twelve years ago I made some experiments on a square-threaded screw with a view to determining the coefficient of friction and the angular pitch of maximum efficiency from an analytical consideration of the forces involved. The efficiency of the screw was determined by experiment, and from it the coefficient of friction was deduced. Diagrams were then constructed for convenient use in the solution of problems involving the application of screws. The results of my investigations were embodied in a paper read before the Engineers' Club of Philadelphia, and afterward reprinted in the *Journal of the Franklin Institute* for February, 1880; but, as the subject still seems to be so imperfectly understood, I will venture to give a brief outline of its consideration without going into a complete analytical demonstration.

The efficiency of a screw will be seen to depend upon the coefficient of friction, the angular pitch of the thread, the shape of the thread, and the mean diameter of the supporting step or collar. All of these quantities must enter into the formula which will truly express the relation between the force applied and the weight raised or lowered by a screw, and assuming (Fig. 325)

d = mean diameter of thread,

D = mean diameter of step or collar,

p = pitch of screw,

α = angular pitch of thread to plane normal to axis of screw,

β = inclination of side of thread to same plane,

φ = coefficient of friction,

F = tangential force required at pitch line of screw to raise a unit of weight,

F_1 = tangential force required at pitch line of screw to lower a unit of weight,

n = ratio of diameters $= \frac{D}{d}$,

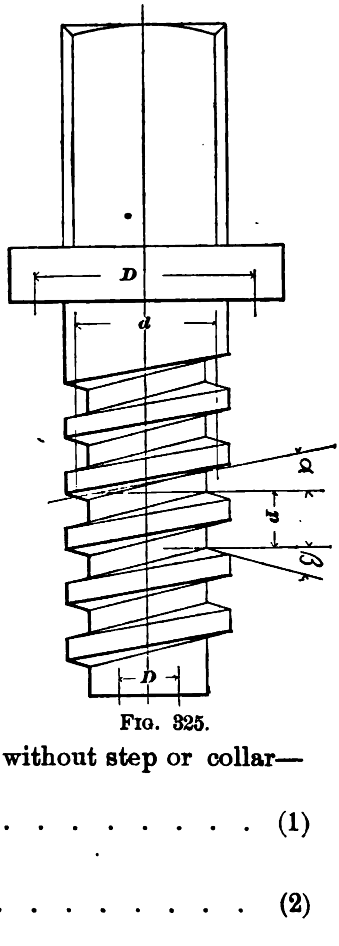


FIG. 325.

we have for a square-threaded screw without step or collar—

$$F = \frac{\varphi \cot. \alpha + 1}{\cot. \alpha - \varphi} \quad \ldots\ldots \quad (1)$$

$$F = \frac{\varphi \cot. \alpha - 1}{\cot. \alpha + \varphi} \quad \ldots\ldots \quad (2)$$

These formulæ refer to the friction on the thread alone, which is entirely independent of step or collar friction, and, to include the latter, it is simply necessary to add $n\varphi$, making for screws with step or collar the formulæ—

$$F = \frac{\varphi \cot. \alpha + 1}{\cot. \alpha - \varphi} + n\varphi \quad \ldots\ldots \quad (3)$$

$$F = \frac{\varphi \cot. \alpha - 1}{\cot. \alpha + \varphi} + n\varphi \quad \ldots\ldots \quad (4)$$

For V-threads the friction on the thread is augmented by sec. β, and the formulæ become—

$$F = \frac{\varphi \cot. \alpha \sec. \beta + 1}{\cot. \alpha - \varphi \sec. \beta} + n\varphi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$F_1 = \frac{\varphi \cot. \alpha \sec. \beta - 1}{\cot. \alpha + \varphi \sec. \beta} + n\varphi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

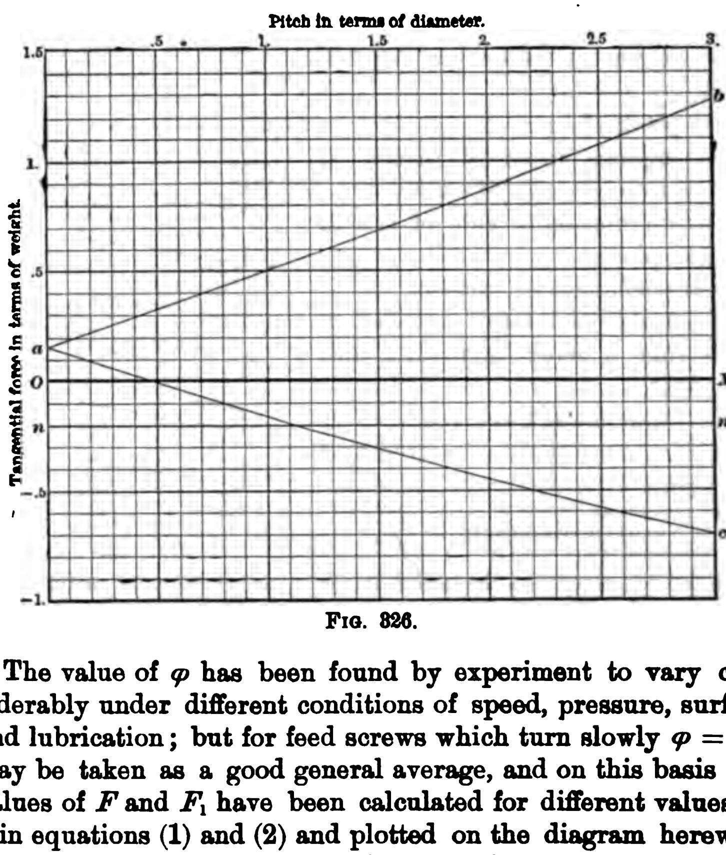


FIG. 326.

The value of φ has been found by experiment to vary considerably under different conditions of speed, pressure, surface and lubrication; but for feed screws which turn slowly $\varphi = .15$ may be taken as a good general average, and on this basis the values of F and F_1 have been calculated for different values of α in equations (1) and (2) and plotted on the diagram herewith (Fig. 326). Abscissæ denote the pitch in terms of the mean diameter of the screw and ordinates the value of F and F_1.

The amount to be added for step or collar friction is determined at once by the value of n, and when this is a fixed quantity, as in ordinary screw bolts, $n\varphi$ may be laid off below OX and a new base line nn adopted for the measurement of ordinates.

It is important to note that where ac crosses nn, the screw ceases to be self-sustaining, and, to insure stability or a positive tendency to run down as may be desired at this point, liberal allowance should always be made for possible variations in the coefficient of friction.

When a screw is used simply as a convenient method of transmitting power, another problem arises as to the angle of maximum efficiency. An increase in the pitch generally increases the efficiency of a screw, but it is evident that the pitch may be so far increased as to prevent the possibility of doing any work at all, and the question becomes, How far is it profitable to increase the pitch?

This can be answered by a consideration of the work done in terms of the useful effect produced, and letting $U =$ work done, we have from equation (3)—

$$U = F \cot. \alpha = \left(\frac{\varphi \cot. \alpha + 1}{\cot. \alpha - \varphi} + n\phi \right) \cot. \alpha \quad . \quad . \quad . \quad . \quad . \quad (7)$$

from which the minimum value of U will be found to be attained when

$$\cot. \alpha = \varphi + \sqrt{\frac{1 + \varphi^2}{1 + n}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

For $\varphi = .15$ and $n = .5$, $\alpha = 46° 23'$, but, for good practical reasons, it is not often desirable to approach this limit.

Having now given a rigid account of the losses in friction for screws in general, I would like to add some shorter and more convenient formulæ, which I have found to be applicable mentally and with a close degree of accuracy to ordinary screw bolts like the one used by Mr. McBride in his experiments, and in fact to most of the cases which occur in practice. These formulæ are given simply as approximations, and their great merit lies in their simplicity.

Let $p =$ pitch of screw.

$d =$ outside diameter of screw.

$F =$ force applied at circumference to lift a unit of weight.

$E =$ efficiency of screw.

$$\text{Then } F = \frac{p + d}{3d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

$$\text{and } E = \frac{p}{p + d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

Applying equation (10) to the bolt used by Mr. McBride we have $d = 2$ and $p = \frac{2}{9}$, whence $E = \frac{\frac{2}{9}}{2\frac{2}{9}} = \frac{2}{20}$ or .10, which is practically the same result as he obtained, agreeing with it closer than two successive experiments would be likely to agree with each other.

Mr. Arthur A. Falkenau.—The question as to what strain could be exerted on a bolt in tightening up a nut was submitted to me some years ago, and as a rough guide to the men then under me, I deduced the following formula, which is based on the principle that the moment of the frictional resistance will finally balance the moment of applied forces. In it I neglect the pitch, considering the nut as a thrust collar. (Fig. 327.)

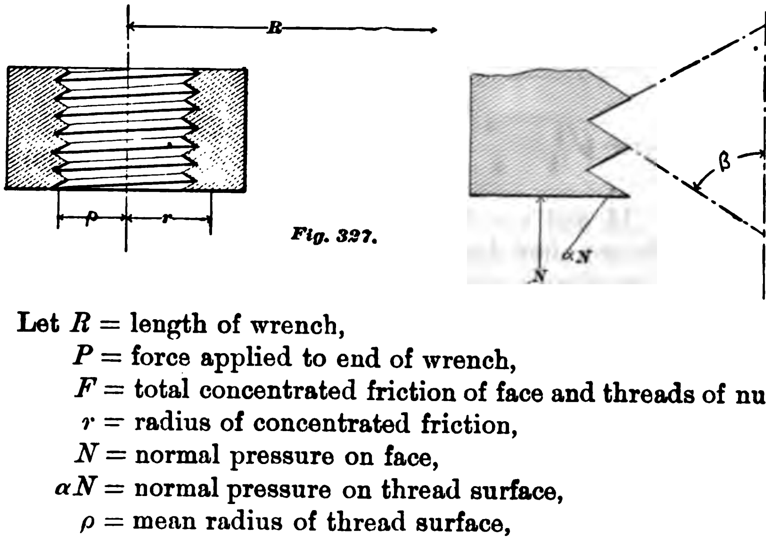


Fig. 327.

Let R = length of wrench,
P = force applied to end of wrench,
F = total concentrated friction of face and threads of nut,
r = radius of concentrated friction,
N = normal pressure on face,
αN = normal pressure on thread surface,
ρ = mean radius of thread surface,
ϕ = coefficient of friction,
β = angle which generating line of thread surface makes with axis of bolt,

The friction on the thread will be $\phi \alpha N$ and its equivalent concentrated at the end of r is $\frac{\rho}{r} \phi \alpha N$ and α = cosec. β.

The fundamental equation must be $PR = Fr$. . . . (1)

but $F = \phi N + \frac{\rho}{r}\phi\alpha N = \phi N(1 + \frac{\rho}{r}$ cosec. $\beta)$, and substituting this value in eq. (1) we have

$$PR = \phi N\ (1 + \frac{\rho}{r}\ \text{cosec.}\ \beta)\ r = \phi N(r + \rho\ \text{cosec.}\ \beta)$$

$$\text{Hence } N = \frac{PR}{\phi(r + \rho\ \text{cosec.}\ \beta)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

For ordinary bolts $\beta = 60°$ and cosec. $\beta = 1.16$.

For wrought iron on wrought iron ϕ varies from .15 to .4.

Applying the formula given in eq. (2) to Mr. McBride's experiment with Clark, and taking $\phi = .15$ we have

$R = 8''$.

$P = 267$ lbs.

$r = 1.5''$.

$\rho = 1''$.

$\beta = 60°$.

and hence $$N = \frac{267 \times 8}{.15\ (1.5 + 1.16)}\ 5{,}353 \text{ lbs.}$$

Mr. McBride having used oil on his bolt might have made ϕ less, and consequently the resulting value of N would be greater than the one just found.

CCCCLI.*

APPLICATION OF HIRN'S ANALYSIS TO ENGINE-TESTING AND A METHOD OF MEASURING DIRECTLY THE QUALITY OF THE STEAM IN THE CLEARANCE SPACES.

BY R. C. CARPENTER, ITHACA, N. Y.
(Member of the Society.)

THE methods proposed by Hirn, to be used in the testing of engines, are familiar to the members of the society from papers which have already been presented by M. V. Dwelshauvers-Dery and others.†

I have made the attempt to reduce the analysis to a form readily applicable to ordinary testing, and a large class of students have made an application to a simple condensing engine with plain slide valve and throttling governor. In addition, other tests with applications of this analysis have been made to an automatic engine and to a triple-expansion engine.

During the tests on the plain slide-valve engine, an attempt was made with a fair degree of success to measure the quality of the steam remaining in the clearance space during compression, and a description of this method constitutes the latter and more important part of this paper.

It is proposed in the paper to present the methods used and the results obtained in those tests and discussions relating to the quality of steam by various authors.

In order to call to mind the character of analysis proposed by Hirn, which I have reason to think has not been applied to engine-testing to any great extent in this country, and to show what Hirn proposed and what has been accomplished, I present the translation—in which I have endeavored to preserve the

* Presented at the Providence meeting of the American Society of Mechanical Engineers (1891), and forming part of Volume XII. of the *Transactions*.

† *Trans.*, Vol. XI., No. CCXIX., and Vol. XI., No. CCCLXXXII., papers by Messrs. Dwelshauvers-Dery and R. H. Thurston. See also the manuals by the last-named on Engines, Boilers, and Engine and Boiler Trials; N, Y; J, Wiley & Sons.

structure of the original—of a few pages from the French edition of the "*Traité des Machines à Vapeur*, par Francesco Singaglia, traduit l'Italien, par E. De Billy." (Paris, 1890.) As, no doubt, now well known, this method requires more data than the usual methods of testing, and requires more labor in reduction; but considering the information regarding the distribution of heat and power which is furnished, it will, I think, repay the closest examination, and the adoption in all tests where it may be found practicable.

HIRN'S THEORY.

2. "*Practical and General Theory of the Engine.*—The practical or experimental theory of the steam-engine, or if you please, the calorimetric method applied to that study, rests upon an examination itself of the motive power, and not upon hypotheses agreeing badly with the facts. One easily understands that an analysis of an actual engine can only be made experimentally, because it is necessary in this study to take account of the modifications arising from the permeability of the metal to heat and of various complex phenomena. To such an extent is this true that in the general theory of a heat engine it is absolutely indispensable to proceed to a special study of each phenomenon independently of the others. We shall see what important practical questions experimental analysis has investigated, and the number of erroneous opinions that it has destroyed. It is that which will resolve, by patient study of facts, all the discussions and controversies really pending."

Notice with what simplicity Professor Dwelshauvers-Dery expresses the theory of Hirn:

Between any two positions of the piston, the quantity of heat which has done external work, and that which is exchanged between the metal and the steam, form a sum equal to the difference between the amount of heat in the steam in those positions; increased, if necessary, *by the heat that may have been introduced with new steam, or diminished by that which may have left the cylinder.*

The *general* thermo-dynamical theory, which takes as true a law of constant expansion in all cases, and neglects the exchange of heat between the metal and steam, furnishes approximate results for the work and expenditure of steam, which often deviate much from the truth. This method can, nevertheless, furnish valuable indications, and lead to the discovery of real facts.

But experiment only, sanctions, corrects, and determines the advantage or inconvenience of any modification whatever, and constitutes a check or corrective for the general theory. In fact, changes which at first sight seem insignificant, perceptibly modify the economy in the development of an engine or machine. Thus the increase of intelligence regarding the exhaust, and the condition of the steam from the admission to the release, have produced in the engine an economy increasing as 9 to 100 in the consumption of steam.

3. It is well known that, in addition to Mariotte's law for the expansion of steam, there is that of adiabatic expansion, which consists in supposing that, the vapor being homogeneous between the metal and liquid in contact, there is no exchange of heat in spite of the difference in temperature. Very probably the steam at the end of admission will not be homogeneous; nevertheless we can allow that it may be so, not knowing what hypothesis to have recourse to, without falling absolutely into arbitrary methods; but it is impossible to admit that there is no exchange of heat.

METHOD OF TESTING USED BY HIRN.

4. "*Experiments at the School in Alsace*—In order to experiment with an engine, it is necessary to make it work for one or more days under a determinate system of pressure and expansion, performing constant work. The quantity of feed-water used by the boiler is measured for one day, for example, while taking care that the level does not vary, or at least that it is the same at the beginning and end of the experiment; then dividing this quantity by the number of strokes of the piston, the amount of water used at each stroke will be obtained. It is also necessary to measure the quantity of water mechanically entrained by the steam entering the cylinder, the quantity of water of condensation, its temperature on entering the condenser, and also its temperature on leaving the condenser, the pressure of the steam, and the amount of work indicated."

When it is possible, and when the force of the engine does not exceed a certain limit, 100 H. P., for instance (for beyond that limit the operation would be dangerous), measure at frequent intervals, from hour to hour, and even oftener—the effective work with a Prony brake, taking care that at each measure the engine returns to its former condition. If one cannot make use of the brake, it is necessary to measure the work of the unloaded engine by means of the Indicator. From this it is easy to calculate the work from friction when the engine works with a useful load.

5. In a series of well-known experiments, Hirn has demonstrated the thermometric action of the metallic walls of the cylinder, and he has determined numerically the error which is made when one estimates the work and expenditure of steam on the hypothesis of impermeability of the walls to heat.

All the ideas that we have regarding heat engines proceed from the science of thermodynamics; as Hirn would say were he able, "without this science any experimental theory is impossible." But we ought to notice that if this science is indispensable in guiding us in the analysis of a machine not already constructed, it cannot be of use to us in the previous study of that machine since we ignore the thermic conditions of the working of the steam.

The quantity of water entrained mechanically, which varies from boiler to boiler, and even in the same boiler varies from one instant to another, the fall of pressure between the boiler and cylinder, which depends likewise on the dimensions of the admission ports, the exchange of heat between the metal and steam, etc., are quantities not to be determined by pure theory. The indicated work is a quantity essentially experimental and can be determined only by the steam-engine indicator. But the indicator diagram once obtained we are able, by the application of physics and thermodynamics, to determine the weight of the steam, the temperature, pressure, the heat given up to the walls or that which has been returned from the walls.

But this only suffices to establish that the practical results are quite different from those which the general theory furnishes; it was still necessary to measure the amount of error made in order to deduce its industrial importance. That is what Hirn has done, aided by two experimenters of uncommon ability, Le Loutre and the regretted Hallauer, who was the most correct expounder of his master's thoughts. Thanks to them it is now established that the walls whose action varies with the construction of the engine, with the state of the steam, saturated or superheated, by the use of the steam-jacket or the condenser, have a chief influence over the yield of work and the thermic transformations.

For V-threads the friction on the thread is augmented by sec. β, and the formulæ become—

$$F = \frac{\varphi \cot. \alpha \sec. \beta + 1}{\cot. \alpha - \varphi \sec. \beta} + n\varphi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (5)$$

$$F_1 = \frac{\varphi \cot. \alpha \sec. \beta - 1}{\cot. \alpha + \varphi \sec. \beta} + n\varphi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (6)$$

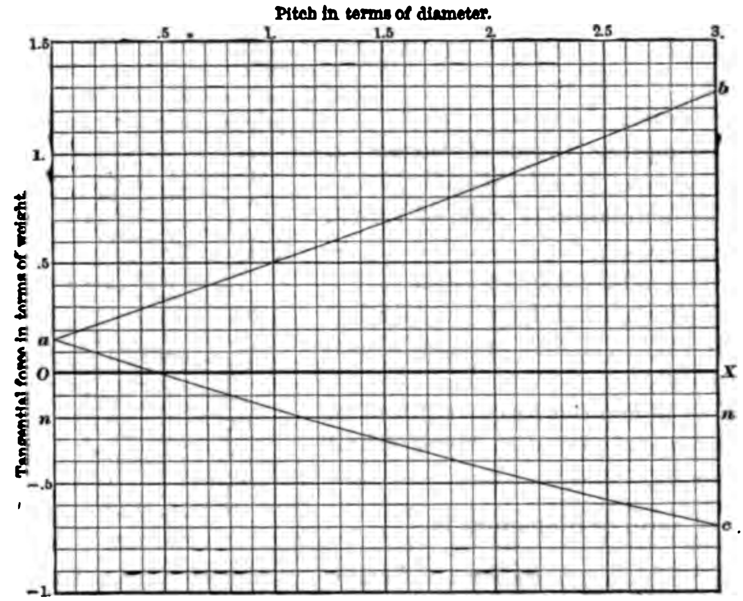


Fig. 326.

The value of φ has been found by experiment to vary considerably under different conditions of speed, pressure, surface and lubrication; but for feed screws which turn slowly $\varphi = .15$ may be taken as a good general average, and on this basis the values of F and F_1 have been calculated for different values of α in equations (1) and (2) and plotted on the diagram herewith (Fig. 326). Abscissæ denote the pitch in terms of the mean diameter of the screw and ordinates the value of F and F_1.

The amount to be added for step or collar friction is determined at once by the value of n, and when this is a fixed quantity, as in ordinary screw bolts, $n\varphi$ may be laid off below OX and a new base line nn adopted for the measurement of ordinates.

It is important to note that where ac crosses nn, the screw ceases to be self-sustaining, and, to insure stability or a positive tendency to run down as may be desired at this point, liberal allowance should always be made for possible variations in the coefficient of friction.

When a screw is used simply as a convenient method of transmitting power, another problem arises as to the angle of maximum efficiency. An increase in the pitch generally increases the efficiency of a screw, but it is evident that the pitch may be so far increased as to prevent the possibility of doing any work at all, and the question becomes, How far is it profitable to increase the pitch?

This can be answered by a consideration of the work done in terms of the useful effect produced, and letting U = work done, we have from equation (3)—

$$U = F \cot. \alpha = \left(\frac{\varphi \cot. \alpha + 1}{\cot. \alpha - \varphi} + n\phi\right)\cot. \alpha \quad . \quad . \quad . \quad . \quad . \quad (7)$$

from which the minimum value of U will be found to be attained when

$$\cot. \alpha = \varphi + \sqrt{\frac{1 + \varphi^2}{1 + n}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

For $\varphi = .15$ and $n = .5$, $\alpha = 46° 23'$, but, for good practical reasons, it is not often desirable to approach this limit.

Having now given a rigid account of the losses in friction for screws in general, I would like to add some shorter and more convenient formulæ, which I have found to be applicable mentally and with a close degree of accuracy to ordinary screw bolts like the one used by Mr. McBride in his experiments, and in fact to most of the cases which occur in practice. These formulæ are given simply as approximations, and their great merit lies in their simplicity.

Let p = pitch of screw.

d = outside diameter of screw.

F = force applied at circumference to lift a unit of weight.

E = efficiency of screw.

$$\text{Then } F = \frac{p + d}{3d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

and $E = \frac{p}{p + d}$ . . . . (10)

Applying equation (10) to the bolt used by Mr. McBride we have $d = 2$ and $p = \frac{2}{9}$, whence $E = \frac{\frac{2}{9}}{2\frac{2}{9}} = \frac{2}{20}$ or .10, which is practically the same result as he obtained, agreeing with it closer than two successive experiments would be likely to agree with each other.

Mr. Arthur A. Falkenau.—The question as to what strain could be exerted on a bolt in tightening up a nut was submitted to me some years ago, and as a rough guide to the men then under me, I deduced the following formula, which is based on the principle that the moment of the frictional resistance will finally balance the moment of applied forces. In it I neglect the pitch, considering the nut as a thrust collar. (Fig. 327.)

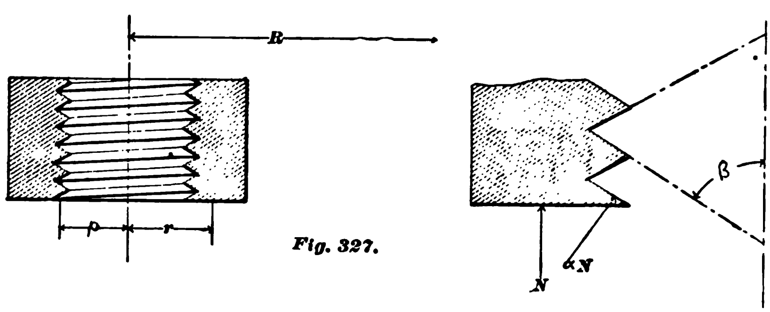


Fig. 327.

Let R = length of wrench,
P = force applied to end of wrench,
F = total concentrated friction of face and threads of nut,
r = radius of concentrated friction,
N = normal pressure on face,
αN = normal pressure on thread surface,
ρ = mean radius of thread surface,
ϕ = coefficient of friction,
β = angle which generating line of thread surface makes with axis of bolt,

The friction on the thread will be $\phi \alpha N$ and its equivalent concentrated at the end of r is $\frac{\rho}{r} \phi \alpha N$ and α = cosec. β.

The fundamental equation must be $PR = Fr$. . . . (1)

but $F = \phi N + \frac{\rho}{r}\phi\alpha N = \phi N(1 + \frac{\rho}{r}$ cosec. $\beta)$, and substituting this value in eq. (1) we have

$$PR = \phi N\ (1 + \frac{\rho}{r} \text{ cosec. } \beta)\ r = \phi N (r + \rho \text{ cosec. } \beta)$$

$$\text{Hence } N = \frac{PR}{\phi (r + \rho \text{ cosec. } \beta)} \quad \ldots\ldots\ldots\ldots \quad (2)$$

For ordinary bolts $\beta = 60°$ and cosec. $\beta = 1.16$.

For wrought iron on wrought iron ϕ varies from .15 to .4.

Applying the formula given in eq. (2) to Mr. McBride's experiment with Clark, and taking $\phi = .15$ we have

$R = 8''$.
$P = 267$ lbs.
$r = 1.5''$.
$\rho = 1''$.
$\beta = 60°$.

and hence $$N = \frac{267 \times 8}{.15\ (1.5 + 1.16)}\ 5{,}353 \text{ lbs.}$$

Mr. McBride having used oil on his bolt might have made ϕ less, and consequently the resulting value of N would be greater than the one just found.

CCCCLI.*

APPLICATION OF HIRN'S ANALYSIS TO ENGINE-TESTING AND A METHOD OF MEASURING DIRECTLY THE QUALITY OF THE STEAM IN THE CLEARANCE SPACES.

BY R. C. CARPENTER, ITHACA, N. Y.

(Member of the Society.)

THE methods proposed by Hirn, to be used in the testing of engines, are familiar to the members of the society from papers which have already been presented by M. V. Dwelshauvers-Dery and others.†

I have made the attempt to reduce the analysis to a form readily applicable to ordinary testing, and a large class of students have made an application to a simple condensing engine with plain slide valve and throttling governor. In addition, other tests with applications of this analysis have been made to an automatic engine and to a triple-expansion engine.

During the tests on the plain slide-valve engine, an attempt was made with a fair degree of success to measure the quality of the steam remaining in the clearance space during compression, and a description of this method constitutes the latter and more important part of this paper.

It is proposed in the paper to present the methods used and the results obtained in those tests and discussions relating to the quality of steam by various authors.

In order to call to mind the character of analysis proposed by Hirn, which I have reason to think has not been applied to engine-testing to any great extent in this country, and to show what Hirn proposed and what has been accomplished, I present the translation—in which I have endeavored to preserve the

---

* Presented at the Providence meeting of the American Society of Mechanical Engineers (1891), and forming part of Volume XII. of the *Transactions*.

† *Trans.*, Vol. XI., No. CCXIX., and Vol. XI., No. CCCLXXXII., papers by Messrs. Dwelshauvers-Dery and R. H. Thurston. See also the manuals by the last-named on Engines, Boilers, and Engine and Boiler Trials; N, Y; J. Wiley & Sons.

structure of the original—of a few pages from the French edition of the "*Traité des Machines à Vapeur*, par Francesco Singaglia, traduit l'Italien, par E. De Billy." (Paris, 1890.) As, no doubt, now well known, this method requires more data than the usual methods of testing, and requires more labor in reduction; but considering the information regarding the distribution of heat and power which is furnished, it will, I think, repay the closest examination, and the adoption in all tests where it may be found practicable.

HIRN'S THEORY.

2. "*Practical and General Theory of the Engine.*—The practical or experimental theory of the steam-engine, or if you please, the calorimetric method applied to that study, rests upon an examination itself of the motive power, and not upon hypotheses agreeing badly with the facts. One easily understands that an analysis of an actual engine can only be made experimentally, because it is necessary in this study to take account of the modifications arising from the permeability of the metal to heat and of various complex phenomena. To such an extent is this true that in the general theory of a heat engine it is absolutely indispensable to proceed to a special study of each phenomenon independently of the others. We shall see what important practical questions experimental analysis has investigated, and the number of erroneous opinions that it has destroyed. It is that which will resolve, by patient study of facts, all the discussions and controversies really pending."

Notice with what simplicity Professor Dwelshauvers-Dery expresses the theory of Hirn:

Between any two positions of the piston, the quantity of heat which has done external work, and that which is exchanged between the metal and the steam, form a sum equal to the difference between the amount of heat in the steam in those positions; increased, if necessary, *by the heat that may have been introduced with new steam, or diminished by that which may have left the cylinder.*

The *general* thermo-dynamical theory, which takes as true a law of constant expansion in all cases, and neglects the exchange of heat between the metal and steam, furnishes approximate results for the work and expenditure of steam, which often deviate much from the truth. This method can, nevertheless, furnish valuable indications, and lead to the discovery of real facts.

But experiment only, sanctions, corrects, and determines the advantage or inconvenience of any modification whatever, and constitutes a check or corrective for the general theory. In fact, changes which at first sight seem insignificant, perceptibly modify the economy in the development of an engine or machine. Thus the increase of intelligence regarding the exhaust, and the condition of the steam from the admission to the release, have produced in the engine an economy increasing as 9 to 100 in the consumption of steam.

3. It is well known that, in addition to Mariotte's law for the expansion of steam, there is that of adiabatic expansion, which consists in supposing that, the vapor being homogeneous between the metal and liquid in contact, there is no exchange of heat in spite of the difference in temperature. Very probably the steam at the end of admission will not be homogeneous; nevertheless we can allow that it may be so, not knowing what hypothesis to have recourse to, without falling absolutely into arbitrary methods; but it is impossible to admit that there is no exchange of heat.

METHOD OF TESTING USED BY HIRN.

4. "*Experiments at the School in Alsace.*—In order to experiment with an engine, it is necessary to make it work for one or more days under a determinate system of pressure and expansion, performing constant work. The quantity of feed-water used by the boiler is measured for one day, for example, while taking care that the level does not vary, or at least that it is the same at the beginning and end of the experiment; then dividing this quantity by the number of strokes of the piston, the amount of water used at each stroke will be obtained. It is also necessary to measure the quantity of water mechanically entrained by the steam entering the cylinder, the quantity of water of condensation, its temperature on entering the condenser, and also its temperature on leaving the condenser, the pressure of the steam, and the amount of work indicated."

When it is possible, and when the force of the engine does not exceed a certain limit, 100 H. P., for instance (for beyond that limit the operation would be dangerous), measure at frequent intervals, from hour to hour, and even oftener—the effective work with a Prony brake, taking care that at each measure the engine returns to its former condition. If one cannot make use of the brake, it is necessary to measure the work of the unloaded engine by means of the Indicator. From this it is easy to calculate the work from friction when the engine works with a useful load.

5. In a series of well-known experiments, Hirn has demonstrated the thermometric action of the metallic walls of the cylinder, and he has determined numerically the error which is made when one estimates the work and expenditure of steam on the hypothesis of impermeability of the walls to heat.

All the ideas that we have regarding heat engines proceed from the science of thermodynamics; as Hirn would say were he able, "without this science any experimental theory is impossible." But we ought to notice that if this science is indispensable in guiding us in the analysis of a machine not already constructed, it cannot be of use to us in the previous study of that machine since we ignore the thermic conditions of the working of the steam.

The quantity of water entrained mechanically, which varies from boiler to boiler, and even in the same boiler varies from one instant to another, the fall of pressure between the boiler and cylinder, which depends likewise on the dimensions of the admission ports, the exchange of heat between the metal and steam, etc., are quantities not to be determined by pure theory. The indicated work is a quantity essentially experimental and can be determined only by the steam-engine indicator. But the indicator diagram once obtained we are able, by the application of physics and thermodynamics, to determine the weight of the steam, the temperature, pressure, the heat given up to the walls or that which has been returned from the walls.

But this only suffices to establish that the practical results are quite different from those which the general theory furnishes; it was still necessary to measure the amount of error made in order to deduce its industrial importance. That is what Hirn has done, aided by two experimenters of uncommon ability, Le Loutre and the regretted Hallauer, who was the most correct expounder of his master's thoughts. Thanks to them it is now established that the walls whose action varies with the construction of the engine, with the state of the steam, saturated or superheated, by the use of the steam-jacket or the condenser, have a chief influence over the yield of work and the thermic transformations.

ZEUNER'S OBJECTIONS.

6. The experiments of the Alsatian school were recently the object of profound criticism on the part of Zeuner, who found fault especially with the relative results in the action of the cylinder walls. The objections of Zeuner were refuted by Hirn and Hallauer.

Zeuner's first objection has reference to an hypothesis of Hirn, who had supposed by inference in his experiments the pressure of the boiler equal to that of the cylinder during admission, thus neglecting the throttling of the fluid vein at the entrance to the cylinder. Hirn recognized the justice of this criticism, but he demonstrated by examples that his calculations contain an approximation more than sufficient for the estimate of heat given up to the walls, if one neglects the weight of steam in the clearance space. Neither does Zeuner admit as correct the method of calculation often used by the Alsatians, which consists in neglecting absolutely to consider the weight of vapor enclosed in the clearance space. It should be observed here that in condensing engines, working without compression, this weight is small because of the feeble pressure of steam, and that afterward it might be computed by supposing that its density remains constant. But it is certainly not the same if there is compression, for this produces phenomena which are far from being negligable.

But the principal point of Zeuner's criticism is, that he offers equal objection to Hirn's second hypothesis. When he wishes to take account of the steam that remains behind the piston at the beginning of compression; that is to say, at the end of the exhaust, Hirn *considers it as saturated and dry*. According to Zeuner the thermic phenomena which the Alsatians attribute to the action of the walls exclusively, would be due in part only to those walls, and would be due in a larger proportion to a certain quantity of water which would always be found in the cylinder, so that the vapor would never be saturated and dry, and that Hirn's hypothesis would be inadmissible. But Hirn denies absolutely that water can exist permanently behind the piston when superheated steam is made use of, or when the cylinder is enclosed by a steam-jacket; he admits that it can exist in a steam-engine using saturated steam and without a steam-jacket; but he shows that this quantity is always very little and that it still comes from the action of the walls. Hallauer has proved in his turn by means of new calculations the hypothesis in question. He has taken up the analysis of the experiments of 1873 and 1875, supposing that, 1st, the compression is too small to be taken into account, and that the weight of the vapor and water behind the piston is nothing; 2d, that the steam which is compressed in the cylinder is not a quantity to be ignored, but that it is saturated and dry; 3d, the vapor which is compressed is not to be neglected, but contains a determinate weight of water.

From his calculations he finds that the differences between the results, which the three hypotheses produce, are within the limits of experimental errors.

In regard to the effect of the steam in the clearance space, Zeuner in *La Theorie Mécanique de la Chaleur*, makes the following statements:

"Clearance exerts an important influence upon the working. Its inconveniences are mainly due to the fact that steam is admitted into a space holding steam of feeble tension, so that the engine uses more than one without such dead spaces. It is true that the work which corresponds to this additional mass of steam is not entirely lost, since it shares in the expansion. In spite of this the work is too great to be neglected." Further, he states that if the compression be carried until boiler pressure is reached, "the engine then acts as one without clearance," but in that case the work of compressing the steam may equal the gain by the process.

"While the piston retrogrades a certain space a corresponding volume of steam leaves the cylinder; the steam remaining behind probably containing a little water. Now as the mixture is compressed, there will probably result a partial or total vaporization of the water; and it may happen that at the limit of compression the steam in the dead space will be superheated. Unfortunately, we do not yet know how the mixture of this superheated vapor and the saturated steam from the boiler is effected; but this question is of little importance, the main object being to show that compression is accompanied by a vaporization of the water, mixed with steam, so as to cause superheating in the steam left behind."

APPLICATION OF HIRN'S ANALYSIS.

The following pages present in as brief a form as possible the method of application of Hirn's analysis, as adopted for a regular laboratory exercise for the advanced students in Sibley College.

The special test presented is one of about thirty which have been made, and in which the results accord fairly well with the average of a term's work.

This test was made under my directions by P. J. Darlington, F. B. Cowan, and S. E. Hitt, students in the senior class.

The detailed method of making the analysis is easily understood, and is as follows: First, make a run as explained with constant load, taking determinations of quality of entering steam, pressure and weight. The latter may be obtained either by condensing the steam or measuring the feed-water.

Also, measure the condensing water and the gain in temperature.

The accompanying log (see form No. 1) is provided with columns for these measurements and such others as may be necessary, or can be taken as a check on the work. The log is especially adapted to the throttling calorimeter, with which most of the determinations of the quality of steam were made. In case the steam was too wet for the throttling calorimeter, which I believe happened in only one case in our trials, the separator calorimeter was used. A calorimeter, to be described later, was attached to the steam-cylinder, and was provided with an automatic valve, so that steam could be admitted to the calorimeter during any desired portion of the stroke by the action of the engine. The quality of the steam in the exhaust may in some instances be a determinate quantity, in which case valuable checks would be had on the accuracy of all the previous calculations, but in the trials in question such determinations were not made, although space is provided in all the forms used. All apparatus used was calibrated and results recorded on the blank form No. 2.

Indicator cards were taken at frequent intervals during the run, as shown on the log. On these cards were laid off initial line for volume, in accordance with the measures of clearance, and initial line of pressure in accordance with the barometer reading. The cards were then subdivided by ordinates drawn to point of cut-off, release beginning of compression and admission, thus dividing the diagram into four parts, corresponding to the periods of admission, expansion, exhaust, and compression.

In the figures (Figs. 230 and 231) these ordinates are drawn from the points *b*, *c*, *d*, and *e* respectively. The measure of these ordinates to the proper scale gives the absolute pressures and are filled in the proper columns in the special blank form 3. The areas, into which the card is divided, are then measured for the external work, thus in Fig. 231 the area *brmncb* is proportional to the external work of admission; *cnpec* to that of expansion; *epste* minus *sfdos* to that of exhaust; *dorbd* to that of compression. It is to

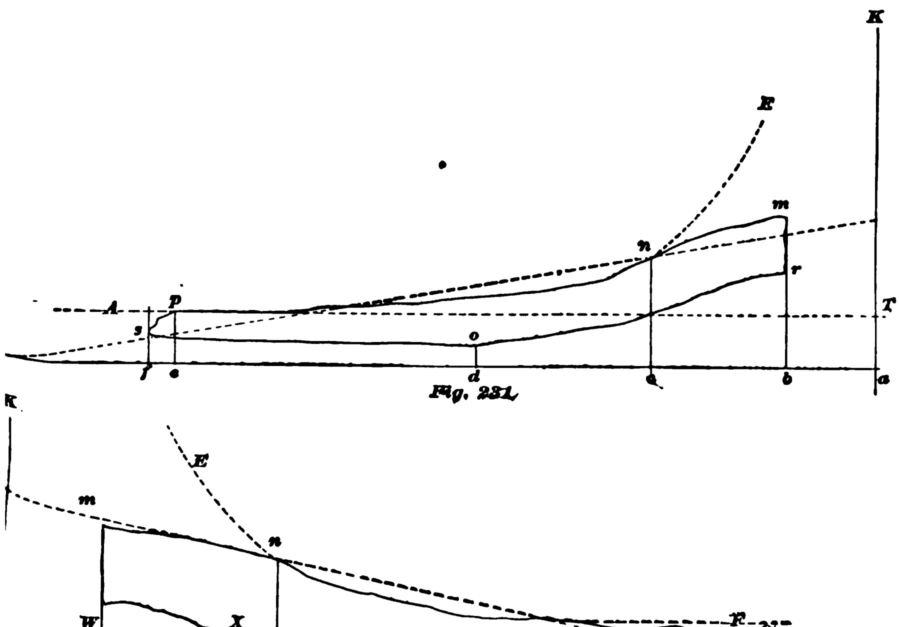


Fig. 231

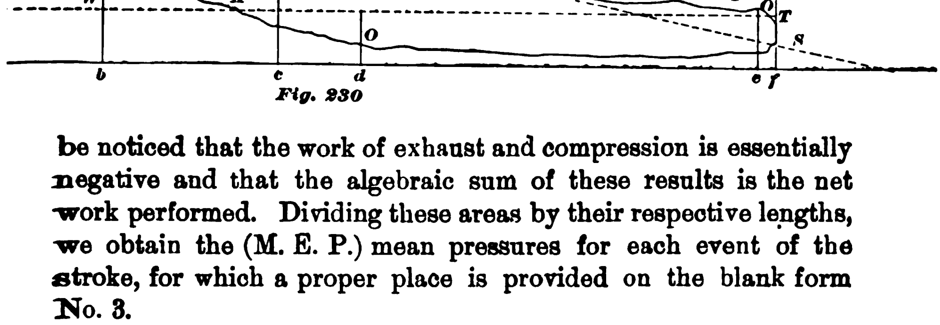


Fig. 230

be noticed that the work of exhaust and compression is essentially negative and that the algebraic sum of these results is the net work performed. Dividing these areas by their respective lengths, we obtain the (M. E. P.) mean pressures for each event of the stroke, for which a proper place is provided on the blank form No. 3.

The external work expressed in foot pounds is obtained by taking the product of the area of the piston in square inches into the distance, into the M. E. P for that event and for each stroke; this was multiplied by 100, thus making the unit 100 strokes instead of one, to avoid small decimals. These were placed in the proper place on Form 3 and reduced to B. T. U. by dividing by Rowland's equivalent, 778.

The remaining portion of Form 3 is filled by reference to a standard steam-table and the quantities arranged so that they can be referred to by using the symbols as given on the sides with subscripts as given at the head of the columns.

Form 4 gives average measurements, dimensions of engine, and symbols used. Form 5 gives the methods of calculation used in the application of Hirn's analysis, as already explained. The symbols used are those of Buels' steam-tables, which Professor Thurston has republished in his work on *Engine and Boiler Trials*, and which was convenient for our use in the mechanical laboratory.

The principle of the analysis has already been explained, and an excellent exposition will be found in the work on *Thermodynamics* by Professor Peabody; yet at the risk of being tedious I will explain some details of the method.

The weight of steam is supplied by weighing feed-water or condensed steam; this was subdivided between head and crank in proportion to the work done at the respective ends of the cylinder, as shown by the indicator cards.

The condensing water was weighed and subdivided in the same manner, thus giving us the value of G in the formula. This division is in some respects an arbitrary one, yet as the work done at the two ends of the cylinder is rarely exactly the same, it seems fair to suppose that the steam consumed is proportional to the work done in either end, although it will probably make very little difference in most cases.

To find the weight of steam in the clearance space M, we need to know its pressure and quality. The pressure is taken from the indicator card, and is already tabulated in Form 3; from this and the known volume (see Form 3) the weight of dry steam filling the clearance space at the instant of admission can be found. The heat supplied is evidently equal to $M(XL + S)$ in which X is the quality of entering steam, S the heat of the liquid, L the latent heat corresponding to the entering pressure, as on Form 3. The heat in the steam in the clearance would be found in a similar way, if we knew its quality; which in this case we did know, by the calorimeter reading, to be slightly superheated at the end of compression. At the best there is some doubt about the accuracy of this determination for quality, and farther the percentage of moisture, while it may be a large percentage of weight, is always a small percentage of volume; it will be much more accurate to ascertain the heat existing at intermediate points from the volume rather than the weight; by consulting Form 5 it will be seen that this method is the one pursued. The steam was assumed to be dry and saturated in the clearance space, which agrees very well with the actual determination.

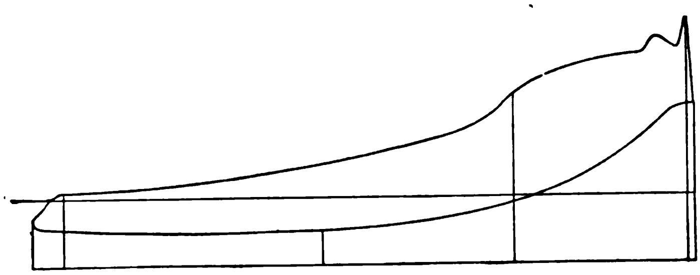


Fig. 232

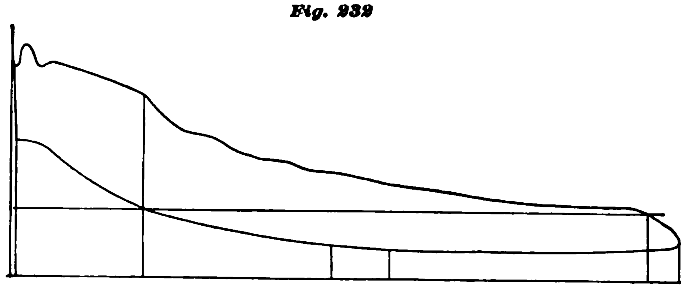


Fig. 233

A check on the numerical work is found by the agreement of the values for D and D' at the bottom of the page. These results are in either case the difference between the heat supplied the engine and that thrown off in the exhaust, in other words radiation-loss and leakage, etc.

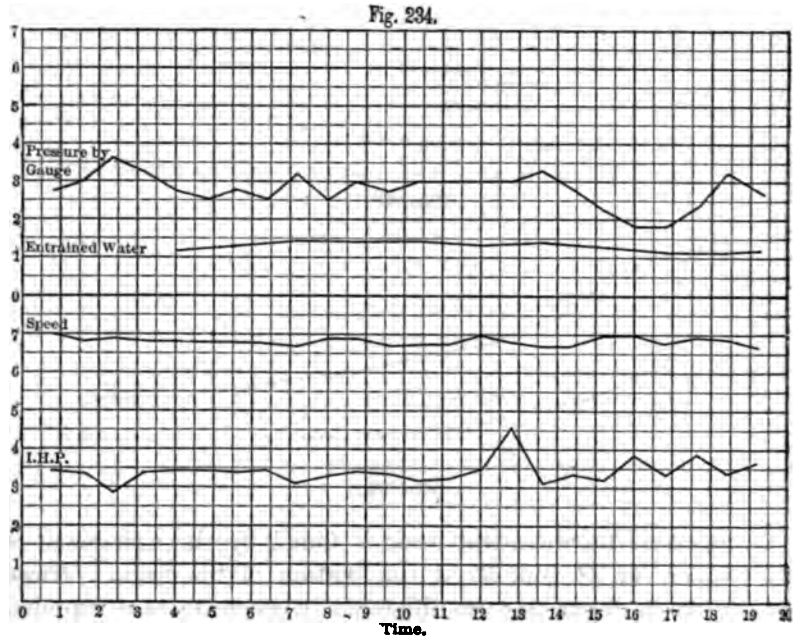
Form 6 is the summary of the preceding forms, and also shows what the quality of steam in various positions in the cylinder must have been in order to harmonize with the heat distribution, as shown in Form 5.

The method of computing the heat is the ordinary calorimetric formula. Weight $(XL + S)$ = heat in B. T. U. The external and internal heat is given for several events of the stroke on Form 5; the corresponding temperatures, pressures, and heat of evaporation are given on Form 3; the weight during admission and compression is equal to $M + M_0$ regardless of the quality, while at the beginning and end of compression the weight is equal to that filling the clearance space at the beginning of admission, or is equal to M_0 regardless of its quality. Solving these various equations for X gives the results shown on Form 6.

$$X = \frac{\dfrac{\text{Total heat}}{\text{Weight}} - S}{L}$$

In case the work of expansion is not considered, the divisor should be I instead of L. In many cases, the quality is more con-

Fig. 234.



veniently found by dividing the weight of dry steam by its actual weight. The weight of dry steam is found, in any case, by dividing the volume by the specific volume of one pound.

Thus in symbols:

$$X = \frac{V_0 + V}{(M + M_0)\,C}$$

The forms and tests referred to are as follows: Fig. 232 and Fig. 233 are tracings of Indicator diagrams taken on the test. Fig. 234 is the Graphical Log of test, made by J. E. Kress at a later date.

MECHANICAL LABORATORY, SIBLEY COLLEGE, CORNELL UNIVERSITY.

[Form 1.] LOG OF ENGINE TRIAL. HIRN'S ANALYSIS, FORM 1. DATE, FEB. 18, 1891.

		Revolut'ns.		Gauge Readings.						Temperatures.										Weights.			Head.		Crank.			Brake.		
				Pounds.			Inches Hg.									Calorimeter.														
Number.	Time.	Continuous Counter	Speed Indicator.	Boiler.	Steam Pipe.	Steam Chest.	Exhaust.	Condenser.	Barometer.	External Air.	Engine Room.	Condensed Steam.	Feed Water.	Injection Water.	Discharge Water.	Steam Pipe.	Steam Chest.	Cylinder.	Exhaust.	Condensed Steam.	Feed Water.	Injection Water.	M. E. P.	I. H. P.	M. E. P.	I. H. P.	I. H. P. Total.	Load.	D. H. P.	Scale of Spring Corrected.
1	220	83312	209	75 pounds.	Not taken.	37.	17.5		29.288.	Not taken.	75°	86	Not taken.	37	55					359.05.	Not weighed.	5627.4 pounds.	14.1	1.65	15.5	1.78	3.43	30.45 pounds constant.		Head 37. Crank 40¼.
2	225	83349	203			40.						88		38	56								14.1	1.63	15.5	1.72	3.35			
3	230	Indicated 24556.8 revolutions.	206			46.	16.00					86		39	67			237	170				11.9	1.39	13.0	1.47	2.86			
4	235		204			42.5	16.75					85		38	79		213	248	170				13.9	1.60	15.8	1.76	3.36			
5	240		204			37.5	18.00					85		40	88		216	252	162				14.1	1.63	15.4	1.71	3.34			
6	245		203			35.	20.00					86		40	93		212		153				14.1	1.63	15.4	1.71	3.34			
7	250		203			37.5	19.25					89		40	97		213	235	157				14.1	1.63	15.6	1.74	3.37			
8	255		203			35.	19.25					94		40	99		213	243	157				14.3	1.65	16.0	1.78	3.43			
9	300		202			42.5	18.00					97		40	101		214	251	165				13.0	1.50	14.4	1.20	3.10			
10	305		206			35.	18.00	20.25				100		39	102		214	245	165				13.6	1.59	15.4	1.74	3.33			
11	310		206			40.	16.00	18.75				100		39	102		213		169				14.1	1.65	15.4	1.74	3.39			
12	315		201			37.5	16.00	18.75				99		39	95		213	227	169				14.1	1.60	15.6	1.71	3.31			
13	320		203			40.	16.00	18.75				97		39	87		213	235	171				13.4	1.54	15.1	1.67	3.21			
14	325		203			40.	15.50	17.75				94		39	91		214	242	173				13.6	1.56	15.4	1.71	3.27			
15	330		210			40.	15.50	18.00				95		40	100		215	242	173				14.7	1.74	15.4	1.76	3.50			
16	335		204			40.	16.00	18.50				101		40	105		213	242	170				19.5	2.25	20.6	2.30	4.55			
17	340		200			42.5	15.50	17.50				110		40	109		215	242	173				13.2	1.50	14.4	1.58	3.08			
18	345		202			38.	15.00	17.00				117		41	112		215	246	174				13.7	1.58	15.6	1.74	3.32			
19	350		209			32.5	15.50	18.25				117		40	112		222	255	172				12.8	1.52	14.2	1.63	3.15			
20	355		211			28.	16.70	19.25				117		40	110		222	255	168				15.5	1.86	17.2	1.98	3.84			
21	400		203			34.	17.50	20.00				114		40	107		214	257	165				13.0	1.50	16.0	1.78	3.28			
22	405		209			42.5	17.50	20.00				110		40	104		217	259	166				16.2	1.90	17.8	2.03	3.93			
23	410		207			36.	17.25	19.50				108		39	103		216	258	185				13.9	1.61	15.2	1.74	3.38			
24	415	94165	202			40.	15.50	18.00				105		39	101		212	258	171				15.7	1.78	17.2	1.89	3.67			
Avge.	2hrs.		204.64	75 lbs.		38.35	16.83	18.61	29.288	40	75	99.28		39.4	95.04		214	260	167.3	359.		5627.	14.2	1.65	15.5	1.76	3.41	30.45		

Clearance, Head lbs. water 1 lb	Length Crank Pin........	*Test Made by*
Clearance, Crank lbs. water 1 lb. 2 oz..........	Dia. Wrist Pin, inches......	P. J. Darlington,
Brake Arm 21.88..............	Travel Valve, inches......	F. B. Cowan,
Dia. Cylinder, inches, 6 oz.....	Lap of Valve, inches.......	and
Length Stroke, 8 oz	Areas, square inches	S. E. Hitt.
Dia. Piston Rod, inches, $1\frac{1}{16}$.	Piston...........	
Dia. Crank Pin, inches	Steam Port...........	
	Exhaust Port...........	

[Form No. 2.]

CALIBRATION OF INSTRUMENTS.

J. E. KRESS, - - - - - - - - - *Thursday*, JAN. 13, 1891.

STEAM ENGINE INDICATORS.

	Head.	Crank.
Maker's Name	Ashcroft	Ashcroft.
Maker's Number	627	627.
Scale of Spring	60 lbs	60 lbs.
Number of Spring	627	627.
When Tested	Jan. 27, 1891	Jan. 27, 1891.
How Tested	Dead weight	Dead weight.
Per cent. Error	0	5% below.

STEAM GAUGES.—(See Fig. 237.)

Maker.	Position.	Number.	Error—Lbs.	When Tested.	How Tested.
Ashcroft	Steam Chest	9239	—10	Jan. 27, 1891	Hg. Column.

THERMOMETERS.

Position.	Registered Number.	Boiling Point.			Freezing Point.			Barometer.
		Reading.	Per Barom'r.	Error.	Reading.		Error.	
Discharge	6450				32.0		0	29.288
Condensing H_2O	6443				31.9		—.1	"
Condensed Steam	6461	210.6	210.7	—.1	32.1		+.1	"
Steam Chest	6437	210.5	210.7	—.2	32.		0	"
Compression	6456	210.0	210.7	—.7	32.		0	"
Exhaust	6476	210.8	210.7	—.1	32.		0	"

[Form No. 3.]

ABSOLUTE PRESSURES FROM INDICATOR DIAGRAMS AND CORRESPONDING PROPERTIES OF SATURATED STEAM.

		Entering Steam.	Cut-off.	Release.	Beginning. Compression.	Beginning. Of Admission.	Symbols. Thurston.	Symbols. Peabody.
Subscripts used			1	2	3	0		
Absolute Pressure	Head	52.6	32.40	14.34	6.63	27.68		
	Crank		37.29	14.55	5.74	26.93	P	p
Heat of Liquid	Head	253.6	223.7	17.94	130.6	214.8		
	Crank		232.1	180.2	136.6	213.3	S	q
Internal Latent Heat	Head	338.3	860.1	894.6	933.2	867.2		
	Crank		853.6	893.9	923.3	868.3	I	ρ
Latent Heat Evaporation	Head	914.9	935.8	966.7	1000.9	942.1		
	Crank		930.0	966.2	996.6	943.2	L	r
Total Heat	Head	1168.5	1159.6	1146.2	1131.4	1156.9		
	Crank		1162.2	1146.4	1133.2	1156.5	H	λ
Vol. 1 lb. Cu. Ft.	Head	7.97	12.63	27.24	72.56	14.65		
	Crank		11.08	26.89	64.67	15.04	C	μ
Volumes Symbols			V_0+V_1	V_0+V_2	V_0+V_3	V_0		
Volumes Head, Cu. Ft.			.0507	.1493	.0820	.0160		
Volumes Crank, Cu. Ft.			.0428	.1380	.0760	.0180		

MEAN PRESSURES AND HEAT EQUIVALENTS OF EXTERNAL WORK.

	Subscripts.	Head End. Mean Pressures	Head End. External Work. 100 Strokes. Foot lbs.	Head End. External Work. 100 Strokes. B. T. U.	Crank End. Mean Pressures	Crank End. External Work. 100 Strokes. Foot lbs.	Crank End. External Work. 100 Strokes. B. T. U.
Symbols.		M E P	W	W÷778	M E P	W	W÷778
Admission	a	10.0	198.4.	24.26	8.28	14976.	19.25
Expansion	b	13.8	26063.	33.50	15.8	28661.	36.84
Exhaust	c	2.6	—4948.	—06.36	2.5	—4559.	—5.86
Compression	d	6.7	—13646.	—17.54	6.2	—11125.	—14.30
Total	......	14.5	26343.	33.86	15.5	27953.	35.93
" from all cards	......	... .	....	(34.23)	....	....	(36.52)

[Form 4.]

APPLICATION OF HIRN'S ANALYSIS.

DATA AND RESULTS.

Test of steam-engine made by P. J. Darlington and F. B. Cowan at Sibley College, Cornell University.

Kind of engine	Plain slide-valve.
Diameter of cylinder	6″
Length of stroke	8″
Diameter of piston-rod	$1\frac{3}{16}$
Volume cylinder, crank end	.1252
Volume head end	.1312
Volume clearance cubic feet, head	.0160
" " " " crank	.0180
Boiler pressure by gauge	80
Boiler pressure absolute	95
Revolutions per hour	12,278
Absolute pressure of steam entering	52.6
Quality of steam in compression	101.3
Weight of condensed steam per hour	179.52
Clearance in per cent. of stroke	12.3
" " " " "	13.5
Barometer	29.288
Boiling temperature, atmosphere pressure	210.8
Steam used during run, pounds	359.05
Quality of steam in steam-chest	.9767
Quality of steam in exhaust	Not determined.
Pounds of wet steam per stroke (average)	.007308
Head	.007893
Crank	.006724
Temperatures condensed steam	99.28
Temperatures condensing water, cold	39.4
Hot	95.04
Pounds of condensing water, per hour	2818.5
Per stroke	.2291

SYMBOLS.

To denote different portions of the stroke, the following subscripts are used: Admission (a); expansion (b); exhaust (c); compression (d).

To denote different events of the stroke, the following sub-numbers are used: Cut-off (1); release (2); compression, beginning of (3); admission, beginning of (0); in exhaust (5); in condenser (6).

Quality of steam denoted by X.

Cut-off, crank end per cent. of stroke	19.74
Cut-off, head end per cent. of stroke	26.52
Compression, crank end per cent. of stroke	46.2
Compression, head end per cent. of stroke	57.00
I. H. P.	3.413
Brake horse-power	2.569
Release, crank end, per cent	95.56
Release, head end, per cent	97.26
Pounds of steam per I. H. P.	52.59
Pounds of steam per brake H. P.	69.9

[Form 5.]

HIRN'S ANALYSIS—DATA AND RESULTS.

PER 100 STROKES.

Payne Engine. - *Date, Feb.* 18, 1891.

QUANTITIES.	SYMBOL.	FORMULA.	RESULTS. Head.	Crank.	Total.
Wt. steam per 100 strokes, lbs	M		.7893	.6724	1.4617
Wt. of dry steam in clearance, lbs	M_0	$V_0 + C_0$	.1078	.1180	.2258
Wt. of steam, total	$M + M_0$		.8971	.7904	1.6875
Condensing water, lbs	G		12.37	10.54	22.91
Heat given to condensing water, B. T. U	K	$G\ (S_k - S_i)$*	705.0	600.5	1305.5
Heat supplied engine, B. T. U	Q	$M(X L + S)$	905.7	771.6	1677.3
Heat retained by compression, B. T. U	Q_0	$M_0 S_0 + 100 \frac{V_0 I_0}{C_0} X$ (a)	126.0	138.1	264.1
External heat steam at cut off, B. T. U	H_1	$(M + M_0) S_1$	200.7	183.4	384.1
Internal heat steam at cut off, B. T. U	H_1'	$100 (V_0 + V_1) \frac{I_1}{C_1}$	345.3	329.7	675.0
Cylinder loss during admission, B. T. U	Q_a	$Q + Q_0 - H_1 - H_1' - \frac{1}{778} W_a$	461.5	377.4	838.9
Loss external heat during expansion	H_2	$(M + M_0)(S_1 - S_2)$	39.74	41.02	80.76
Internal heat after "	H_2'	$100 (V_0 + V_2) \frac{I_2}{C_2}$	490.3	458.7	949.0
Cylinder loss during expansion, B. T. U	Q_b	$H_2 + H_1' - H_2' - \frac{1}{778} W_b$	—138.8	—124.8	—263.6
External heat at exhaust	H_e	$(M + M_0) S_2$	160.9	142.4	303.3
External heat at compression	H_3	$M_0 S_3$	16.52	16.12	32.64
Internal heat at compression	H_3'	$100 (V_0 + V_3) \frac{I_3}{C_3}$	111.5	109.1	243.3
Heat delivered from condenser	H_c	$(M - M_x) S_6$	53.10	45.24	98.34
Heat carried off in exhaust	H_4	$M(X L_5 + S_5)$ (per calorimeter)	2		
Cylinder loss, exhaust, B. T. U	Q_c	$H_e + H_2' - H_3' - K - H_3 - H_c - \frac{W_c}{778}$	—273.94	—164.0	—438.1
" " " "	Q_c	$H_e + H_2' - H_3' - H_3 - H_4 - \frac{W_c}{778}$	....	....	....
External heat, loss during compression	H_5	$M_0 (S_3 - S_0)$	—7.76	—9.05	—16.81
Internal heat at admission	H'	$100 \frac{V_0 I_0}{C_0}$	102.9	112.7	215.6
Cylinder loss during compression, B. T. U	Q_d	$H_5 + H_3' - H' - \frac{W_d}{778}$.—Calorim. loss + $M_0 S_3$	10.31	1.65	11.96
Heat admitted	Q	....	905.7	771.6	1677.3
Heat discharged and external work	B	$H_c + K$ + total $W \div 778$ + Cal. loss	801.13	681.7	1482.8
Loss	D	$Q - B$	104.6	90.0	194.6
Loss	D'	$Q_a + Q_b + Q_c + Q_d$	104.5	90.1	194.6

*$S_k - S_i$ = gain of temperature condensing water. (a) Corrected for calorimeter determination. M_x = calorimeter loss.

[Form 6.]

MECHANICAL LABORATORY, SIBLEY COLLEGE, CORNELL UNIVERSITY.

HIRN'S ANALYSIS—SUMMARY AND RESULTS.

Name of Observers, Darlington and Cowan. - *Date, Feb.* 13, 1891. - *Engine, Payne.*

Quantities.	Symbol.	Formula.		Results. Head.	Crank.	Total.
Total heat per 100 strokes, admitted	Q	Q	B. T. U.	905.7	771.6	
" " " at cut-off	h_1	$H_1 + H_1'$	"	546.0	513.1	
" " " at release	h_2	$He + H_2'$	"	651.2	601.1	
" " " exhausted	h_5	$Hc + K$	"	758.1	645.8	
" " " at compression	h_3	$H_3 + H_3'$	"	121.5	125.2	
" " " at admission	h_0	$H_3 - H_5 + H'$ *	"	126.0	137.9	
Quality of steam, entering	X	Per calorimeter	Per cent.			97.67
" " cut-off	X_1	$[h_1 + (M + M_0) - S_1] + I_1$	"	46.75	44.75	48.74
" " release	X_2	$[h_2 + (M + M_0) - S_2] + I_2$	"	62.95	61.04	64.86
" " in exhaust	X_5	$[(h_5 + M) - S_5] + L_5$ *	"	78.81	72.68	78.30
" " "		Per calorimeter	"			
" " at compression	X_3	$[(h_3 + M_0) - S_3] + I_3$ *	"	99.1	99.75	99.5
" " at admission	X_4	$[(h_0 + M_0) - S_0] + I_0$	"	104.13	104.3	104.3
" " in compression		Per calorimeter	"	101.3		
Per cent. lost, admission, initial cyl. condensation	a	$Q_a + Q$	"	50.95	48.91	
" restored, expansion	b	$-(Q_b + Q)$	"	15.33	16.18	
" lost, exhaust	c	$Q_c + Q$	"	—27.73	—21.26	
" lost compression	d	$Q_d + Q$	"	3.64	.21	
" utilized, work	w	$\frac{1}{778} W + Q$	"	3.78	4.78	
" lost, radiation	r	$D + Q$	"	11.51	11.65	
" lost (as per card), radiation	r'	$D' + Q$	"	11.51	11.65	
Ratio, radiation to work		$r + w$		3.05	2.46	
" cylinder condensation to work		$a + w$		14.48	15.33	
Thermodynamical efficiency %	E	$(t - t_3) + (461 + t)$	Per cent.			15.68
Actual efficiency %	E_1	$\frac{1}{778} W + Q$	"	3.78	4.78	
Efficiency, compared with perfect engine %	E'	$E_1 + E$	"	24.10	30.19	
Radiating surface of engine	s	To be measured	sq. ft.			5.2
Loss per sq. ft. per hour	f	$D + S$ (time of 100 revs., hrs.)	B. T. U.			4463.
" " " per degree, diff. temp.		$f + (t' - t'')$	"			

t = temp. entering steam; t_3 = temp. exhaust; t'' = temp. room; t' = av. temp. steam; S_5 = heat liquid in exhaust.
Subscript 5 applies to exhaust.
*Small corrections for calorim. loss.

CALORIMETRIC MEASUREMENTS OF THE QUALITY OF STEAM DURING COMPRESSION.

The attempt to measure the quality of the steam during compression was made as follows: A throttling calorimeter of the form shown in the cut, Fig. 235, was connected directly to the cylinder-head, with a whistle valve intermediate, which was arranged as is usual with such valves, with a lever for opening and a spring for closing. The opening lever was modified by the addition of a second lever, thus virtually making a compound lever, arranged so that a very slight motion of the first lever was suffi-

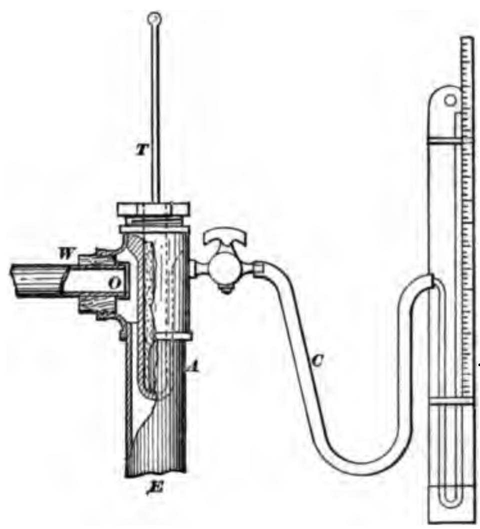


FIG. 235.

cient to open the valve fully. The lever was connected by a wire to a rod, moved by the reducing motion of the engine, by a cam which was adjustable and capable of being set so that the calorimeter valve could be opened at any required point of the stroke. A second cam could be set to release it at any instant, in which case the rod and valve actuated by springs closed instantaneously. The principal part of this ingenious mechanism was designed by Mr. C. L. Stanton, one of the mechanicians and instructors in Sibley College shops. The cut, Fig. 236, shows the apparatus complete and in working order, except for the lagging which was around it when in use. By referring to the indicator diagram, Fig. 230, it is seen that a very small portion of the compression line lies above that of the atmosphere, and that the calorimeter could discharge into the air only during the time that the piston was passing from x to w. As the engine made about 200 revolutions per minute, this practically amounted to an instantaneous opening occurring 200 times per minute. Great pains were taken to see that the calorimeter valve opened and closed at the proper instant, and that it was tight when closed. So far as could be determined, this action was perfect, the strong springs drawing the valve closed, the instant the trigger was released, with the rapidity of a gun-lock. Now, as to the results: We found that it took a long time, frequently an hour, for the thermometer in this calorimeter to reach its maximum temperature, but that except in a few cases (some thirty trials being made) this thermometer showed either slightly superheated steam, or steam very nearly saturated. The pressure of the steam was calculated as the average of that shown by the indicator diagram for the time the valve was open.

There was a throttling calorimeter attached to the steam-chest in order to measure the quality of the entering steam. The distance between the two calorimeters was not over 18 inches. Both were lagged and in similar conditions, yet the thermometer in the one on the cylinder and open only during compression would often attain a reading 30 or more degrees higher, and invariably indicated 2% to 3% less moisture. Thus in the trial already presented (see Form 1), this thermometer read 258°, when the thermometer in the other calorimeter read only 216°.

It seems hardly probable or possible that the maximum reading of the thermometer opened for so short an interval could in any event be equal to the temperature of the steam in the interior of the cylinder, yet it will be interesting to compare this with the results of the calculations made on Form 6. In these calculations it was usual to assume the steam filling the clearance space as dry and saturated, as explained previously; but in this particular trial the calorimetric determination, as made by observation, was substituted. We have, then, the following result (Form 6): Quality of steam in clearance space, 101.3, or in other words superheated 25.8°; quality of entering steam, 97.67%; quality of steam at cut-off, 48.74%; quality of steam at end of expansion, 64.86%; quality of steam in the exhaust, 73.81%; that remaining at the beginning of compression, 99.5%, while that at the end of compression is 104.13%, which corresponds to a superheat of 18°.3, thus making the cycle nearly perfect, and indicating a

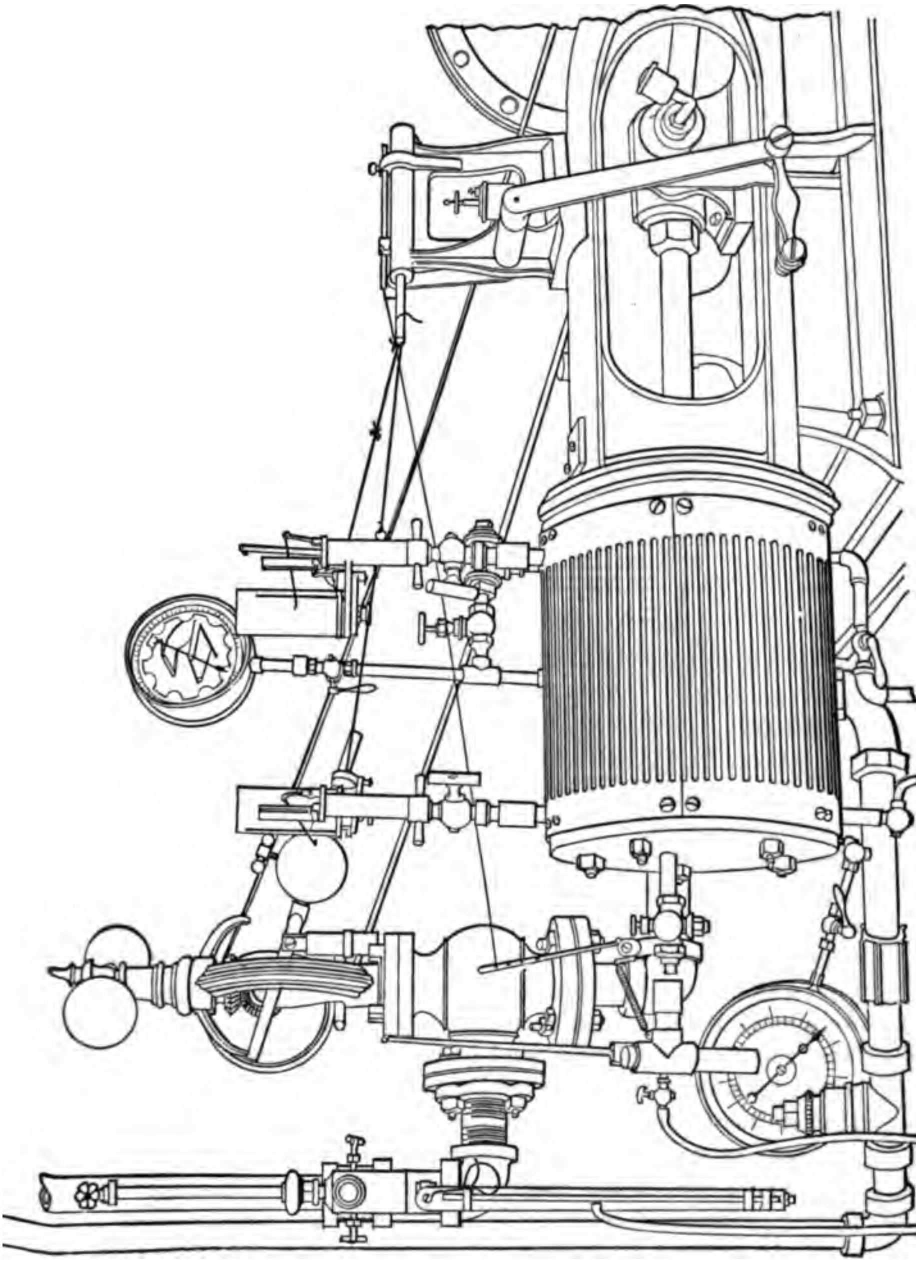


FIG. 236.

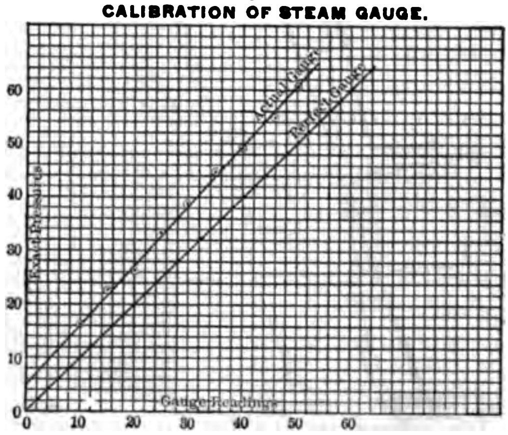
very small error. This investigation seems to indicate that Zeuner was correct in regard to many, perhaps all, of his statements regarding the quality of the steam in the cylinder, but that the error in the assumption of dry and saturated steam in the clearance space, as usually made in the application of Hirn's analysis, is not a serious one.

QUALITY OF THE EXHAUST STEAM.

The quality of the steam in the exhaust was determined by Hirn in 1869, by attaching a calorimeter made of a brass coil with a closed vessel attached to its lower extremity. By filling the closed vessel with water and boiling it, air would be expelled

Fig. 237.

CALIBRATION OF STEAM GAUGE.



from the coil, and by closing a valve and condensing the remaining steam, a vacuum would be formed. Attaching the coil in this condition to the exhaust-pipe and surrounding the coil and vessel with condensing water; by drawing in steam from the exhaust-pipe and condensing it, and by measuring the temperatures and weights of the entering steam, the condensing water before and after the steam was condensed, all the elements for computing the quality were obtained. According to Hallauer, Hirn obtained as the results of some of these trials the following figures as the percentages of moisture in the exhaust:

1st. In a condensing engine using superheated steam, at 231° C., the per cent. of moisture in the exhaust was equal to 4.67.

2d. In several trials with different engines using moist steam, he obtained respectively 10.91%, 11.80%, and 25.23%.

In the measurement of the moisture in the steam exhausted from the high-pressure cylinder to the intermediate cylinders not jacketed, made at Rochester, N. Y., in January, 1891, I obtained respectively: 1st, 14.04%; 2d, 12.12%; 3d, 12.5%; 4th, 13.35%; or, on the average, 13.35%.

The moisture can be computed by considering the condenser as a calorimeter, in which case

$$X = \frac{\frac{K}{M} - \left(S_5 - S_6\right)}{L_5} \text{ symbols as before.*}$$

In this case $x = 81\%$, or about 6% in excess of that obtained by computation from the indicator card, as explained in Form 6. These measurements are, however, not necessarily the same, since the results in Form 6 are computed for the instant of leaving the cylinder, and this latter for the steam in the condenser, when the pressure was lower, and when consequently more of the water would be vaporized.

HALLAUER'S EXPERIMENTS.

In connection with the loss by cylinder condensation and the consequent condensation of steam, we quote the following from O. Hallauer, in article on "Dead Space in Woolf Engines," *Van Nostrand's Magazine*, Vol. XIV. The article is largely devoted to an investigation of Zeuner's assumption, already referred to, of the effect of the steam occupying the clearance spaces of the engine; the result of the calculation confirming Zeuner's hypothesis of a gain of heat during compression, in excess of the work of compression. In the example considered of a Woolf engine with a jacket, working with and without compression, the gain amounted to 9 95%.

"It is to be regretted that Zeuner has not taken into account the disturbing effects of the walls of the cylinders; an effect pointed out by Hirn in his edition of 1865, and which experiments have established beyond a doubt. . . . The walls which absorb heat during admission condense often a considerable portion of the steam which flows from the boilers, and restore a part during the period of expansion. This occurs by means of the coating of water covering the walls.

* *Engine and Boiler Trials;* Thurston. N. Y.: J. Wiley & Sons, 1891, §§ 41–46.

"The action is quite complicated and modifies in a great degree all the series of phenomena, of which work is the resultant. It seemed impossible to neglect its effects in the determination by experiments of the influence of compression and clearance upon consumption. . . .

"In single-cylinder engines in which the clearance can be reduced to a small percentage of the volume, and in which the remaining steam is not greatly compressed, all the walls of the dead space as well as the cylinder are cooled during the period of discharge into the condenser. When the next charge of steam comes from the boiler, it encounters walls not sufficiently heated by the compression, and condenses, giving up a considerable quantity of heat. This condensation may amount to 60% of the steam admitted when there is no jacket.

"The heat is partially restored during expansion; it allows of a continual evaporation of the coat of water which covers the walls previously warmed. At last the envelope of steam diminishes condensation and increases evaporation in the cylinders.

"These processes, combined with the disappearance of heat due to expansion, are the two sole causes that affect the law of expansion, and fix the value of the exponent of that law." As the surfaces differ, the condensation during admission and the evaporation during expansion will be changed as well as the work of expansion, all being causes that modify the exponent of the law of expansion, and which show the falsity of the hypothesis upon which the formulas for work have been based." He states further: "In engines, such as those of Hirn and Corliss, the effect of the clearance space can be neglected, without serious error; but in engines where it approaches 10 to 15%, this is no longer the case, and the effect must be considered."

The following tables, taken from the work on Thermodynamics, by Prof. C. H. Peabody, will be interesting as showing the quality of the steam as obtained by calculation in two noted series of experiments:

QUALITY OF STEAM IN THE CYLINDERS.—ISHERWOOD'S EXPERIMENTS.

Manner of running.	Revolutions.	Cut-off small cylinder.	Cut-off large cylinder.	Total expansions.	Boiler pressure by gauge.	Per cent. of steam in the cylinder. Cut-off small cylinder.	End of stroke small cylinder.	End of stroke large cylinder.	Steam per horse-power per hour.
Steam cut-off in both cylinders by independent cut-off valves.	221.5	0.4	0.361	7.12	129.4	98	*102	81	16.4
	215.9	0.4	.336	7.12	127.2	90	*103	80	16.0
	192.2	.35	.335	7.93	104.5	87	93	76	16.7
	181.1	.335	.346	8.21	91.5	81	93	72	17.4
	166.5	.336	.371	8.21	75.0	80	91	74	18.7
	145.5	.353	.368	7.89	57.7	67	78	64	20.9
	111.5	.358	.376	7.79	32.4	66	49	45	25.0
	94.7	.361	.349	7.74	21.3	56	78	83	32.7
Steam cut-off in small cylinder only.	188.1	.329	....	8.35	103.7	87	96	80	18.5
	167.3	.325	....	8.42	74.6	83	96	85	20.1
	145.9	.347	....	7.99	55.8	75	86	80	23.9
Steam cut-off in large cylinder only.	197.9	....	.297	3.20	61.7	......	*113	85	21.0
	122.5	....	.363	3.20	21.0	......	83	64	28.6
No. cut-off.	189.5	....	....	3.20	57.0	......	*105	86	28.1
Large cylinder used as a simple engine.	191.5	....	0.335	2.66	44.9	......	......	89	25.5
	147.6	....	0.371	2.43	21.4	....	....	81	32.0

* Superheated.

Average barometer, 30.2.
Quality of entering steam not given.

QUALITY OF STEAM IN THE CYLINDER.—HALLAUER'S EXPERIMENTS.

Name of engine.	Date of test.	Quality of steam supplied.	Revolutions per minute.	Real cut-off, or reciprocal of ratio of expansion.	Horse-power cheval à vapeur.		Absolute pressure in kilos per sq. meter.		Per cent. of water in cylinder.		
					Indicated.	Net.	Boiler.	Back pressure.	At cut-off.	At release.	Entrance to condenser.
1 Hirn	Nov. 18, 1873.	Superheated 231°	30.1736	0.2570	144.36	....	48900	3680	6.5	12.0	4.9
2 Hirn	Nov. 28, 1873.	Saturated	30.5494	.2570	136.46	....	46380	3670	30.4	25.2	9.4
3 Hirn	Aug. 26, 1875.	Superheated 215°	29.969	.2139	135.77	....	49938	1919	20.38	17.5	7.9
4 Hirn	Aug. 27, 1875.	*Superheated 223°	30.306	.4539	125.17	114.0	48075	1900	−1.5	13.2	3.1
5 Hirn	Sept. 7, 1875.	Superheated 195°	29.98	.1628	113.08	102.0	49680	1881	24.64	21.38	8.1
6 Hirn	Sept. 8, 1875.	Saturated	30.41	.1628	107.81	95.0	49706	2134	36.00	35.19	10.63
7 Hirn	Sept. 29, 1875.	†Superheated 220°	30.13	.4539	99.58	....	50255	1759	2.52	15.85	1.10
8 Hirn	Oct. 28, 1875.	‡Superheated 220°	30.00	.2867	78.30	....	43754	....	12.00	Dry.	Dry.
9 Corliss	1878.	Saturated Jacketed	50.41	$\frac{1}{11}$	105	92	....	1480	38.30	21.7	....
10 Corliss	1878.	Saturated Jacketed	51.12	$\frac{1}{8}$	137	125	....	1690	31.70	19.2	....
11 Corliss	1878.	Saturated Jacketed	49.34	$\frac{1}{6}$	138	142	....	1840	25.3	18.5	....

* Throttle valve partly closed. † Valve nearly closed. ‡ Non-condensing.

DISCUSSION.

Prof. R. H. Thurston.—During the last forty years there has been more time and thought and labor given to the study and experimental examination of the internal wastes of the steam-engine, and to its more complete thermal, as distinguished from the pure thermo-dynamic, theory, than to any other branch of the philosophy of the heat-engine. During this period the advance has been great, though hampered by the continual handicap of the old theory, and the lack of comprehension on the part of its originators and their disciples of the actual differences between the ideal and the actual case. The two differ very much in the same way, though far more widely, as do the old theory of ballistics in a vacuum and the modern theory in which air-resistance is involved, and which, therefore, represents the real and practical case. After Watt had discovered the facts which give rise to this difference, and Clark had measured the wastes in the locomotive, Hirn those of the mill engine, and Isherwood of the old form of marine engine, these differences between the old and the new treatment became obvious, as essential; and the contest between Hirn and his colleagues and Zeuner, now famous, settled all questions on this subject and placed the modern method of treatment firmly in its proper place. It is not likely, hereafter, to have opponents among well-read engineers or men of science.

Hirn's analysis of the operation of the steam-engine, as first practised by him and by Hallauer, and as later worked into a form, algebraically, more consonant with recent methods of treatment, especially by Dwelshauvers-Dery, endeavors to give measures of all the varying quantities of energy and of the state of the working fluid during a cycle of the engine, thus tracing the process of transfer and of transformation of heat throughout, and giving a clew to the nature and extent of those wastes by internal and somewhat obscure ways, the gross result of which is found by the experimental methods of earlier engineers. The outcome of this system is well shown by the paper of Prof. Carpenter. There still remain some problems to be solved, and some related methods to be sought out; but this, as it stands, is a most important advance upon the earlier method and theory. It is most satisfactory to find these latest facts and methods coming into the text-books and treatises on the steam-engine, and, especially into the practice of the technical schools and of engineers. Sinigaglia, who was, I think, the first, as a pupil of Dwelshauvers and Madamet, and others who have effected this advance, are entitled, as it seems to me, to great credit. Prof. Peabody has performed this service in this country; and his excellent work is the only one, so far as I know, in the English language, at present, to give the full account of Hirn's method. The late Prof. Chas. A. Smith, in his *Steam-using*, gives the best early translation of Hirn's and of Hallauer's papers.

I do not know how far the adaptation of the Hirn system, and the Dwelshauvers algebraic expression of the analysis, have been brought into the customary operations of the experimental departments of our colleges; but they have now been some time in the regular schedule of work and practice in the Sibley College laboratories, and have been found perfectly practicable and satisfactory in operation, and are most valuable as giving the student a good idea of the facts involved and the advances made of late years in the correct theory of the engine. Prof. Carpenter has worked it into good shape, and the methods and results are well illustrated in this paper. It has been applied, especially, to the small engine here described, but also to a number of cases in which important engine trials have been made; and all our "experimental engines" will be thus treated, including the large special construction now under way, and fitted for a most extensive range of variation of all conditions, kinematic, thermal, and other, thanks, in great degree, to the ingenuity and interest of Mr. Edwin Reynolds, who has given personal attention to the carrying out of our requirements in this flexibility of the machine. We hope to be able, after a time, to present some exceedingly interesting work, especially in the solution of hitherto unsolved, often unstated, problems.

One such problem is referred to in this paper: that which determines the quality of steam, in the various portions of the engine cycle, by direct measurement. Prof. Carpenter tells, in a general way, how this is done, and will, I hope, at some future time give a fuller account of apparatus, methods, and results of these investigations, which have great interest and importance, both intrinsically and as bearing upon the Zeuner-Hirn controversy, corroborating the latter authority in every essential matter. The device employed is most ingenious and exceedingly satisfactory; though I have no doubt that it will be found capable of improvement by its inventive author.

Prof. Cecil H. Peabody.—This method is of great interest, and must be considered an important step forward. Yet, while no one doubts the truth of the usual assumption of dry steam at compression, it cannot be admitted that this paper entirely disproves Zeuner's claim that a film of moisture may adhere to the cylinder wall at that time, for the existence of such a film is not impossible, though dry steam is received by the calorimeter. This method must, therefore, be considered as an important though an incomplete investigation of the subject, and I hope that the experimenter may later show that a film does not exist.

There has been some question in this paper and in others as to whether this method has been used in America. In preparing my work for students, it appears to be essential that I might have a few problems in which I had the entire data. For that reason we had a series of five experiments made upon a small Corliss engine in the laboratory with entire success some three years ago.

Prof. Geo. I. Alden.—Referring to the statement which Prof. Peabody has made in regard to Zeuner's theory that there is moisture upon the surface of the cylinder when the body of the steam in the cylinder is dry, I would like to call attention to the form of the compression line on the low-pressure card of a small compound engine (Fig. 312). This engine is described and illustrated in Mr. W. W. Bird's paper, which is in your hands, and which is to be presented.*

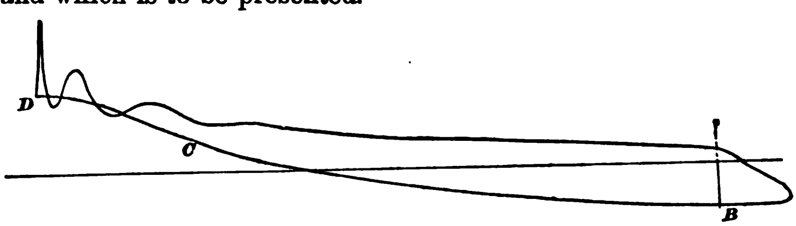


FIG. 312.

There is no cut-off in the low-pressure cylinder. Compression begins at *B*. The compression line *BCD* has a reversed curvature, the part *BC* being concave upward, and the part *CD* convex upward. It is also found that the compression line runs considerably above the theoretical compression line, constructed

* No. CCCLIII., page 873, Volume XII., *Transactions A. S. M. E.*

by computing the weight of steam compressed and drawing its adiabatic through B. It is further found that to account for the increased pressure which the actual card shows by the superheating due to compression and to heat received from the cylinder walls would require the steam to be at a higher temperature than that of the steam which enters the low-pressure cylinder.

It therefore seemed to me that there was moisture upon the cylinder surfaces which during compression from B to C was reëvaporated, thus increasing the volume of the steam undergoing compression. That near the point C the steam had risen to the temperature of the surrounding walls, and, therefore, the reëvaporation was diminished or ceased entirely, and the curve from C to D rose in pressure much less rapidly than before, causing the reversed curvature shown. My point is that the indicator card shown seems to confirm Prof. Zeuner's theory.

Mr. Geo. H. Barrus.—I have the gravest doubts whether the application of Hirn's analysis to engine testing, in the manner proposed by Prof. Carpenter and others, is advisable for purposes of instructing the students in thermo-dynamics. There is so much mathematical work about it that, in my opinion, the mind of the average student would be confused in the study of the question, and lose sight of the main facts which he might learn by the employment of the more simple methods which have heretofore been in vogue, and which are more easily comprehended. This, however, is a matter of opinion, and there is no need of dwelling upon it. My attention, in reading this paper, has been specially directed to that portion relating to the determination of the quality of the steam in the clearance spaces, which is the striking feature of the paper. I would first, however, call attention to certain errors or inaccuracies in the text, and to indications of careless work in the conduct of the experiments referred to. On page 803, at the bottom of the table marked "Form No. 3," under the head of "Mean Pressures, Head End," the symbol given is M.E.P., and the quantities in that column are the mean pressures measured from the line of no pressure, or perfect vacuum. It is commonly understood that the term "mean effective pressure" refers to the average pressure determined from that portion of the indicator card which is enclosed within the line forming the diagram. The pressures referred to in the column noted being measured from the zero line, are usually designated by the term "total pressure," and not "effective pressure." The symbol should be changed accordingly.

I notice on the eighth page that the diagrams which are reproduced are exceedingly uneven in form. They are also of different lengths, and I assume that one was taken with the indicator at one end of the cylinder, and the other with that at the other end. The difference in the lengths is about 5%. If these are given as samples of the diagrams taken during the tests, they show, to say the least, carelessness on the part of those who manipulated the indicators.

I have been trying to decipher what scale was used in plotting the various results of the tests on the chart given in Fig. 234. The average speed, for example, is put down in the table as 204.64. On the chart it lies between the 6.5 mark and the 7 mark. The average gauge pressure is given in the table as 38.35 lbs., and on the chart it runs between the 1.5 mark and the 3.5 mark.

In the table which is marked "Form 1," the average indication of the calorimeter in the steam-chest is put down as 214°. This temperature is so near the normal corresponding to the pressure that there is every indication that the steam issuing from the calorimeter was in a wet state, and that the instrument failed to show the real condition of the steam.

It will be seen by reference to Fig. 236, which shows the complete arrangement of the apparatus used on the test, that the indicator cords pass from the driving motion to the two indicators in a direction which appears to form a considerable angle with the motion of the cross-head, and that there is no pulley made use of to change the direction of the motion. While discussing this cut, another thing may be referred to, and that is the method of driving the releasing device for operating the whistle valve attached to the calorimeter. This appears to be done by the use of a cord attached to the indicator driving rig. It seems to me a device of very doubtful utility, considering the fact that this valve should be operated with such extreme precision. I should hardly think that a flexible connection, such as that shown, would answer at all.

Passing now to the main point of my discussion, I wish to call attention, in most emphatic terms, to what appears to me the utter unreliability of the tests which pretend to show, by means of the calorimeter, the actual quality of the steam in the clearance space of this engine. So far as I know, this experiment has not heretofore been attempted, and I wish to give Prof. Carpenter the credit of a high degree of enterprise in his efforts to determine, experimentally, some facts regarding this interesting question. So far, however, as getting any results from the work which throw any light upon the subject, I am more than convinced that he has utterly failed. He seems to doubt himself the reliability of his work in this line, for he says on page 798: "At the best there is some doubt about the accuracy of this determination of quality." Let us see now what the facts are which the experiments show: Assuming that the whistle valve, which introduced the steam into the calorimeter, was adjusted so as to open the communication during that part of the compression period of the stroke when the pressure was above the atmosphere (though, considering that the engine speed was 200 revolutions per minute, it is hardly possible, with the means employed, that this could have been done with the necessary precision), we have the calorimeter subjected to a pressure varying from zero to about 13 lbs. above the atmosphere, the average being about 6.5 lbs. above the atmosphere. Under this pressure I estimate (assuming that the calorimeter had an orifice ⅛ inch in diameter) that the quantity of steam which was used in the calorimeter amounted to not over 10 lbs. in weight per hour, provided the flow of steam through it was continuous. Considering that it was not continuous, but occupied only that portion of the time of revolution corresponding to the stated period of compression, which is about ⅙ of the revolution, the actual quantity of steam discharged through the calorimeter was at the rate of not over 2 lbs. in weight per hour. The indications of the calorimeter, in the summing up of the experiments, have not been corrected for radiation. In order to determine the quality of the steam indicated, it is necessary to make a correction for this loss. In my own calorimeter work, I have found that for 100° difference of temperature between the throttling part of the calorimeter and the air, the quantity of heat lost by radiation when 60 lbs. of steam per hour is passing through amounts to 3.5° temperature indicated by the thermometer in the wire-drawn steam. I have taken the difference of temperature for this instrument to be 200°, and on the basis of the radiation noted, the number of degrees correction, supposing 60 lbs. of steam were passing through per hour, would be 7°. Considering, now, that only 2 lbs. is passing per hour, the effect of radiation would be multiplied in the inverse proportion of these two quantities. That is, it would be multiplied thirty times, and, consequently, the correction to be applied for the loss by radiation is $7 \times 30 = 210°$. From the indication of the calorimeter thermometer, which was 258°, it appears that, without correcting for radiation (and omitting for brevity the calculation), the amount of superheating in the steam drawn from the cylinder which is shown is 33.5°. Adding the correction for radiation just found, we have, as the real showing of the calorimeter, a superheating of $33\frac{1}{2}° \times 210° = 243\frac{1}{2}°$. According to this indication, the temperature of the steam in the cylinder during the compression period in question (the normal corresponding to the average pressure being 231°), is $243\frac{1}{2}° \times 231° = 474\frac{1}{2}°$. If, therefore, we take the indications of the author's experiments for what they really show, we must conclude that the steam in the cylinder, during the period of compression which we have been considering, is at a temperature of 474.5°. It is simply ridiculous to believe anything of the kind, and Prof. Carpenter evidently had good reason for distrusting his experiments. In point of fact, the experiments simply show that the metal of the cylinder head was at a sufficiently high temperature to conduct the heat along the walls of the pipe connecting it to the calorimeter to such an extent that the temperature of the thermometer cup was raised to 258°. The indications of the calorimeter point in no way whatever to the quality of the steam in the clearance space, but simply show the temperature of the metal of the instrument which had been heated by contact with the hot cylinder.

Prof. Carpenter assumes that the quality of the steam in the clearance space is 100%, and by thermo-dynamic computations he makes out that at the beginning of compression the steam contained one-half of one per cent. of moisture, while at the end of compression it was superheated to an amount, erroneously stated to be 8.3°, but, in reality, according to his data, 81°. It seems to me that he has no right to make the assumption that the quality of the steam in the clearance space is 100%. The probabilities are that the quality of the steam at the beginning of the compression is the same as it is shown to be during the exhaust—that is, 73.3%. It could not be dry steam at the beginning of the compression, for in that case the quality of the steam at the beginning of the exhaust would have to be 46.6% in order that the average shown at the condenser should be that found, namely, 73.3%. This cannot be, for the reason that at the release the quality is nearly 20% higher than this, namely, 64.9%. In my opinion, the quality of the steam at the beginning of compression is 73.3%, or thereabouts, while that at the end of compression, having been improved by the heat received during the process of compression, is 88.5%.

Prof. Carpenter should either defend his experiments with data which carry more weight than any given in his paper, or he should eradicate from the paper all reference to the calorimeter work, which can do nothing more than encumber the Transactions of our Society and throw discredit upon them.

*Prof. R. C. Carpenter.**—In reply to the extended remarks by Mr. Geo. H. Barrus, it seems hardly necessary to take much valuable space. In the first place, very many of his remarks indicate that he has not even taken the trouble to read exhaustively the article which he so severely criticises, and, again, he has supposed the existence of conditions which did not exist, and which were not described because they had no bearing on the subject under discussion; but from these supposed conditions conclusions are drawn, which, as he shows, give in many cases absurd results.

Thus, for instance, he states that the connection to the calorimeter valve was probably by a string, yet on page 807 it distinctly states that *this connection was by a wire.* It is to be supposed that he assumes that Fig. 236 represents more than the text claims for it, whereas by reference to the eighteenth page he will see that it is merely intended to show the position and method of connection of the calorimeter. The apparent angularity of the indicator cords were due to the location of the camera and to the peculiar lens used, as in reality they were parallel and horizontal in use. As the test of itself was novel only in some particulars it seemed unnecessary to give a long and detailed description of methods employed in taking cards. A careful reading of the text would have shown that Fig. 231 and Fig. 230 are merely what they claim to be, diagrams showing the method of subdividing the card into parts, and do not belong to the specific test commencing on page 801. The

* Author's Closure.

indicator diagrams, Fig. 232 and Fig. 233, as described at the bottom of page 802, do belong to the test in question, and are not open to the objections which he states characterize the work of the test.

In regard to the calorimeter work it is hardly to be supposed that we should satisfy Mr. Barrus under the existing circumstances, but here again he presupposes conditions which had no existence. In the first place, he supposes the passage of heat along the iron to the thermometer in the calorimeter by a way which he does not specify and from a source of heat not mentioned, as in this case the highest temperature in the cylinder could not have been more than 8° or 10° in excess of the thermometer reading in a calorimeter nearly a foot away and subjected to more or less radiating influence. It is absurd to expect heat from this source sufficient to influence the thermometer, and there is no extraneous supply of heat. Again, his inference that the calorimeter determination of the entering steam was not reliable, because it was so near the boiling point, was evidently made without considering the altitude of Sibley College and the table of boiling points which was prefixed, and which was obtained from mercurial barometer readings. The barometer in question was itself a United States standard instrument and had been frequently compared with the one in use by the State Weather Service, as standard.

A manometer reading for back pressure on the calorimeter was taken, and no source of error can be imagined sufficient to overcome the difference of 3.3°; again, any error in that direction would tend to increase instead of to diminish the slight discrepancy between our calculated and the measured quantities at the end of the cycle.

It is very evident from the calculations made by the disputant that he has not only not read the paper carefully, but that he does not understand fully the principles of the analysis in question. This is clearly shown by his proposition to calculate the heat remaining in the cylinder from that carried away by the exhaust, and which bears evidently no relation whatever to that remaining in the cylinder at the beginning of compression; nor is it clear why in one place he tells us that our thermometer in the calorimeter reads too high, and must have received heat from an external source, and in another place figures out with equal gravity that it should have read much higher.

In conclusion, I can only say that I hope in his own work as an expert my critic is not so free to assume conditions or data which are not given by observation as in the discussion of the paper, and that his consequent conclusions are more reliable and are founded on a basis of facts and observation, rather than on some preconceived notion. In regard to the paper, I am satisfied that the methods outlined are correct, and that the various observations cited were made with care, and I am yet to be convinced of any serious errors in the calculation of the test cited. On the other hand, the close agreement of the actual calorimeter reading with the results that would be indicated by methods of measuring the heat remaining in the cylinder, as explained, emphatically show that our calorimeter readings are reliable, though no doubt subject to some correction yet to be determined.

CCCCLII.*

NOTES REGARDING CALORIMETERS.

BY R. C. CARPENTER, ITHACA, N. Y.

The subject of calorimetry is considered one of considerable importance in the experimental laboratory connected with Sibley College. It forms one of the regular exercises which students are all required to take, and the laboratory is equipped with nearly all of the forms which have been described in early papers before the society and in Thurston's "Steam Engine and Boiler Trials," besides possessing a number of new forms, some of which are under trial and perhaps not worth a description, while a few are quite promising. The following paper contains a discussion of some experiments regarding the best methods of obtaining steam for calorimetric use, a proposed general classification of calorimeters, and a general discussion of some of the various classes, also a short description of one or two new forms of calorimeters; but the principal portion of the paper is devoted to the throttling calorimeter and some rapid methods of reducing the determinations, for which purpose some diagrams and tables are given.

THERMOMETER CUPS.

The thermometer cups used at Sibley College screw into a ¾-inch pipe fitting, and are of the form and dimensions shown in the cut—Fig. 239. These are made from a solid piece of iron or brass. The liquid for filling these cups is a matter of considerable importance; we have used either cylinder oil or mercury. For most purposes there is no particular difference, but we find that cylinder oil is not quite so sensitive to changes of temperature, and is very sensitive to the presence of moisture. The effect of moisture in a cup filled with cylinder oil is, if the temperature is high, an explosion, which is likely to blow the oil out from the cup, or if the temperature is not high enough for this, the gradual

---

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. *Transactions.*

evaporation of the moisture. This latter will sometimes keep the reading of a thermometer suspended in the oil cup at the boiling point of water regardless of the temperature surrounding the cup. With suitable precaution cylinder oil gives satisfactory results. The use of thermometers, screwed directly into the steam-pipe has not been practised except in throttling calorimeters when pressure is very low. In that case we have tried a thermometer cup, made of wire gauze, held in place by a perforated rubber cork. The perforation in the cork fits the thermometer tube so closely as to prevent leakage.

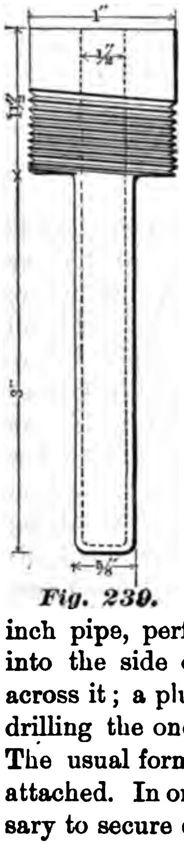


Fig. 239.

This form of throttling calorimeters is described later. From one or two experiments, the results seem to be the same as with the ordinary cup, Fig. 239.

METHODS OF OBTAINING A FAIR SAMPLE OF STEAM.

The usual method of obtaining a sample of steam for calorimetric determination, adopted by the laboratory, consists in the use of a nipple of one-half inch pipe, perforated with one-fourth inch holes and screwed into the side of the main steam-pipe, so as to extend nearly across it; a plug is placed in the open end, and care is taken in drilling the one-fourth inch holes that they do not come in line. The usual form is shown in Fig. 240, with throttling calorimeter attached. In order to ascertain whether this precaution was necessary to secure dry steam, the following experiments were made:

On the same piece of horizontal pipe, three inches in diameter, three attachments for calorimeters were placed, which are numbered respectively No. 2, No. 3, No. 4, and are shown in Fig. 241. No. 2 and No. 4 were of the usual forms, as already described. No. 3 was varied from time to time and is described in each case. In the first trial No. 3 connection consisted of a bent nipple without perforation, arranged in such a manner that the nozzle could be turned in any direction with relation to the steam current. See Fig. 242. The amount of moisture was determined in each case by throttling calorimeters, placed on the main pipe in essentially the same manner and as shown in Fig. 241. The distance between these calorimeters was about one and one-half feet.

During the test a current of steam was kept passing through the calorimeter by leaving the drip valve of the main pipe, as shown in Fig. 242, partially open.

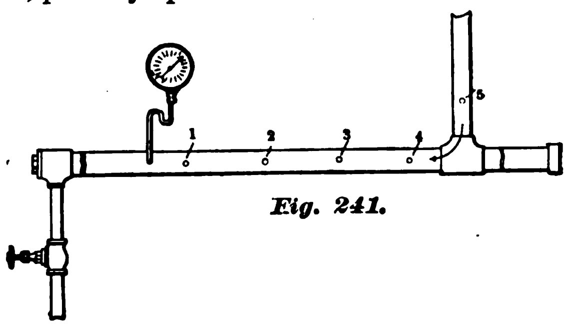


Fig. 241.

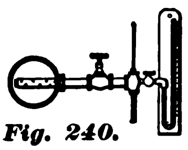


Fig. 240.

A manometer for measuring the back pressure was attached to each calorimeter.

TRIAL No. 1.

Standard connections for Calorimeter No. 2 and Calorimeter No. 4.

Nozzle of No. 3 turned upward as in Fig. 243.

Instruments used:—Steam Gauge, calibrated and adjusted; Thermometers, in No. 2, 5260; in No. 3, 5255; in No. 4, 5258; by H. W. Greene; calibrated for boiling point and found correct.

OBSERVERS, MESSRS. HEILMAN AND CLARKE.

No.	Time.	Gauge.	Calorimeter No. 2.			Calorimeter No. 3.			Calorimeter No. 4.		
			Tempt.	Manometer. Inches.	Superheat.	Tempt.	Manometer. Inches.	Superheat.	Tempt.	Manometer. Inches.	Superheat.
1	3.0	62	246	.4	34	245	.8	33	245		28
2	3.01	61.5	246½		34½	245.5		32.5	245		28
3	3.02	61	246.5		34½	245.25	.8	32.2	245.25	4.8	28.4
4	3.03	60	247		35	245		32	245	4.6	29
5	3.04	59.5	246.5	.45	34	244.75		31.75	245		29
6	3.06	58.5	244.75		32	245		32	245		29
7	3.07	59	244.75		33	245		32	244.5		29.5
8	3.08	59.5	244.75		32	244.75		31.75	244		29
9	3.09	61.3	244.25		32	244.25	.8	31.25	244		28
10	3.10	61.5	244.25	.45	32	244.25		31.25	245	5.	27

Per cent. of moisture shown by Calorimeter No. 2 equal 1.66.
" " " " No. 3 " 1.56.
" " " " No. 4 " 1.77.
Average moisture No. 2 and No. 4, 1.71 per cent.

53

TRIAL No. 2.

Nozzle of No. 3 turned horizontally facing the current of steam.
Calorimeter No. 2 and No. 4 as before.

OBSERVERS, MESSRS. HEILMAN AND CLARKE.

No.	Time.	Gauge.	Calorimeter No. 2. Tempt.	Calorimeter No. 2. Manometer. Inches.	Calorimeter No. 2. Superheat.	Calorimeter No. 3. Tempt.	Calorimeter No. 3. Manometer. Inches.	Calorimeter No. 3. Superheat.	Calorimeter No. 4. Tempt.	Calorimeter No. 4. Manometer. Inches.	Calorimeter No. 4. Superheat.
1	8.28	61	251.25		39¼	227		15¼	244		27
2	29	60.5	251		39	227		15¼	244		27
3	30	60	251	.45	39	227	.2	15¼	244	4.55	27
4	31	60	251		39	227.25		15¼	244		27
5	32	60	250		38	227	.2	15¼	244		27
6	33	60	250	.45	38	227.25		16	243.5	4.6	26.5
7	34	60.25	250		38	229		17¼	243		26
8	35	60.25	250		38	229.25	.2	15¼	243		26
9	36	61	250.5	.45	38.5	230		18¼	243		26
10	37	61	250		38	230	.2	18	243.5	4.7	26

Per cent. of moisture of steam by Calorimeter No. 2 equal 1.20.
" " " " No. 3 " 2.25.
" " " " No. 4 " 1.85.
" " " average No. 2 and No. 4 equal 1.52.
" " " excess in No. 3 above average, 0.73.

A subsequent trial under a little different conditions gives a result of only .25 of one per cent. difference.

TRIAL No. 3.

Nozzle of No. 3 turned toward bottom of pipe. See Fig. 242.
Other calorimeters as before.

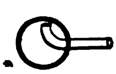


Fig. 242.

OBSERVERS, MESSRS. HEILMAN AND CLARKE.

No.	Time.	Gauge.	Calorimeter No. 2. Tempt.	Calorimeter No. 2. Manometer. Inches.	Calorimeter No. 2. Superheat.	Calorimeter No. 3. Tempt.	Calorimeter No. 3. Manometer. Inches.	Calorimeter No. 3. Superheat.	Calorimeter No. 4. Tempt.	Calorimeter No. 4. Manometer. Inches.	Calorimeter No. 4. Superheat.
1	8.44	60.5	251.25	.45	38.25	231	.2	19	244	4.35	27
2	45	58.5	251.25		38.25	235		23	244		27
3	46	58.0	251.25		38.25	235.5		23.5	244		27
4	47	57.75	251		38	236	.17	24	244		27
5	48	56.75	250.5		37.5	237		25	243		26.5
6	49	55.50	250		37	237.25		25.25	242.5	3.85	26.0
7	50	55	249	.35	36	237.5		25.5	242		25.5
8	51	56	248.5		35.5	238		26	241.5		24.5
9	52	56.25	248.25		35.25	238		26	241		24
10	53	56.75	248	.4	35.0	238	.19	26	241.5	4.3	24

Per cent. of moisture of steam by Calorimeter No. 2 equal 1.5.
" " " " No. 3 " 1.9.
" " " " No. 4 " 1.8.
" " " average of No. 2 and No. 4 equal 1.65 per cent.

Excess of No. 3 above average, .25%.

TRIAL No. 4.

Nozzle of No. 3 turned horizontally away from current of steam.

OBSERVERS, MESSRS. HEILMAN AND CLARKE.

No.	Time.	Gauge.	Calorimeter No. 2. Tempt.	Manometer. Inches.	Superheat.	Calorimeter No. 3. Tempt.	Manometer. Inches.	Superheat.	Calorimeter No. 4. Tempt.	Manometer. Inches.	Superheat.
1	4.52	56.5	247.5		35.5	223.5		11.5	243		26
2	53	56.0	247.5		35.5	223.5	.4	11.5	243	3.85	26
3	54	55.5	247	.4	35	223.5		11.5	243		26
4	55	55.0	247.5		35.5	223.5		11.5	243		26
5	56	55.0	247.25	.35	35.25	223.5	.4	11.5	243	3.75	26
6	57	56.0	247		35	223.5		11.5	243		26
7	58	57.5	247		35	223		11	243		26
8	59	59.0	247		35	223		11	243		26
9	5.00	60.0	247	.45	35	223	.4	11	243	4.50	25
10	5.01	60.5	247		35	223		11	243		25

Per cent. of moisture in steam by Calorimeter No. 2 equal 1.37.
" " " " No. 3 " 2.57.
" " " " No. 4 " 1.70.
" " " " No. 2 and No. 4, average 1.53.
Excess of moisture in No. 3 above average, 1.04 per cent.

TRIAL No. 5.

Calorimeter No. 3, attached to a nipple without holes, extending nearly across the main steam-pipe.
No. 2 and No. 4 as before.

OBSERVERS, MESSRS. HEILMAN AND CLARKE.

No.	Time.	Gauge.	Calorimeter No. 2. Tempt.	Manometer. Inches.	Superheat.	Calorimeter No. 3. Tempt.	Manometer. Inches.	Superheat.	Calorimeter No. 4. Tempt.	Manometer. Inches.	Superheat.
1	5.29	48	238.25	.25	24.25	227	.5	15	237	3.1	21
2	30	48.5	239		27	229		17	237		21
3	31	49	239		27	230		18	238		22
4	32	49	240	.35	28	231	.5	19	238	3.2	22
5	33	49.5	240		28	231		19	239		23
6	34	50	241.5		29.5	222		20	239.5		23.5
7	35	50.5	241	.3	29	233	.5	21	240	3.35	24
8	36	51	241.5		29.5	233.5		21.5	240		24
9	37	51.5	241.5	.25	29.5	234		22	241		25
10	38	52	242	.2	29	235	.5	23	241	3.45	25

Per cent. of moisture shown by Calorimeter No. 2 equal 1.35.
" " " " No. 3 " 1.85.
" " " " No. 4 " 1.60.
Average per cent. of moisture in No. 2 and No. 4, 1.47.
Excess of moisture in No. 3 above average, .38 per cent.

These trials seem to show that the difference between the quality of steam taken from the interior and that taken from near the surface of the main steam-pipe, may differ an amount sufficient to cause an error of one half of one per cent. Calorimeter No. 4 gave a reading ten to fifteen degrees less than No. 2 during the whole trial, which would indicate that the amount of moisture in the steam increased with the distance from the drip pipe.

FAILURE OF CALORIMETERS.

The investigations just cited show a great difference in the quality of steam existing in different parts of the same pipe, but do not nor cannot determine what system of connection will give an average sample of steam.

It seems quite probable that, in obtaining steam for calorimet-

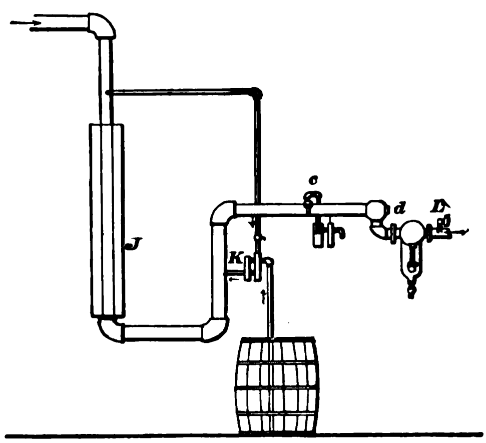


Fig. 243.

ric determinations, the sample is more likely to be dryer than the average, rather than wetter, and the resulting observations consequently show steam too dry. This is emphasized by the following experiment, which would also seem to indicate, that if for any reason a large amount of water is thrown into the pipe, a very large portion of it will be precipitated, and will run along the bottom and sides of the pipe as water, and will even be lifted up quite a distance vertically, yet not remain in the steam in the state of vapor.

In the experiment referred to, Messrs. Brill and Meeker were measuring the efficiency of various mechanical separators for steam-engines, and it was quite necessary to have very wet steam which could be varied in quality through a wide range.

The first attempt to make the steam wet was the addition of a jacket which could be filled, at varying heights, with cold water; the calorimeter attached at *C*, as shown in the illustration (Fig. 243), showed constantly about 4% of moisture, no matter what height of water was maintained in the jacket.

An injector was then attached, so as to draw water from a barrel and deliver it in a constant stream at *K*; the result was the same as before, although a steady stream of water was passing out from the separator, which by a rough calculation must have been at least 60% in weight of the steam.

In order to get more moisture in the calorimeter sample, a connection was made to the vertical pipe, at *d*, by means of which the moisture in the sample was increased from 4% to 40%, in some cases, but it was concluded necessary to condense all the steam to be certain of its quality. It was also concluded that a fair sample could not be selected by a perforated nipple.

CLASSIFICATION AND GENERAL DISCUSSION OF ERRORS FROM USE OF CALORIMETERS.

The following classification has been found very convenient in the study of calorimeters, and it is presented as a means of showing readily the principle or method of operation of the principal instruments that have been used.

CLASSIFICATION OF CALORIMETERS.

The methods which have been employed to obtain the amount of moisture in steam may be divided into three general classes: first, calorimetry proper, in which the moisture is determined by comparing the heat in one pound of the mixture with that given for one pound of dry saturated steam in standard steam tables; second, mechanical methods of precipitating and removing a portion or all the water; third, chemical methods of ascertaining the moisture in steam by the proportionate amount of some chemical solution in a given weight of steam, compared with that in the same weight of water from the boiler.

In the first class the heat in one pound of steam is measured, either by condensing the steam and finding the heat by its capacity to heat a known amount of water, or by superheating it, which makes it possible to find the amount of heat expressed in

British Thermal Units (B. T. U.) per pound of the steam. These classes can be farther subdivided, as shown below :

- Calorimeters........
  - Condensing..........
    - Jet Condenser........
      - Hirn's Barrel or Tank.
      - Injector—continuous.
    - Surface Condenser...
      - Barrus—continuous.
      - Coil—continuous.
      - Hoadly Calorimeter.
      - Kent—Tank Calorimeter.
  - Superheating..........................
    - External—Barrus Superheating.
    - " Universal.
    - Internal—Peabody Throttling.
- Direct determination of moisture...
  - Separator.
  - Chemical.

DISCUSSION OF ERRORS IN THE USE OF CONDENSING CALORIMETERS.

One equation may be used to represent the quality of steam for all cases of condensing calorimeters. Thus let x represent the quality of steam or per cent. of dry steam in the mixture, W the weight of condensing water, w the weight of condensed steam. Let q represent the heat of the liquid, L the latent heat of evaporation, T the temperature of the steam due to pressure as obtained from steam tables. Let t' represent the temperature of the injection water, or of the unheated condensing water. Let t represent the temperature of the discharge or heated condensing water. Then, since the heat lost by one pound of the steam is equal to that gained by $W \div w$ pounds of the condensing water, we have for the quality of the steam $x = \dfrac{\frac{W}{w}(t - t') - (T - t)}{L}$.

The effect of minute errors in any one quantity can be readily observed by differentiating with respect to that quantity considered as a variable.*

Thus,

$$\frac{\Delta x}{\Delta W} = \frac{(t - t')}{wL} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (1)$$

Relative effect of minute error in condensing water.

$$\frac{\Delta x}{\Delta w} = \frac{-W}{w^2}\frac{(t - t')}{L} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (2)$$

Relative effect of minute error in condensed steam.

* An excellent and full discussion of the effect and relation of these errors and their respective weights is to be found in *The Transactions of the Society of American Mechanical Engineers*, Vol. X., page 874, by Prof. Denton.

$$\frac{\Delta x}{\Delta t} = \frac{1}{L}\left(\frac{W}{w} + 1\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Relative effect of minute error in temperature of warm water.

$$\frac{\Delta x}{\Delta t'} = \frac{-W}{wL} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Relative effect of minute error in temperature of cold water.

With respect to L and T,

$$dx = \frac{1}{L^2}\left[-\frac{W}{w}(t - t') + T - t\right]dL - \frac{dT}{L}.$$

Since $dL = -dT$, nearly, for working pressures of steam, and further, is a function of the pressure, which we will represent by P, approximately, $dP = dT = -dL$, the error being small for working pressures of steam.

Hence we have the approximate expression,

$$\frac{\Delta x}{\Delta P} = -\frac{1}{L^2}\left[-\frac{W}{w}(t - t') + T + L - t\right] \quad . \quad . \quad . \quad (5)$$

Relative effect of error of steam pressure.

Assuming values of

$$W = 360, \quad t = 110°, \quad t' = 50, \quad w = 20,$$
$$P = 88 \text{ pounds absolute}, \; L = 890, \quad T = 318,$$

we shall have the following results by substituting in Formula (1)

$$\Delta x = \frac{60}{20 \times 890}\Delta W = \frac{1}{297}\Delta w.$$

Suppose ΔW, 1% or 3.6 pounds,

$$\Delta x = 1.2\%.$$

Treating the other formulas in a similar way, we can tabulate the results as follows, thus showing the error in quality due to a single error, in the data as given:

TABULATION OF ERRORS.

In Condensing Water. *Error.*		In Condensed Steam. *Error.*		Tempt. = 50° F. In Tempt. Cold Water. *Error.*		Tempt. = 110° F. In Tempt. Warm Water. *Error.*		Pr. = 88 lbs. Steam Pressure. *Error.*		Resulting Error in the Quality. Per Cent.
Lbs.	Per ct.	Lbs.	Per ct.	Degs.	Per ct.	Degs.	Per ct.	Lbs.	Per ct	
Total wt.	= 360 lbs.	Total wt.	= 20 lbs.							
3.6	1.0	0.2	1.0	.53	1.2	.65	0.60	7.0	8.0	1.2
1.8	0.5	0.1	0.5	.27	0.6	.30	0.30	3.5	4.0	0.6
1.5	0.40	.08	0.4	.18	0.5	.25	2.5	3.0	3.5	0.5
0.3	0.08	.016	0.08	.045	0.1	.05	0.50	0.6	0.7	0.1
Total wt.	= 300 lbs.	Total wt.	= 20 lbs.							
1.5	0.5	0.1	0.5	0.2		.25		2.2		0.5

The weights of water usually taken with the barrel calorimeter are from 300 to 400 lbs., while the weight of the steam condensed varies from 16 to 20 lbs., and the corresponding temperatures have a range of 50° to 70°. For these cases it will be found that the percentage of error in quality, supposing other data correct, is approximately the same as the percentage of error in the weights, and the limits of accuracy in the result would be determined by the percentage of error of weighing. The error in thermometer determination has nearly the same effect, whether made before or after the water has been condensed, and, for the weights usually used, the error of one-fifth degree in temperature has about the same effect as one-half of one per cent. error in weight, with the resulting effect of making an error of about the same amount expressed in percentage in the quality of steam.

The practical deductions from this analysis are that the best determinations require very accurate instruments, and that scales weighing the condensed steam must read accurately to twentieths of one pound, and thermometers must give accurate temperatures to one-fifth degree to make the result from single observations correct within one-half of one per cent.

Furthermore, it is not probable that, even with more accurate instruments, the resulting errors of observation, which include loss of heat by the calorimeter, can be made less than those quoted, and it is hardly probable but that errors of observation are likely to amount to as much as one-half per cent., with the apparatus in the form actually used in most boiler trials.

Determinations made with the best apparatus and with the favorable surroundings of a laboratory, often produce one or two discordant and impossible results in each series. For instance, in a series which generally shows moist steam of a uniform quality, one or two observations often show superheated steam as in the case submitted, which was made with great care, and with accurate instruments.

This superheating can only be accounted for by an error in weight of the condensed steam, or by an error in the temperature. The error in temperature would probably tend to make the per cent. of moisture too great—it would seldom produce the converse effect; the error in weight is often caused by the withdrawal of a couple of ounces of water with the hose that was used to supply the steam, for the reason that such an error would produce that effect and is very likely to occur in spite of precautions that may be taken. To counteract that effect the writer has recently used an iron pipe with a hose connection, and an air cock in-

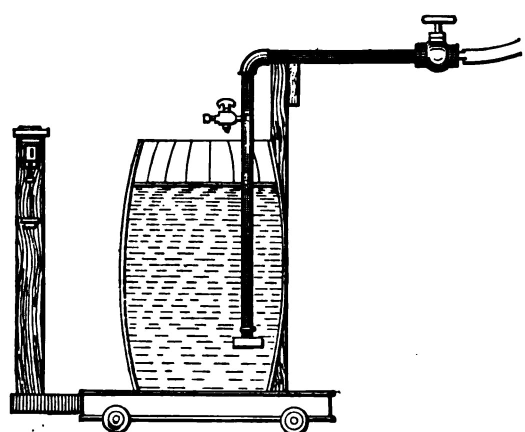


Fig. 244.

serted above the water line, which is not removed from the barrel during the operation of weighing. (Fig. 244.) In this case the weight must be corrected for displacement of water by the pipe.

THE INJECTOR CONTINUOUS CALORIMETER.

Continuous calorimeters of the condensing type seem to present the advantage over the barrel calorimeter in general, of better opportunities for measuring the temperatures of the condensing water before and after condensation of steam has taken place; while in some instances the separate weighings of the condensed steam and condensing water give an opportunity for greater accuracy, since the error in the result is for a given error in weighing, nearly inversely as the weights. A continuous calorimeter which is simple in form has been used in the laboratory to a considerable extent the past winter, and, although it presents in theory many objectionable features, gave us in practice quite uniform and satisfactory results. From the fact that the instrument used in most of the observations was a small injector, I have given it the name of the Injector Calorimeter. The method of connecting and of using is shown clearly in Fig. 245. *M* is the main steam-pipe, *P* a steam-gauge, *S* is the pipe leading steam to the calorimeter, *B* is a tank of cold water on scales from which the supply

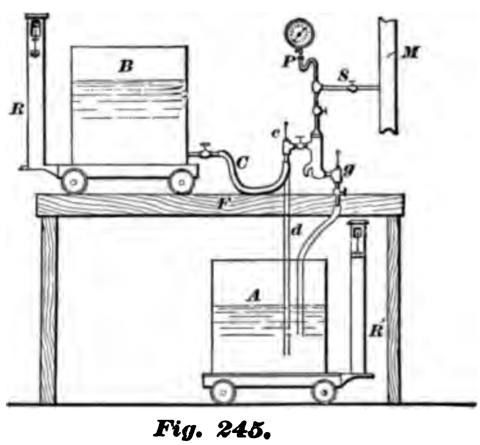


Fig. 245.

is taken, *A* is a receiving tank on a pair of scales. The temperature of the injection water is measured at *e*, and of the discharge water by a thermometer at *g*. The tank *B* was set so as to deliver water to the injector with little or no head, so as to obviate any correction for external work. The steam was regulated, so that the temperature of discharge was kept about constant at 110° F. In using the calorimeter, it is first started, and after reaching a constant working condition the poise on the scales *R* is set at a given point, and the instant the scale beam rises, the discharge water is transferred to the scale R^1, the reading of which had previously been taken.

The run was terminated in the same way, without interfering with the operation of the instrument. A design for the same instrument made of pipe fittings is shown in Fig. 246.

The use of two scales of equal accuracy is required by this method, and constitutes a serious objection to this form of calorimeter; but the range of temperatures can be maintained very uniform, and if the discharge temperature is maintained about as much higher than the temperature of the room as the injection water is below, the correction for radiation may be neglected.

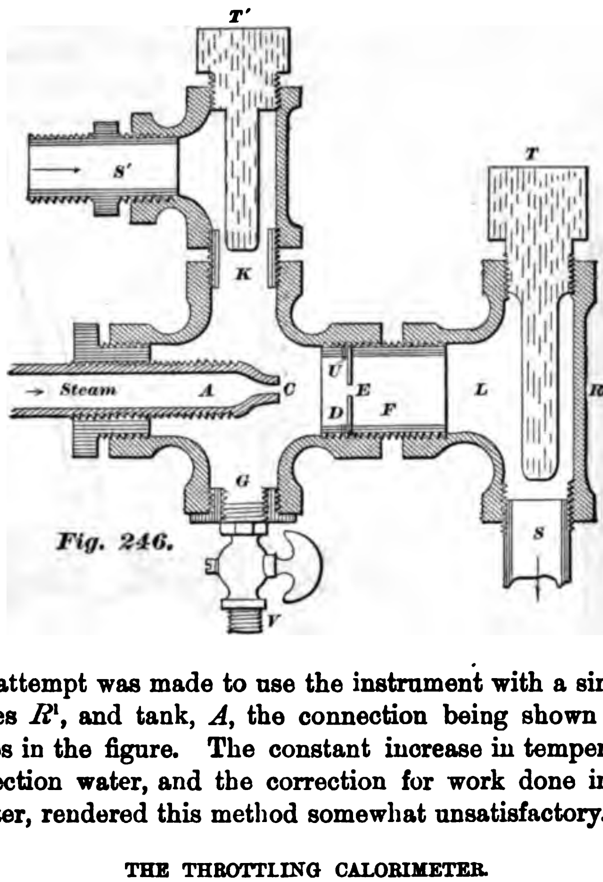


Fig. 246.

The attempt was made to use the instrument with a single pair of scales R^1, and tank, A, the connection being shown by dotted lines in the figure. The constant increase in temperature of the injection water, and the correction for work done in raising the water, rendered this method somewhat unsatisfactory.

THE THROTTLING CALORIMETER.

We are indebted to Prof. Peabody for this calorimeter, the equation for its use, and the first discussion of its results.* In the use of this instrument in the laboratory, Mr. C. L. Heisler

* Attention has been called to the fact that Patent 401,111, dated April 9, 1889, and filed in Nov., 1888, in Claim IV., may cover the throttling calorimeter as described by Prof. Peabody in October, 1888, Vol. X., page 237, and also the Heisler calorimeter.

and Mr. C. Hicks, in 1889, confirmed, experimentally, Peabody's statement that in a general way the sensitiveness and accuracy of this instrument were increased as its size was diminished. This led to the design of the Heisler calorimeter. This calorimeter has a total length of 14 inches and a total weight

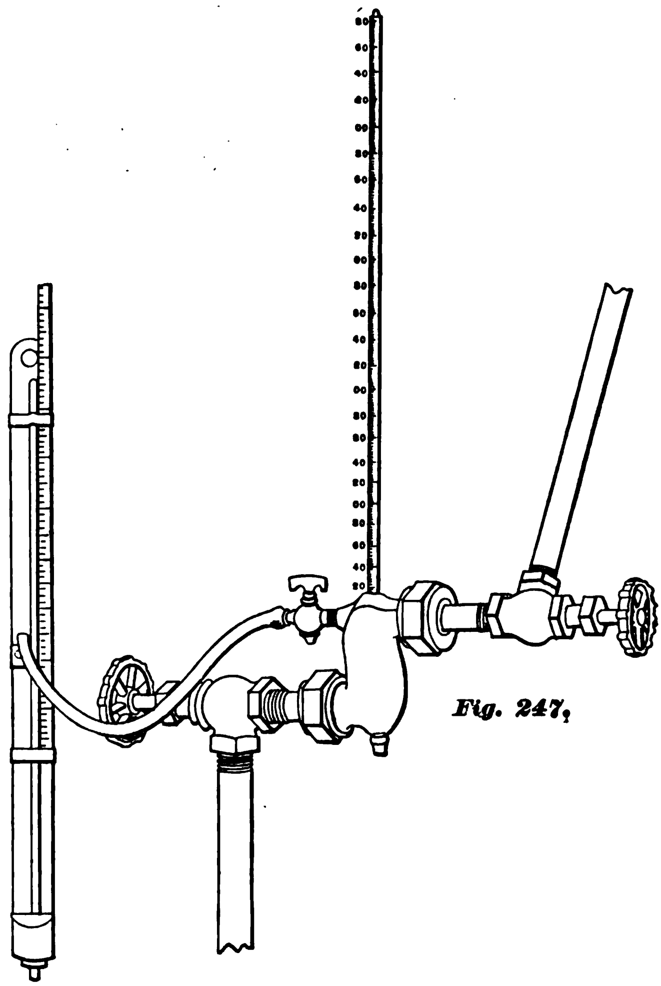


Fig. 247.

of $4\frac{1}{2}$ pounds, including valves. It is constructed ready for attachment to half-inch pipe, and as shown in the following cuts. (Figs. 247 and 248.)

This instrument consists essentially of a small angle globe valve secured to the swivel union, which holds the non-conducting washers and removable throttling disk *directly* against the wall of the S-shaped chamber enclosing the copper thermometer tube. At the lower opening of this chamber is also a swivel-connected angle discharge valve, the conductivity of heat to the discharge pipe being prevented by the washers in the union; the conduction to and from the apparatus being thus reduced to a minimum.

An examination of the peculiar interior construction will show that the steam from the throttling disk impinges directly, at right angles, against the thermometer tube; the steam must cross the plane of the thermometer tube three times before finding its exit from the chamber, thus securing a good circulation. The weight

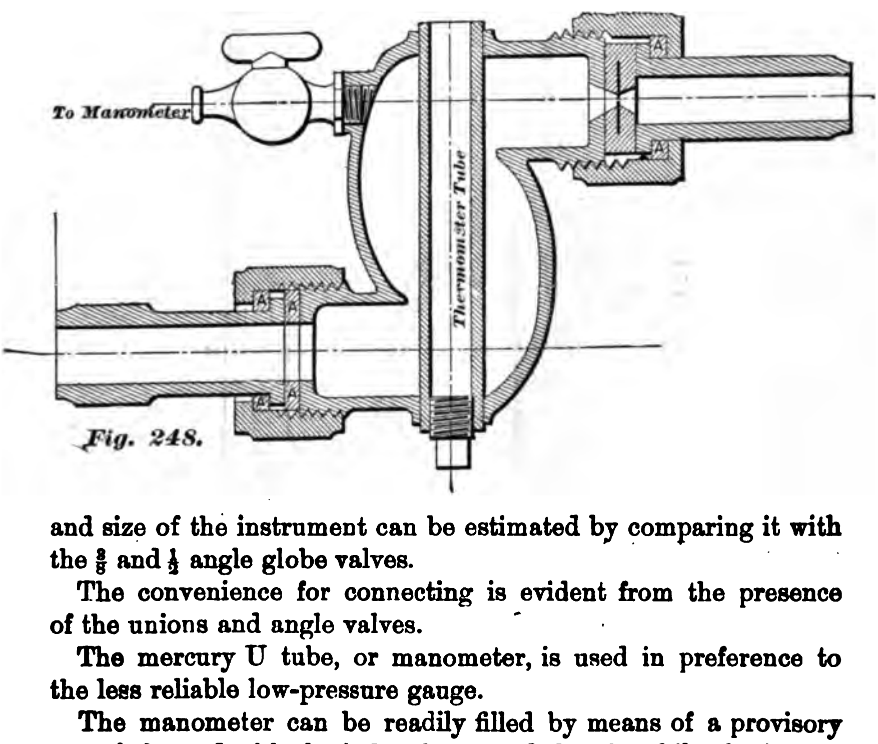


Fig. 248.

and size of the instrument can be estimated by comparing it with the ⅜ and ½ angle globe valves.

The convenience for connecting is evident from the presence of the unions and angle valves.

The mercury U tube, or manometer, is used in preference to the less reliable low-pressure gauge.

The manometer can be readily filled by means of a provisory tunnel, formed with the index finger and thumb, while the instrument is held in an inclined position within a pasteboard box, or other suitable receptacle, to avoid a waste of mercury. The manometer should be placed, with respect to the position of the calorimeter, about as shown in the cut, thus avoiding the slight possible error due to the head of the water collecting in the rubber tube connection. The small thumb cock must be opened cautiously to prevent the danger of blowing the mercury from the tube.

The form of the Peabody Calorimeter shown in the following cut (Fig. 249) is also in use in the laboratory, and was designed by the writer. It has the merit of cheapness and portability. Its cost without valves or thermometer cup need hardly exceed 50 cents, and its weight without valves is 2 pounds. It is constructed as follows, out of gas pipe and fittings: Using a tee for three-quarter inch pipe, screw into the top a thermometer plug; put a bushing into the side, opening into which screw a three-eighth inch nipple, the end of the nipple being capped at *O*, and in which

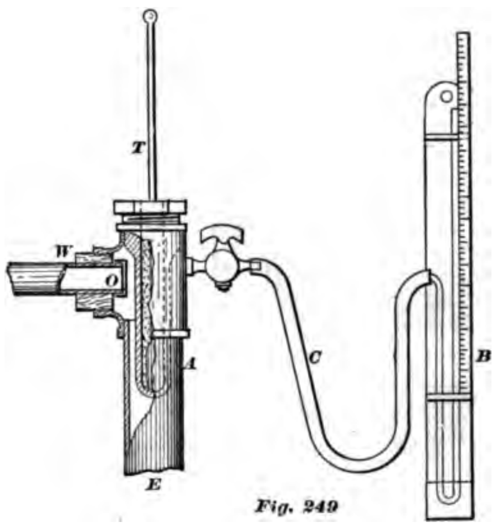


Fig. 249

a hole one-eighth inch in diameter is drilled and reamed out so as to present a thin edge to the incoming steam and make an orifice of known area. A one-eighth inch air cock is inserted in the tee somewhat out of line of the inflowing steam, to which can be connected the tube *C* of a mercury manometer, *B*, for measuring the pressure in the calorimeter when desired. The steam may be exhausted directly into the air through the nipple *E*, or a hose may be attached, and the steam condensed.

The thermometer cup may be omitted and the thermometer held in place by a perforated rubber cork, pressed into a top bushing. In this case no wooden or other insulating device to prevent the transmission of heat from the high-pressure steam to the calorimeter is used.

The construction is the same as shown in Fig. 249, except that the thermometer cup is omitted, and to steady the lower portion of the thermometer a basket of wire gauze is inserted and held in place by the outward pressure of the rubber cork.

The use of these instruments does not differ in any particular from that described by Prof. Peabody, in paper, page 327, Vol. X., for one of different form. The method is substantially as follows: Connect the calorimeter so that the sample of steam shall first pass through the small orifice and expand in the calorimeter to a pressure shown by the attached manometer. Take the temperature of the steam in the calorimeter, which should exceed the normal temperature due to that pressure; if there is no superheat in the calorimeter the instrument is not operative, due to an amount of water exceeding the limits of the calorimeter; if the steam flows directly from the calorimeter into the air, a manometer on the calorimeter is generally unnecessary, as the pressure will not sensibly exceed that of the atmosphere; a connecting pipe or hose for the exhaust rarely gives a back pressure exceeding one-half inch of mercury, which difference will make only about one thermal unit in difference of temperature, which is an almost unappreciable amount when reduced to percentage of moisture.

The equation for the use of the throttling calorimeter can readily be reduced to a form which, for the conditions of ordinary use, are easily tabulated or represented graphically, so that the results may be taken directly from a table or diagram without previous calculation. Thus, if we represent the quality of the steam by x, the latent heat of the steam in main steam-pipe by L, the heat of the liquid by q, the total heat of steam due to pressure in the calorimeter by H, the specific heat of steam by k, the reading of the thermometer in the calorimeter by T, and the normal temperature of the steam at calorimeter pressure by t, we have, supposing the expansion to occur without gain or loss of heat, $xL + q = H + k\ [T-t]$.

If the pressure in the calorimeter is atmospheric or nearly so H and t will be constant and have a value due to atmospheric pressure; that is, H will equal 1146.6 and t will equal 212. In such a case the value of x is the equation of a straight line corresponding to the ordinate, and $T-t$ is the variable corresponding to the abscissa, $\frac{k}{L}$ is the tangent of the angle made with the axis of abscissa, and $\frac{H-q}{L}$ is the point where the line cuts the axis of ordinates. The equation being $x = \frac{k\,[T-t] + H - q.}{L}$ The equation for the per cent. of moisture $1-x$ is also a straight line. Since $1-x = \{L - [H - q + K(T-t)]\} \div L.$ Since L diminishes with the steam pressure, the lines representing the different equa-

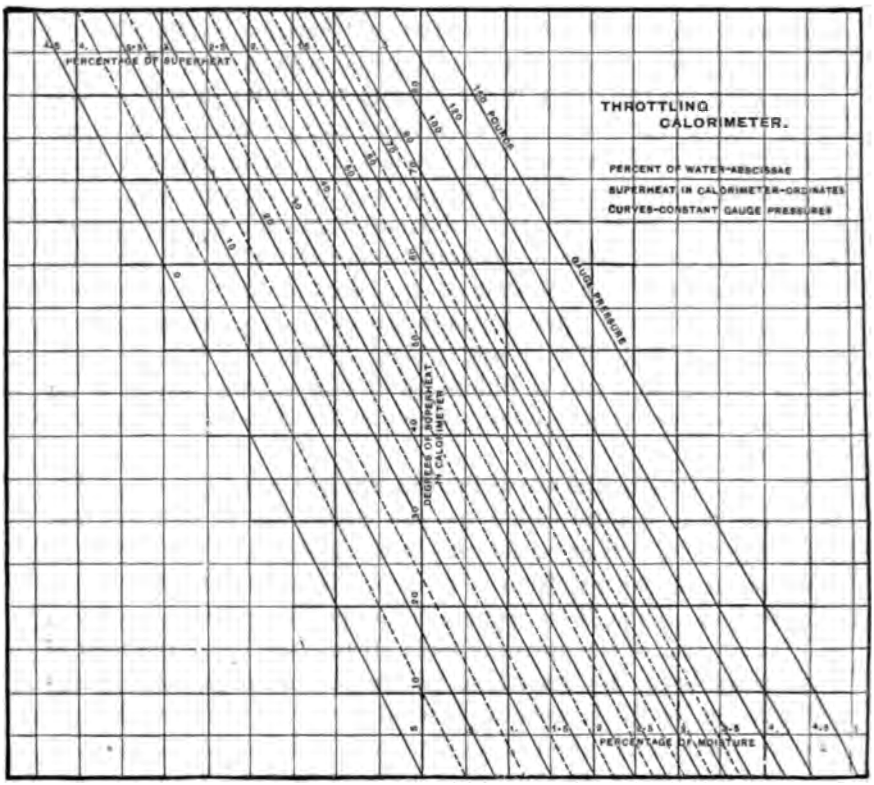


FIG. 250.

tions for pressures will not be quite parallel, although nearly so. The accompanying chart (Fig. 250) is a graphic representation of these equations for the usual range of steam pressures, and on which the per cent. of moisture can be read to one-fourth of one per cent., the diagram shows the per cent. of moisture or $1-x$, as abscissa the amount of superheat in the calorimeter, or $T-t$ as ordinates, and the steam pressure, absolute, by the diagonal lines. The chart will save considerable labor in reducing the

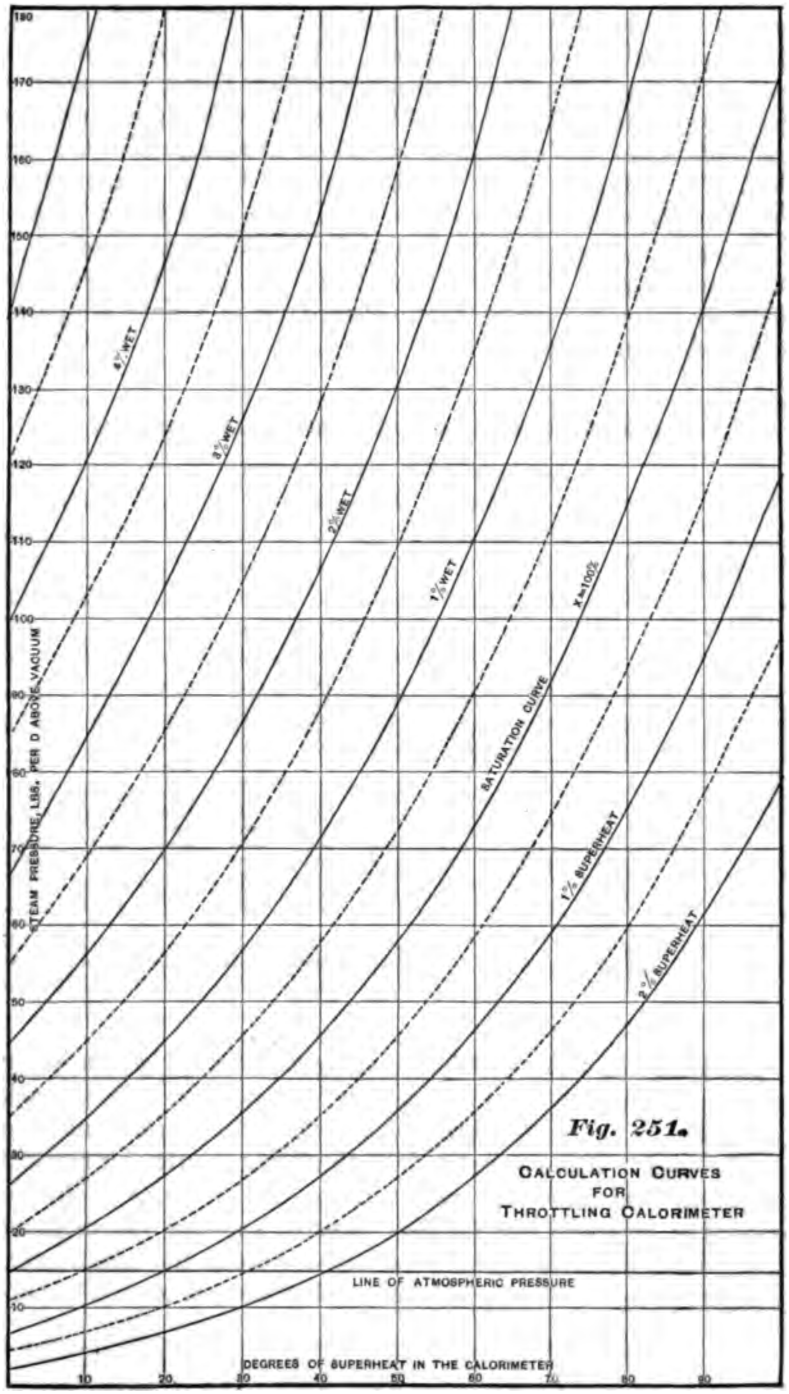


Fig. 251.

CALCULATION CURVES
FOR
THROTTLING CALORIMETER

results of determinations made with the throttling calorimeter. It also shows the limit of moisture which may be determined at each pressure. These properties are perhaps better exhibited in the diagram (Fig. 251), in which absolute steam pressures are made ordinates, the amount of superheat abscissa, while the per cent. of moisture is shown by the curves. By use of the chart, Fig. 251, superheat and pressure given, the per cent. of moisture in steam may be read off directly, as closely as required in ordinary practice. The following table gives, for the ordinary steam pressure, the per cent. of moisture due to a given temperature in the calorimeter:

WATER IN STEAM.—BY THROTTLING CALORIMETER.

COMPUTED BY P. J. DARLINGTON.

(Per cent. of moisture.)

TEMPT.	GAUGE PRESSURE.								
F	40	45	50	55	60	65	70	75	80
215	0233	0253	0271	0290	0307	0322	0338	0354	0368
220	0207	0227	0245	0263	0280	0296	0311	0327	0340
225	0181	0201	0218	0237	0253	0269	0284	0300	0313
230	0154	0173	0192	0210	0227	0242	0257	0273	0287
235	0128	0147	0165	0184	0200	0215	0230	0246	0260
240	0102	0122	0139	0157	0173	0189	0204	0219	0233
245	0076	0095	0112	0130	0147	0162	0177	0192	0206
250	0049	0069	0086	0104	0120	0135	0150	0165	0179
255	0023	0042	0059	0077	0093	0108	0123	0138	0152
260	0005—	0016	0033	0051	0066	0081	0096	0111	0125
265	0030—	0010—	0006	0024	0040	0055	0069	0084	0098
270	0057—	0037—	0020—	0002—	0013	0028	0042	0057	0069
275	0083—	0063—	0047—	0029—	0013—	0001	0015	0030	0042
280	0109—	0089—	0073—	0056—	0040—	0026—	0011—	0003	0015
285	0136—	0116—	0100—	0082—	0067—	0053—	0038—	0024—	0012—
Diff. 1° F.	00052	00052	00053	00053	00053	00054	00054	00054	00054

The foregoing tables and the following remarks apply principally to the calorimeter when exhausting into the air without back pressure, this being the condition under which the calorimeter is usually used. This property of the throttling calorimeter renders it very convenient to use, as by means of the above table the amount of moisture can be read directly, knowing the pressure in the main steam pipe and the temperature in the calorimeter. A negative sign following a number denotes per cent. of superheat.

EFFECT OF ERRORS.

In obtaining percentages small errors in pressure gauge or in thermometer make very little difference in the resulting moisture.

Thus at 80 lbs. absolute steam pressure an error of one pound in the gauge reading will make a difference of only .07 of 1% of moisture, while an error of 2° in thermometer reading will make a little over 0.1%; this permits the use of instruments of ordinary accuracy and small first cost. If the calorimeter exhausts directly into the air, the back pressure may in general be neglected, and a thermometer will seldom be required reading higher than 260° or 300° F.

The error caused by loss of heat in the calorimeter by radiation is very difficult to estimate. This loss is probably a function of the surface of the calorimeter, divided by the amount of steam passing through it in a given time. It is very small for a large flow of steam, as the reading of the thermometer is often not changed by putting on or taking off a felt covering from the calorimeter, but on the other hand may be very large for a small flow. The following is a comparison of two of these calorimeters, both on the same steam line about 6 feet apart; No. 4 being on a vertical pipe, No. 3 on a horizontal pipe, with an elbow between them. These calorimeters were alternately clothed and unclothed with a covering of hair felting three-quarters of an inch thick.

The sketch shows the location of the calorimeters. (See Fig. 241.)

COMPARISON OF TWO THROTTLING CALORIMETERS.

FIRST RUN.

TEST BY MR. ARNOLD.

Neither calorimeter covered. No. 4 in vertical pipe; No. 3 in horizontal.

No.	Time.	Steam Pressure.		Thermometer Reading in Calorimeter.		Manometer Reading in Calorimeter.		Per Cent. of Moisture.	
		By Gauge.	Absolute.	No. 4.	No. 3.	No. 4. Inches.	No. 3. Inches.	No. 4.	No. 3.
1	3.25	56	71	256	247	¼	¼	00.95	01.40
2	3.30	60	75	254	243	2	¼	01.15	01.50
3	3.35	61	76	255	239	2	¼	01.05	01.70
4	3.40	64	79	258	244	2	¼	01.00	01.35
5	3.45	65	80	259	250	2¼	¼		

Average moisture in Calorimeter No. 4, 1.06 of one per cent.

Average moisture in Calorimeter No. 3, 1.56 of one per cent.

Difference in favor of No. 4, 0.50 of one per cent.

Second Run.

Calorimeter No. 4 covered with a covering of hair felting ¾ of an inch thick. No. 3 as before.

No.	Time.	Steam Pressure.		Thermometer Reading in Calorimeter.		Manometer Reading in Calorimeter.		Per Cent. of Moisture.	
		By Gauge.	Absolute.	No. 4.	No. 3.	No. 4. Inches.	No. 3. Inches.	No. 4.	No. 3.
1	3.55	56	71	263	249	1.65	.75	00.55	01.10
2	4.00	52	67	257	245	1.45	.55	00.65	01.30
3	4.05	53	68	261	251	1.30	.50	00.65	00.95
4	4.10	47	62	260	256	1.20	.40	00.35	00.50
5	4.15	41.5	51.2	257	251	1.10	.35	00.30	00.90

Average moisture in Calorimeter No. 4, 0.50 of one per cent.
Average moisture in Calorimeter No. 3, 0.93 of one per cent.
Difference in favor of No. 4, 0.43 of one per cent.

In the foregoing case, the effect of the covering was hardly perceptible, as the difference in the amount of moisture shown by the two calorimeters remained nearly constant at 00.5 of 1%. In the trial which follows No. 1 and No. 2 were tested, first by carefully wrapping No. 1 in hair felting and leaving No. 2 uncovered, and then reversing the conditions. Calorimeter No. 3 was used to compare the other results with; it was connected to an unperforated nipple, extending nearly across the pipe, carefully clothed, and remained unchanged during both trials.

COMPARISON OF TWO THROTTLING CALORIMETERS.

First Trial.

Calorimeter No. 1 clothed with hair felting ¾ of an inch thick. No. 2 not clothed.

No.	Time.	Gauge Pressure.	Calorimeter No. 1.			Calorimeter No. 2.			Calorimeter No. 3.			Pounds of Steam during Run.
			Tempt.	Manometer. Inches.	Amount of Superheat.	Tempt.	Manometer. Inches.	Amount of Superheat.	Tempt.	Manometer. Inches.	Amount of Superheat.	
1	4.22	38	253.5		41	221.5		9	235		23	From No. 1,
2	4.23	39	253		40.5	221		9	235.5	1.4	23	2.5
3	4.24	39.5	253		40.5	219		7	236		24	
4	4.25	39.5	253	1.4	40.5	220	1.5	8	239		27	
5	4.26	39	253		40.5	220.5		9	239.5	1.3	27	From No. 2,
6	4.27	39	253		40.5	221		9	240		28	1.0
7	4.28	41	253		40.5	219	1.6	7	240		28	
8	4.29	41.5	253		40.5	219		7	240.5		28.5	
9	4.30	42	253		40.5	219		7	240		26	From No. 4,
10	4.31	42	253	1.6	40.5	221	2.1	9	240.5	1.3	28.5	2.0

SECOND TRIAL.

Calorimeter No. 1, not clothed; Calorimeter No. 2, clothed.

No.	Time.	Gauge Pressure.	Calorimeter No. 1. Tempt.	Calorimeter No. 1. Manometer. Inches.	Calorimeter No. 1. Amount of Superheat.	Calorimeter No. 2. Tempt.	Calorimeter No. 2. Manometer. Inches.	Calorimeter No. 2. Amount of Superheat.	Calorimeter No. 3. Tempt.	Calorimeter No. 3. Manometer. Inches.	Calorimeter No. 3. Amount of Superheat.	Pounds of Steam during Run.
1	4.50	46.5	249		36	241		29	235		23	From No. 1,
2	4.51	49	249		36	240.5		29	235.5		23.5	3.5
3	4.52	51	248	2.4	36	242.5	1.2	30	239	1.6	27	
4	4.53	49.5	248		35	243		31	241		29	
5	4.54	48	248.5		35.5	244	1.8	32	242		30	From No. 2,
6	4.55	45.5	249	2.1	35	245.5		33	243	2.3	31	1.60
7	4.56	44.5	250		37	246		34	244		32	
8	4.57	45	249		36	247	1.5	35	241		29	
9	4.58	45	249	2.3	36	247.5		35	240	1.4	28	From No. 4,
10	4.59	46.5	248		35	247		35	239.5		27	2.5

In the first case Calorimeter No. 1, clothed and discharging two and one-half times as much steam as Calorimeter No. 2, which was not clothed, showed 0.32% of moisture, as against 2.20% in Calorimeter No. 2.

In the second case, Calorimeter No. 2, clothed, No. 1, not clothed, but No. 1 discharging twice as much steam as No. 2, gives as a result 0.85% of moisture in Calorimeter No. 1, as against 0.86% in Calorimeter No. 2.

Calorimeter No. 3, clothed in both cases, and in same condition, showed in the first case 1.05% moisture, and in the second case 1.1% moisture.

Supposing the quality of steam to remain the same in the two cases, which is seen to be true within .05 of 1% by a comparison of the determinations from Calorimeter No. 3, we see that with the Calorimeter No. 1, having a flow of steam from 25 to 35 lbs. per hour, the removal of the clothing increased the amount of moisture 0.53 of 1%, while with Calorimeter No. 2, flow of steam from 10 to 16 lbs. per hour, the addition of clothing decreased the amount of moisture 1.14%.

This trial seems to show the importance of radiation determinations, which are probably only to be made by comparing the result of a trial made on steam of a known quality.

ERROR DUE TO SPECIFIC HEAT.

The effect of error, from the fact that the specific heat of steam at high pressure is not known, cannot be determined except by a series of observations on steam of a known quality. It seems to the writer that this calorimeter affords a ready means of determining the variation in the specific heat of steam, at different pressures, even if the results cannot be obtained in absolute units. Thus, if steam of a known quality be used, and expanded to a lower pressure by passing into the calorimeter, supposing no loss or gain of heat, all the quantities involved would be known excepting the specific heat of steam. Thus supposing the original steam slightly superheated, and let H represent the normal heat in a pound in B. T. U. of dry and saturated steam, T its normal temperature, T' its actual temperature, K' its specific heat, also let H, K, t, and t' represent similar quantities for a second calorimeter, using the same steam, then,

$$H' + K'(T' - T) = H + K(t' - t),$$

from which the ratio of K to K' may be found.

This method is now the subject of investigation in the laboratory.

THE CHEMICAL CALORIMETER.

Two observers, Messrs. Bierbaum and Fitts, made a number of experiments on a calorimeter of this sort. The results were very promising, but some trouble with standard solutions prevented exact results in some cases.

The method used consisted in adding a little common salt, say three or four ounces, to the water in the boiler, it having been previously found by trial that dry steam would not absorb any of this salt; if a given weight of the steam be condensed, and if it be found to contain salt, it will be due to water mechanically entrained.

The proportion that the salt in a given weight of condensed steam bears to that in a given weight of water, drawn from the boiler, is the percentage of moisture in the steam. The method of analysis is a volumetric one, and is very accurate and rapid.

The method is as follows: Draw from the boiler a small amount of water and condense an equal weight of steam, which are to be kept in separate vessels. Add to each of them a few drops of neutral chromate of potash, but in each case an equal quantity, which amount may be measured by a pipette; the same amount should also be added to a vessel containing an equal weight of distilled water, in order to obtain a standard or zero point for the scale used in the analysis.

By means of a graduated pipette a titrated solution of nitrate of silver is permitted to flow, a single drop at one time, into each of the three solutions. The effect is to cause the formation of the chloride of silver, and until that formation completely takes place the resulting liquid will be whitish or milky, but because of the presence of the bichromate, the instant the chloride has all been precipitated the liquid turns red. The amount of nitrate of silver required is measured by the graduated pipette, and gives the information regarding the salt present.

The following example is taken from the French journal *L'Industria.*

Using in each case 100 cubic centimetres of liquid, containing a few drops of neutral chromate of potassium, and dropping a titrated solution holding 10.8 grammes of silver to the litre, the following result was obtained :

AMOUNT OF NITRATE OF SILVER REQUIRED TO TURN 100 c. c. RED.

100 c. c. of	Trial First.	Trial Second.	Trial Third.	
Condensed Steam	0.1 c. c.	0.05 c. c.	0.1 c. c.	*a*
Water from the Boiler	18.6 c. c.	14.0 c. c.	18.35 c. c.	*b*
Distilled Water	0.05 c. c.	0.05 c. c.	0.05 c. c.	*c*
Amount Moisture	0.00369	.0	0.00375	Average, 0.00248

Letting the results with each of these samples be denoted by *a*, *b*, and *c* respectively, and the amount of moisture $1 - x$

$$1 - x = \frac{a - c}{b - c}.$$

This gives the following results:

	First Trial.	Second Trial.	Third Trial.
Amount Moisture	$\frac{0.1-0.05}{18.6-.05}=.0037$	$\frac{0.05-0.05}{14.0-0.05}=0$	$\frac{0.1-0.05}{18.35-0.05}=.00375$

Average $= 0.0025$.

This method is evidently applicable only in determining the amount of moisture in the steam as it leaves the boiler, and will give us no information regarding the additional moisture that may be added to the steam by condensation.

It will determine the amount of priming of the boiler only.

A SEPARATOR CALORIMETER.

The following form of calorimeter has been in use in the laboratory experiments at Sibley College for some months, and, from comparisons made with other calorimeters, promises to be sufficiently accurate for ordinary commercial purposes; it presents the advantage over the throttling calorimeter, of not failing to operate for a small quantity of moisture, and over the other forms, of being more portable and less likely to be deranged. It can be made to give quite accurate determinations without the use of pressure gauge or thermometer, in fact for ordinary qualities of steam neither of these instruments is required. The accuracy of the instrument depends upon the complete mechanical separation of the entrained water from the steam, the water being discharged and caught below; the dry steam is discharged through the orifice O, and condensed in a supplemental vessel of water and weighed, or, knowing the pressure and area of orifice O, it may be computed.

The proportion and form of chamber and ratio of area of discharge to that of supply have been the subject of careful experiments; and in the recent forms of instrument with supply pipe one-half inch in diameter and discharge orifice one-sixteenth inch in diameter, we have been able to secure not nearly dry but perfectly dry steam in the exhaust through a wide range of steam pressure and quality of steam. The accuracy of the instrument depends entirely upon the exhaust or discharge steam being perfectly dry; to determine that a throttling calorimeter was attached directly to the exhaust orifice of the form previously shown, and a calibrated pressure gauge was attached to the calorimeter at G.

The following show some of the results of these determinations. The first case is a test of the exhaust steam from a high-pressure cylinder to the intermediate in a triple-expansion engine. The temperature and quality of the exhaust steam were obtained from the throttling calorimeter, attached as described.

SEPARATOR CALORIMETER DETERMINATION OF QUALITY OF EXHAUST STEAM FROM HIGH-PRESSURE CYLINDER.

The water was maintained as nearly as possible at the same level in the gauge glass during the run.

The per cent. of moisture was calculated by the formula

$$1 - x = \frac{W}{W + w}$$

in which $W =$ the weight of water precipitated, corrected for radiation loss of the instrument, $w =$ the weight of steam condensed or calculated. The following table exhibits data and results :

	First trial.	Second trial.	Third trial.	Fourth trial.	
Beginning run time	7:35	8:41	10:08	11:42	
Ending run time	7:50	9:01	10:23	11:57	
Duration	15 m.	20 m.	15 m.	15 m.	
Steam in run condensed	19 oz.	24 oz.	20 oz.	20 oz.	
Steam per hour (w)	76 oz.	72	80	80	
Gauge pressure	25	40	35	36	
Absolute pressure	39.5	54.5	49.5	51.5	
Water precipitated in run	3¼ oz.	3½ oz.	3	3¼	
Water precipitated per hour (w')	13 oz.	10½ oz.	12	13	
Throttling Calorimeter. — Temperature of exhaust steam	246°	258°	255°	257°	
Throttling Calorimeter. — Quality exhaust steam	99.95	99.93	99.99	99.98	Calculated from diagram, Fig. 251.
Radiation correction to precipitated water per hour	−¾ oz.	−¾ oz.	−¾ oz.	−¾ oz.	Computed.
Correct weight entrained water (w)	12¼	10	11¼	12¼	
Quality, steam x	85.96	87.88	87.5	86.c5	
Per cent. moisture, $1-x$	14.04	12.12	12.5	13.35	

The determinations by the throttling calorimeter show dry steam within the limits of errors of observation.

The following are tests of the quality of steam discharged from the orifice O, in a few of the determinations made by Messrs. Brill and Meeker, on a test of the comparative values of different separators for steam-engines. In this case two calorimeters of this type were used. One, marked A on trials, was placed in advance of the separator to be tested; the other, marked B on the trials, was placed so as to test the quality of the steam after passing the separators. The readings of calorimeters A and B were taken simultaneously. To change the quality of steam the main pipe was surrounded with a jacket filled with cold water, which was maintained at a greater or less height as desired, thus producing more or less moisture in the steam-pipe. (See Fig. 243.) The complete observations are given on four trials, after that simply the averages of the separate trials. From these it is seen that, for quality of steam entering the calorimeter, varying from 70% to 100%, the discharge is dry, within limits of error of observation.

SEPARATING CALORIMETER.

TRIALS BY BRILL AND MEEKER.

Observations on Separator Calorimeter.						Examination of Exhaust Steam from Calorimeter by Throttling Calorimeter.		
	T	*P*	*W*	*w*	*X*	*t*	*x*	
Calorimeter.	Duration Run. Minutes.	Gauge Pressure. Pounds.	Pounds Separated Water in Run.	Pounds Condensed Steam in Run.	Quality Steam. Per Cent.	Tempt. in Calorimeter.	Quality Steam in Exhaust.	No. of Observations.
		86				282	99.0	
		77½				282	100.0	
		77½				282	100 0	
A	25	83				286	100.10	
		79½				289	100.35	
		80				287	100.20	
			0.15	4.85	97.0			
		85				277	99.5	
		76				282	100.02	
		76				280	99.98	
B		76				280	99.98	
		81½				282	99.9	
		78				284	100.05	
		78				281	99.95	
	25		0.175	5.55	96.94			
		81½				288	100.1	
		80½				289	100.25	
		80				283	100.2	
A		81½				248	100.15	
		81½				289	100.2	
	25	77½				288	100.25	
			1.25	4.61	78.63			
		80				282	99.95	
		79½				283	100.00	
		78½				282	99.80	
		80				282	99.95	
B		80				283	99.98	
		76				282	100.05	
	25		0.151	5.45	97.82			
Average of 24 observations							99.9964	

	T	*P*	*W*	*w*	*X*	*t*	*x*	No. of Observations.
A	25	81.5	1.15	4.45	79.46	281	99.95	6
B	25	78.2	0.15	5.20	97.2	281 3	100.00	6
A	25	80.8	0.525	4.25	89.005	286.5	100.00	6
B	25	79.5	0.150	4.75	96.94	281.8	99.95	6
A	25	78.5	0.800	5.000	94.34	282.8	100.00	6
B	25	77.6	.150	5.45	97.32	282.3	100.00	6
A	24	79.5	1.8	4.55	71.65	280.1	99.94	6
B	24	78.5	1.4	4.90	77.77	279.5	99.9	6
A	20	83.5	1.15	4.1	77.67	286.5	100.00	5
B	20	81.6	1.70	4.75	73.64	283.7	99.98	5
April 17	20	74.8	0.65	3.95	85.87	283.7	100.05	5
	20	82.0	0.85	3.95	82.29	246.8	100.05	5
	20	82.6	0.35	4.15	92.23	285.6	100.0	5
	20	81.5	0.20	3.95	95.15	285.2	100.05	5
18 *A*	20	81.4	2.20	4.325	66.28	283.1	100.0	5
B	20	80.3	0.30	4.55	93.81	282.8	100.0	5
20 *A*	20	82.0	0.20	4.65	95.8	282.8	99.98	5
B	20	81.1	0.20	4.40	95.7	284.0	100.0	5
Average of 18 trials, involving 98 observations							99.998	

From a remark by Mr. Barrus, Vol. XI., page 793, in a paper on Universal Steam Calorimeter, I conclude that his attention had evidently been called to a calorimetric method on this plan, for he states: "When the drip-box is used in this way, the author

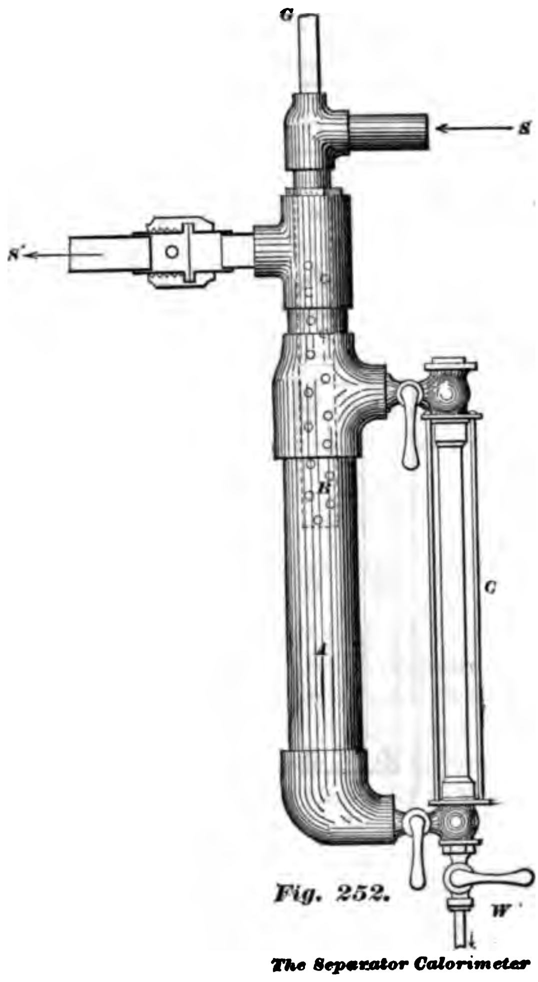
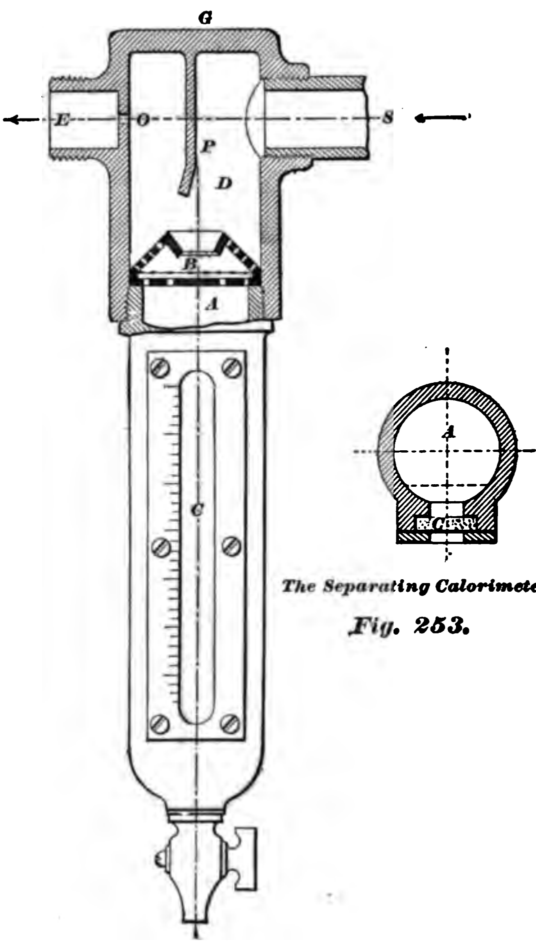


Fig. 252.

The Separator Calorimeter

has found that almost the whole quantity of moisture in a sample will be deposited here, and very little moisture will be left to pass over into the heat-gauge. Indeed, the experiments show that the drip-box alone, with a suitable orifice or valve provided at the top, so as to obtain a proper circulation through it, would form a very satisfactory instrument for determining the quantity of moisture in any case where the steam contained much of it." We find in our trials that it is even as reliable and as accurate as the barrel calorimeter, for small quantities of moisture, and even for small degrees of superheat, when the radiation loss or condensation



The Separating Calorimeter.

Fig. 253.

deposits some water, so that the escaping steam is not superheated. For these cases, however, a throttling calorimeter is more convenient, and one is readily formed by screwing on to the discharge nipple for the sample, a tee, opening downward and containing a thermometer cup screwed into the top opening.

FORM OF THE CALORIMETER.

Two forms of the calorimeter have been in use, one shown in Fig. 252, made up of pipe fittings; steam is supplied through the half-inch pipe at *S*, the water is caught in the chamber *A*, and the dry steam is discharged through the orifice *O*. A hose can be attached to the exhaust pipe and carried to a vessel of water, where it may be condensed.

The form more recently adopted, shown in Fig. 253, is a steam separator in miniature; steam is supplied at *S* through a half inch pipe and deflected by the diaphragm on to the perforated partition *B*; the dry steam rises and passes out through the orifice *O*, while the moisture and entrained water is precipitated in the chamber *A*. The height of this water can be told by the glass *C*, while the graduated scale along the edge gives the weight in hundredths of a pound. In this form, gauge or thermometer connections (not shown in figure) may be provided at *G*, and water glass connections are made somewhat heavier than shown in the figure. This form has proved less likely to leak than the former, is more compact in shape, and more convenient to use.

METHODS OF USING.

In the use of the instrument it has been our practice in general to lead a hose from the exhaust to a condensing vessel, and condense the steam for a run of one-half or one hour in length. This is readily done, since with 80 lbs. steam pressure the weight of dry steam discharged will not much exceed 12 lbs., and would require a condensing vessel containing only about 200 lbs. of cold water. The entrained water is to be drawn off from time to time during the run, keeping it at about the same level; this may be done almost automatically by setting the cock so as to drip slightly; this water may be discharged directly into a glass flask through the cock *W*, which will in general cool it sufficiently to prevent any vaporization; in fact, in most cases we have found that the slow drip of water through the tube was effectual in cooling it as low as 100° Fahr.

In case the weighing is to be done by the attached graduated scale, the cock can not be set so as to drip, and if the capacity of the vessel is not sufficient for the water during the run, it must be drawn off at intervals. In starting and stopping the test, the instrument should be working under uniform conditions, and as these are likely to be somewhat affected by drawing out or putting in a hose in the condensing water, more accurate results are attained by setting the poise of the scale containing the condensing water somewhat in advance of the weight, and the instant this scale beam rises, commence the run. Stop in a similar manner. In general, however, I would advise an accurate determination of the area of orifice *O* and the use of a pressure gauge on the calorimeter, then compute the discharge of the dry steam by Napier's Rule :

$$\text{Flow in pounds per second} = \frac{\text{absolute pressure} \times \text{area in sq. inches}}{70}$$

as more accurate than the process of weighing. The error by the use of this rule was shown by Prof. Peabody not to exceed one or two per cent. The following table is convenient for calculation :

TABLE OF DISCHARGE OF STEAM IN POUNDS PER HOUR CALCULATED BY NAPIER'S FORMULA.

Absolute pressure pounds.	POUNDS OF STEAM.		
	Diameter of orifice 1/16 inch.	Diameter of orifice 1/32 inch.	Diameter of orifice 1/8 inch.
1	0.039	0.158	0.631
2	0.079	0.316	1.262
3	0.118	0.473	1.893
4	0.158	0.631	2.524
5	0.197	0.789	3.155
6	0.237	0.947	3.786
7	0.276	1.104	4.417
8	0.315	1.262	5.048
9	0.354	1.420	5.680
10	0.395	1.578	6.311
20	0.789	3.155	12.623
30	1.183	4.733	18.937
40	1.578	6.311	25.244
50	1.972	7.889	31.556
60	2.367	9.467	37.867
70	2.761	11.045	44.179
80	3.156	12.623	50.488
90	3.550	14.200	56.800
100	3.947	15.778	63.115

ERRORS FROM USE OF THE INSTRUMENT.

Let W = the weight of entrained water.

Let w = the weight of the dry steam discharged in a given time. Let $1-x$ = per cent. of moisture in the steam.

Then allowing no loss for radiation

$$1 - x = \frac{W}{w + W}$$

From the examples actually quoted w is likely to run less than 5 lbs.; in this case the error of one-tenth of a pound in weighing the entrained water W might make an error of 2% in the result. To make the error less than one-fifth of 1% the value of W must be taken accurately to one one-hundreth of a pound, or to about one-sixth of an ounce. If the run is longer, the error would have less effect, as it diminishes in proportion to the amount of steam discharged.

Condensation Loss.—Since such small quantities of water have such a marked effect on its accuracy it will evidently be necessary to ascertain and correct for the condensation produced by the instrument itself. We have made this correction by drawing off samples of steam nearly dry, taken as nearly as possible under the same conditions, in a throttling calorimeter carefully wrapped up and in the separating calorimeter. The quality shown by the throttling calorimeter is assumed to be correct, and knowing this the corresponding condensation loss is computed as already explained.

Let this loss in pounds for the same length as the run, which should for convenience be reduced to a standard of an hour in length, be denoted by r. Then the corrected value of per cent of moisture would be

$$1 - x = \frac{W - r}{w + W + r}.$$

With careful handling the instrument will give results within one-half per cent., depending largely on the method of handling the instrument and the accuracy with which the radiation loss can be determined. This radiation loss is no small amount, often in itself being sufficient to make a difference of over 2% in the determination.

The following is a determination of radiation loss, made by Brill and Meeker:

COMPARISON OF QUALITY DETERMINATIONS ON SAME SAMPLE OF STEAM, WITH THROTTLING AND SEPARATING CALORIMETERS, TO DETERMINE RADIATION LOSS OF SEPARATING CALORIMETER.

AVERAGE OF 3 RUNS, 20 MINUTES EACH.

Steam Pressure Gauge.	Separating Calorimeter.					Throttling Calorimeter.	
	Wt. of Steam during Run. Pounds.	Wt. of Entrained Water.	Total Wt. of Steam.	$\frac{W}{W+w}$.	Radiation Loss by Condensation. Pounds.	Temperature in Calorimeter.	Per Cent. of Moisture.
P	w	W	$W+w$.		r		$1-x$.
76	4.775	0.1718	4.947	.0347	.047	234.2	2.55
77	4.65	0.1406	4.79	.02935	.053	247.4	1.8
81.5	4.85	0.14859	5.00	.0297	.053	248.0	1.9
83.0	4.70	0.15062	4.85	.03105	.048	246.0	2.10
					.050		
Per hour.					.1506		

In order to avoid this correction for condensation due to radiation, Mr. Daniel Royse suggests the use of two separating calorimeters of the form shown in Fig. 252, arranged so that the exhaust from the first calorimeter shall pass directly into the second. The orifice between the two calorimeters should be fully one-eighth inch, that from the second to be less than one-sixteenth inch in diameter. The first will discharge dry steam into the second, which will deposit moisture due to condensation only; if the two calorimeters are exactly alike this amount will be the radiation loss of the first instrument.

Then let W equal the entrained water obtained from the first instrument, let r equal the entrained water obtained from the second instrument, let w equal the weight of dry steam discharged from the second instrument.

We should have

$$1 - x = \frac{W -}{W + w + r}$$

DISCUSSION.

Prof. R. H. Thurston.—Since the time of Hirn, who first showed and practically proved by actual use the necessity and value of the calorimeter in determining the quality of steam in all engine and boiler trials, the subject has attracted much attention from engineers dealing with the steam-engine, and many questions have been raised, and some have been settled, since his time. I do not know to what extent the lead of Hirn was followed in earlier years in Europe. In this country, so far as I know, but little had been done in this direction until Mr. Emery applied the instrument in his extensive work at the Centennial Exhibition of 1876. Mr. John D. Van Buren and I, when on duty at the Naval Academy in our later navy days, about 1865–70, became interested in the matter, and he then devised a form of continuous calorimeter which we used later in boiler trials, and especially in those of the Howard and other boilers at the American Institute Fair, the late Theron Skeel handling the instrument very successfully.

It was in 1871, I think, that, the question arising whether the sampling of the steam could be depended upon, I was authorized by a committee, of which I happened to be chairman, to take advantage of the liberality of the late Mr. John B. Root, one of the competitors at the exhibition of that year, to make use of a Root boiler as a condenser, and thus to condense all the steam made by the competing boilers—our mammoth calorimeter serving both to give us absolute measures of the quality of the steam by gauging it all, and to show the probable accuracy of the calorimetric measures by sample, previously and since necessarily relied upon. We had a singular variety of size and type of boiler, and the result was taken as satisfactorily proving that the ordinary calorimeter could be made to give useful information, and could be adopted as both useful and essential in the determination of the real efficiency of the boiler and the quality of the steam entering the steam-engine, the efficiency of which is largely dependent upon the dryness of the steam supplied.

The ingenuity and skill of later engineers and inventors, especially of Prof. Peabody and of Mr. Barrus, have given us new varieties of this instrument, and we are now able to say that this essential of the accurate work of the engineer in this direction may be depended upon to give us the needed facts with all required accuracy. Something has remained to be done, however, in the determination of the conditions favoring accuracy, or the reverse, both in the sampling of the steam and in the use of the instrument; and this paper of Prof. Carpenter, detailing some of our attempts to throw light upon the matter in the course of regular work in our steam-laboratory, seems to me to supply some of the information most desired in practical operation in boiler and engine trials. His discussion of errors is, so far as I now call to mind, the most satisfactory that I have met with, and gives a good idea of the conditions to be sought, and the nicety of work in handling the apparatus demanded to insure accurate results. Mr. Kent has taken a still simpler way of illustrating these points.

It is easily seen how it may occur that the Hirn instrument, the simplest of all forms, may give widely inaccurate single measures, and how averaging, and the discarding of such obvious errors, may still enable the observer to obtain with it the knowledge that he seeks.

The form of calorimeter which Prof. Carpenter has himself devised as a "separator calorimeter" is the simplest of that class yet produced, and, I am inclined to think, represents the instrument reduced to most perfect simplicity. Our work with it also indicates that it may—in all ordinary cases at least—be relied upon to do its work thoroughly. Where it fails, should it fail at all, it will be certain to be in cases in which the quality of the steam is so low that it would be condemned in any event, and slight error in measures of very wet steam would be of no consequence. The invention strikes me as a very interesting and admirable one.

With classes of seniors and graduates aggregating seventy-five or a hundred men, mainly bright, interested, and well-trained by an earlier systematic course of practice, we are able to get a very large amount of work done in defined directions, and, we hope, may find ways of investigating a still greater variety of forms of the instrument, and under still more varied conditions of operation. In the course of this work the expedients necessarily resorted to to secure efficient instruction of such large numbers have led us to many such products of the ingenuity of the instructors as are illustrated by some here described. Among these perhaps the most novel is Prof. Carpenter's application of the calorimeter to the measurement of the quality of steam in the engine cylinder at any and every point in the stroke, or of the cycle, taking a sample at every revolution and from any desired part of the diagram or revolution, including the compression period, even. While I think it is still uncertain to what extent accuracy of measurement has yet been carried, I am inclined to think that it is fairly accurate at least, and it seems to me likely that it may prove a very valuable method of studying the engine cycle thermally, and of supplementing Hirn's and Donkin's methods.

Prof. C. H. Peabody.—This paper shows that by the aid of the throttling calorimeter and the separator calorimeter the quality of steam may always be determined with ease. It also shows that the two types should never be combined, for if steam is so moist that the throttling calorimeter cannot be used directly, then the determination of the quality by the separator calorimeter can be made with an error of not more than half of one per cent., which is sufficient for all engineering work. When much moisture is present an error of 2% may be of small moment, in which case the flow of steam may be gauged by allowing it to flow through an orifice. I must express my satisfaction that Prof. Carpenter has *not* used that method of determining the discharge of steam from his separator calorimeter in his tests.

Attention should be called to the fact that the determinations of the total heat of steam by Regnault are affected by an error of one in one thousand, or, perhaps, of one in five hundred. It therefore appears to be unnecessary to carry calculations of the quality of steam further than tenths of a per cent. I suppose that the next figure given by Prof. Carpenter may be considered to guard the tenths.

I am most interested in the methods employed for loading steam with water, having made some rather unsuccessful attempts in that line during the past year. In our work we forced water through a sprayer in a vertical 5-inch pipe and supplied steam at the same time through a 2-inch pipe. At the top of the 5-inch pipe there was an elbow and then 40 feet of horizontal 2-inch pipe. As the work was not considered satisfactory, it will suffice to say that a throttling calorimeter at the end of the horizontal pipe showed the steam to be nearly dry, while a surface condenser used as a calorimeter showed several times as much water as steam.

I have long been aware that a claim of a patent on calorimeters could be made to cover all throttling calorimeters, or, perhaps it may be more proper to say would do so were it not a fact that a printed description of such calorimeters preceded the patent.

The form of throttling calorimeters proposed by Mr. Heisler is very neat and compact, and deserving of commendation. We have tried our ingenuity in our laboratory to improve on the first rough type of the throttling calorimeter only to find that the old type appears the most satisfactory. We make them of 4-inch pipe 10 inches long, with a thermometer cup long enough to include the stem of the thermometer nearly to the end of the mercury thread. We open the supply valve till the thermometer ceases to rise, and then know that the error from radiation has disappeared; then we throttle the exhaust till the pressure is 10 or 15 pounds above the atmosphere in the calorimeter, because the temperature of saturated steam then varies less with a small change of pressure. Our only trouble is that the students always get their work right even when they are a little careless.

In the description of the "injector" calorimeter Prof. Carpenter says that the tank delivering water to the injector should be set so as to give little or no head. Now, it is of no consequence whether there is a head or not, but it may not be improper to note that the flow of water to an injector is caused by the difference between the pressure of the air and the pressure in the injector, plus the head; of which the former is the greater in most cases, for even a non-lifting injector will show a pretty good vacuum in the combining chamber.

Finally, there is one thing that I wish to criticise, and that is the use of the expression "per cent. of superheating." Of course we all know what is meant, but such expressions are liable to lead to confusion in a subject in which all is clear and easy if one always thinks and says what he means, but in which much confusion of thought exists at the present time among half-informed men. If any calorimeter appears to show x, the quality, greater than unity, then of course the steam is shown to be superheated and it is easy to calculate the temperature of the steam supplied. Thus for the throttling calorimeter, using Prof. Carpenter's notation,

$$k(t_s - t_1) + L + g = H + k(T - t),$$

in which I have taken t_1 to be the temperature of saturated steam at the pressure in the supply pipe, and t_s to be the temperature of superheated steam in that pipe.

Mr. Geo. H. Barrus.—I dislike to criticise adversely a paper which was, no doubt, intended to be a valuable contribution to the literature regarding steam calorimeters; but there are so many points in this paper which are misleading, not to say erro neous, that they should not pass unnoticed.

THERMOMETER CUPS.

Prof. Carpenter refers to the form of thermometer cups which are in use in the laboratory of Sibley College, and he makes a statement which throws doubt upon the usefulness of cylinder oil for filling the cup. He says that cylinder oil is peculiarly sensitive to moisture, and that if moisture is left in the cup it will be a long time before the thermometer assumes the temperature of the surrounding fluid. I do not know what ground he has for making this statement. I have used thermometer cups of a form similar to the one he describes for a period extending over some fifteen years, and the number of times which I have employed them amounts to several hundred. I have never had any difficulty with cylinder oil, which it is my practice to use, and the statement which he makes does not accord with my experience. I would say, however, that when a thermometer cup leaks at the bottom through defective construction there is difficulty in getting satisfactory indications, and I suspect that the trouble about which the author complains may have arisen from some disorder of this kind.

METHODS OF OBTAINING A FAIR SAMPLE OF STEAM.

The practice of the author's laboratory, he says, is to use a ½-inch pipe, perforated with ¼-inch holes, etc. In the accompanying sketch, Fig. 313, I have represented such a pipe on an enlarged scale, corresponding as near as possible to the dimensions given in Fig. 240 of the paper.

I see no object in making these holes so large. It will require only four ¼-inch holes, such as he employs, to carry all the steam which is used by the ½-inch pipe, and, considering that the calorimeter is drawing much less than the capacity of the pipe, it would be an easy matter for the steam to be drawn from a single one of these holes rather than a little from each. This practice would vitiate the only object of extending the pipe into the interior and perforating it, which is to obtain a sample of the steam carried through it.

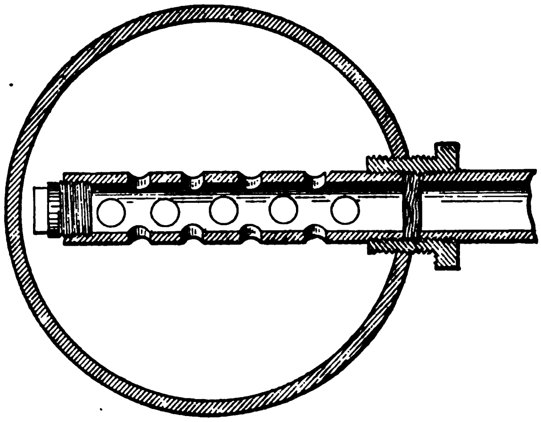


FIG. 313.

My practice is to use $\frac{1}{8}$-inch holes, and this practice is indicated in the accompanying Fig. 314, which is drawn to the same scale as the preceding one.

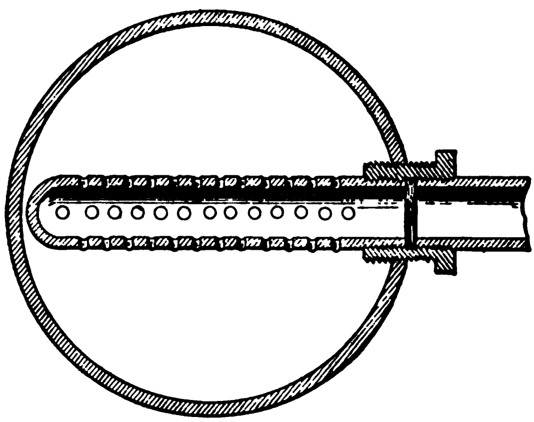


FIG. 314.

I think no one will question that the use of the small holes is much to be preferred.

PROF. CARPENTER'S TESTS OF VARIOUS MODES OF ATTACHMENT.

I have taken pains to examine quite carefully the results of the five trials which are given, showing the effect of various methods of attaching the supply pipe. The results of these trials are summarized in the following table. In order that they may be thoroughly understood at a glance, small sketches are given in column No. 4, Fig. 315, opposite the various tests, to show the arrangement of the supply pipe at the point where it enters the main.

1	2	3	4
No. of trial, as designated in the paper.	Average percentage of moisture shown by the calorim. on each side of the one with adjustable connection, *i.e.*, average of No. 2 and No. 4 calorimeters.	Percentage of moisture shown by calorim. with adjustable connection, *i.e.*, calorim. No. 3.	Sketch showing form and position of adjustable connection of calorim. No. 3.
1	1.71	1.56	
3	1.65	1.90	
2 Ditto.	1.52 Difference of .25.	2.25	
4	1.53	2.57	
5	1.47	1.85	

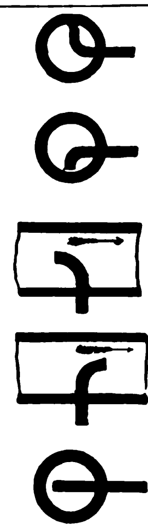


Fig. 315.

It can hardly be imagined that these experiments are trustworthy. It will appear by examining the table, especially the trials numbered 2 and 4, that the steam was wetter in Trial No. 4, with the connecting pipe pointing away from the current, than it was in Trial No. 2, where the connecting pipe was pointed toward the current. In the former case the particles of moisture in the steam would certainly be carried by their momentum beyond the opening and away from the pipe, while in the latter case they would be caught by the open mouth and conducted into the pipe. The indication of the calorimeter when the supply pipe was turned toward the current ought, therefore, to show a larger amount of moisture than that shown when the pipe was turned away from the current, and I can imagine no reason why this showing should not have been made if the tests were correct. A further indication of the untrustworthiness of the experiments is shown by the two trials which are designated No. 2, made under the same conditions, one of which shows a difference between columns 2 and 3 of .73, while the other shows a difference of .25.

Again, it can hardly be imagined that the experiments are reliable when they show that the further the calorimeter is applied from the point of supply of the steam in the pipe the dryer the steam appears. It is commonly accepted that the effect of radiation from a pipe is such that the further steam travels away from the point of supply the wetter it becomes, owing to condensation from this loss. In every case of Prof. Carpenter's experiments the No. 4 calorimeter shows a larger amount of moisture than the No. 2 instrument, No. 4 being applied at the point nearest to the incoming steam. Fig. 316, which is appended, shows graphically these results. The paper does not state it, but it may be presumed that the various trials were all made in one day, considering that the first trial lasts from 3 o'clock until 3:10, the second trial from 3:28 to 3:37, the third from 3:44 to 3:53, the fourth from 4:52 to 5:01, and the fifth trial from 5:29 to 5:38 o'clock. The indications of calorimeters No. 2 and No. 4, which were in constant use unmolested, may therefore be taken to indicate the quality of the steam passing through the main supply pipe at the different hours named. Referring to the sketch No. 3, it will be seen that with No. 4 calorimeter the moisture begins at 1.77%, increases to 1.85%, and then gradually falls to 1.6%. No. 2 calorimeter begins at 1.66%, drops to 1.2%, then rises to 1.5%, and finally falls gradually to 1.35%. If, now, the experiments had been trustworthy, the two lines representing the performance of the two calorimeters would have run in a parallel direction. In point of fact they are anything but parallel, although during the periods from 4 to 5:30 P. M. there is some approach to parallelism.

It would be interesting to know exactly what effect the various systems of sampling steam for calorimeter work produce so as to be able to employ the right method. There is no doubt that the difference between the different methods is small at best, and experiments which are intended to show these differences should be conducted with extreme care, and so planned as to avoid all sources of error. The experiments of Prof. Carpenter are not only inconsistent in their results, but they were not planned in a way to determine nice questions, however well they might have been conducted. The defects in the plan on which they are made are as follows:

First.—The comparison was made between three instruments applied continuously at three different points, and no correction in the readings of the various instruments was made for their individual errors. Without such corrections the results are worthless. It would have been better to avoid making correc-

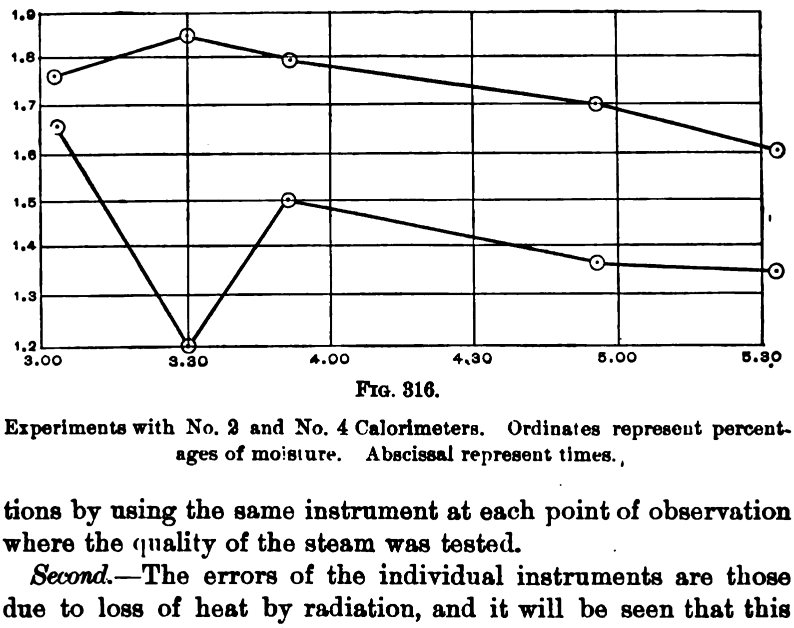


FIG. 316.

Experiments with No. 2 and No. 4 Calorimeters. Ordinates represent percentages of moisture. Abscissal represent times.

tions by using the same instrument at each point of observation where the quality of the steam was tested.

Second.—The errors of the individual instruments are those due to loss of heat by radiation, and it will be seen that this loss varied with the different instruments, on account of the varying temperature of the outflowing steam.

In trial No. 2, for example, the outgoing temperature varied from 250° to 227°; in trial No. 3 it varied from 251° to 231°; in trial No. 4 from 247½° to 223°; and in trial No. 5 from 242° to 227°.

Third.—The effect of radiation from the instruments is not constant for the three instruments, for the reason that the various calorimeters discharge widely different quantities of steam. Assuming that the size of the orifice was the same in each, the difference in the discharge is seen in the reading of the manometer, which in trial No. 1 is .4 inch for one instrument, .8 inch for another, and .5 inch for the third. A similar difference exists in all the other trials.

Fourth.—Looking at the series of tests as a whole, it will be noticed that the pressure in the supply pipe varied from 62 lbs. at 3 o'clock to 48 lbs. at 5:29 o'clock. To make experiments reliable the pressure of the steam should have been constant throughout the whole series.

Fifth.—Examining each trial by itself, it is seen that from one end of the trial to the other there is a wide variation in the steam pressure. In the case of trial No. 3 the steam pressure varies from 55 lbs. to 60.5 lbs., and this occurs in 7 minutes time. In connection with this change of pressure there is a similar variation in the temperature shown by the thermometer in the calorimeter. In this particular trial with calorimeter No. 3 the variation is from 231° to 238°. There is an absolute need of a steady pressure and steady conditions in work of this kind to get trustworthy results. I speak from a wide experience on this point, and I am sure it will be confirmed by every one who has experimented with calorimeters.

Sixth.—The individual experiments are of too short duration. Readings were taken every minute for 10 minutes, and this length of time is not sufficient to get the instrument thoroughly heated to its normal condition, to say nothing of its being too short to obtain proper averages.

The report of the tests, as given in the paper, is not full enough carefully to weigh the results. One thing omitted is information regarding the size of the main pipe from which the calorimeters were supplied, and the condition of this pipe, whether covered or uncovered, and, if covered, what sort of material was used for its protection. Nothing is stated in regard to the quantity of steam which was passing through this pipe, it being simply mentioned that the "drip valve was kept partially opened." The quantity of steam passing through the pipe has an important bearing upon results of such tests. No mention is made of the condition of the calorimeters themselves, whether they were covered or uncovered.

On the whole, these experiments seem to be worse than valueless, for they pretend to give information on matters of calorimeter work which will be accepted as authoritative by persons who have not the means of analyzing the results, while, in reality, they give no trustworthy information whatever. If Prof. Carpenter really desires to have tests on this subject reported in the *Transactions* of the Society he should repeat the experiments *himself* by making longer and more careful runs, remedying the defects of management which are here referred to, and report the work in such shape that all the points can be understood.

I might go on and criticise further the later portions of this

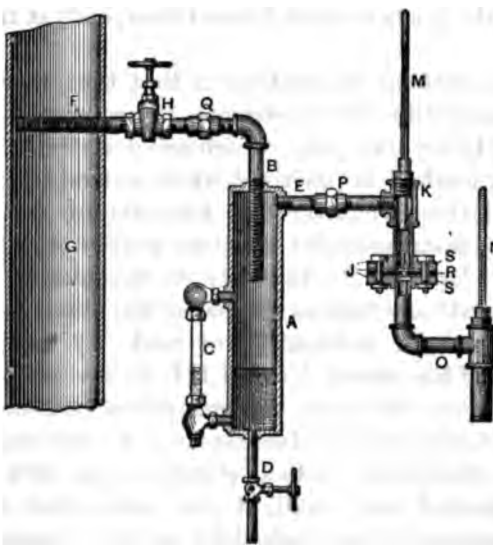


FIG. 317.*

paper, but with so much in the first part which is objectionable it seems hardly worth while to discuss the remaining portions. I cannot, however, refrain from calling attention to the close resemblance between the instrument which Prof. Carpenter devised, a cut of which is given in Fig. 249, as also the separating device, which is given in Fig. 252, to parts of the "Universal Calorimeter," which I devised in September, 1888, and which was fully described in the *Transactions* of the Society, pages 205, 206, Vol. XI., 1890, and pages 790-827 of the same volume. The resemblance is partly in the form of the instrument and partly in the ideas involved. A cut of this instrument is repro-

* Cut taken from page 206, Vol. XI., Trans. A. S. M. E.

duced from the paper referred to and given below for convenience of reference.

I would criticise Prof. Carpenter's throttling instrument in providing no means for preventing the conduction of heat from the metal surrounding the high-pressure steam and that surrounding the low-pressure steam. This is an important point, and it tends to make the indication of the thermometer higher than it ought to be, and, consequently, the percentage of moisture lower than the actual. The orifice in the instrument is placed in a vertical plate, and there is opportunity for water to collect in the pocket which forms below, and, at times, vitiate the results.

I must also criticise the statement that the error caused by radiation is very difficult to estimate. For practical work this is not true. In my own tests, which are described in the paper referred to, a method is practised which entirely overcomes the effect of radiation, whatever this amounts to, and makes the reading of the instrument for moisture absolutely correct.

*Prof. R. C. Carpenter.**—In reply to the inquiry from Prof. Peabody, I would say that, as the paper was about to go to press, a formal demand for indemnity was made by the patentee for infringement of his patent No. 401,111, by the use of the throttling calorimeter of Heisler's design, and on various other forms used by the Sibley College laboratory. An examination of the patent claim elicited the facts as stated on page 832, and, as the amount demanded was small, it was paid, after considerable delay and discussion, and such right as the patentee may have rightly owned to use of the patent was obtained. It seemed to the author that this patent claim was a matter of vital importance in relation to the early invention of the throttling calorimeter, and, besides, it seems but justice that the public should know of the nature of the lien existing on the use of this simple instrument.

In regard to the discussion by Mr. Barrus, he has taken considerable space to throw doubt on a series of observations, and he makes them appear doubtful by assuming some circumstances which did not exist, and which had nothing to do with the experiment in question. The simple facts are that the nearest calorimeter to the boiler was more than 200 feet away, that the farthest one was nearest a drip, which acted in a

* Author's Closure.

measure as a steam separator. Taking these circumstances into account, the graphical chart which has been carefully made shows every observation, with one exception, as *accordant* and *reasonable*.

The single non-accordant observation would have been omitted by many observers, but was published, as all non-accordant observations should be, to allow individual readers to form their own conclusion. The reason the method used by Mr. Barrus to determine the radiation from the calorimeter was not used is because, in my opinion, his method is erroneous, as it involves the assumption that steam in a long pipe not in motion is perfectly dry and not affected by condensation, and, second, that the radiation correction for the instrument is the same during the whole test. I fail to see any method at all in such a proceeding, and it strikes me that it is little better than a guess. I do, however, believe that the method proposed by Prof. Peabody will give accurate results.

I am somewhat surprised at the statement regarding the use of cylinder oil in thermometer cups, and can only say that if he will put some water in a thermometer cup, under such conditions, and insert it in high-pressure steam, he will obtain some experience that will be valuable even to an expert; it will not be necessary to use a leaky cup either.

In regard to the resemblance of the calorimeter shown in Fig. 249 to the patented one shown in Fig. 317, I have reproduced here the drawing of the original Peabody calorimeters, as given in the *Journal of the Franklin Institute* for August, 1888. See Fig. 333. The resemblance between the instrument shown in Fig. 249 and the original shown in Fig. 333 is striking, the principal change being as regards the size of the chamber *A*, in the use of a mercury manometer instead of a steam-gauge, and in the use of a standard orifice, by means of which the throttling is done.

The most striking resemblance in form is, however, shown between Mr. Barrus' calorimeter, as shown in Fig. 317, and the original calorimeter, Fig. 333.

It is seen that the original calorimeter was provided with a drip, which we know now would have the same effect in removing the moisture as the especial drip box of Mr. Barrus' calorimeter.

The instrument lacks only one important accessory—a gauge

duced from the paper referred to and given below for convenience of reference.

I would criticise Prof. Carpenter's throttling instrument in providing no means for preventing the conduction of heat from the metal surrounding the high-pressure steam and that surrounding the low-pressure steam. This is an important point, and it tends to make the indication of the thermometer higher than it ought to be, and, consequently, the percentage of moisture lower than the actual. The orifice in the instrument is placed in a vertical plate, and there is opportunity for water to collect in the pocket which forms below, and, at times, vitiate the results.

I must also criticise the statement that the error caused by radiation is very difficult to estimate. For practical work this is not true. In my own tests, which are described in the paper referred to, a method is practised which entirely overcomes the effect of radiation, whatever this amounts to, and makes the reading of the instrument for moisture absolutely correct.

*Prof. R. C. Carpenter.**—In reply to the inquiry from Prof. Peabody, I would say that, as the paper was about to go to press, a formal demand for indemnity was made by the patentee for infringement of his patent No. 401,111, by the use of the throttling calorimeter of Heisler's design, and on various other forms used by the Sibley College laboratory. An examination of the patent claim elicited the facts as stated on page 832, and, as the amount demanded was small, it was paid, after considerable delay and discussion, and such right as the patentee may have rightly owned to use of the patent was obtained. It seemed to the author that this patent claim was a matter of vital importance in relation to the early invention of the throttling calorimeter, and, besides, it seems but justice that the public should know of the nature of the lien existing on the use of this simple instrument.

In regard to the discussion by Mr. Barrus, he has taken considerable space to throw doubt on a series of observations, and he makes them appear doubtful by assuming some circumstances which did not exist, and which had nothing to do with the experiment in question. The simple facts are that the nearest calorimeter to the boiler was more than 200 feet away, that the farthest one was nearest a drip, which acted in a

* Author's Closure

measure as a steam separator. Taking these circumstances into account, the graphical chart which has been carefully made shows every observation, with one exception, as *accordant* and *reasonable*.

The single non-accordant observation would have been omitted by many observers, but was published, as all non-accordant observations should be, to allow individual readers to form their own conclusion. The reason the method used by Mr. Barrus to determine the radiation from the calorimeter was not used is because, in my opinion, his method is erroneous, as it involves the assumption that steam in a long pipe not in motion is perfectly dry and not affected by condensation, and, second, that the radiation correction for the instrument is the same during the whole test. I fail to see any method at all in such a proceeding, and it strikes me that it is little better than a guess. I do, however, believe that the method proposed by Prof. Peabody will give accurate results.

I am somewhat surprised at the statement regarding the use of cylinder oil in thermometer cups, and can only say that if he will put some water in a thermometer cup, under such conditions, and insert it in high-pressure steam, he will obtain some experience that will be valuable even to an expert; it will not be necessary to use a leaky cup either.

In regard to the resemblance of the calorimeter shown in Fig. 249 to the patented one shown in Fig. 317, I have reproduced here the drawing of the original Peabody calorimeters, as given in the *Journal of the Franklin Institute* for August, 1888. See Fig. 333. The resemblance between the instrument shown in Fig. 249 and the original shown in Fig. 333 is striking, the principal change being as regards the size of the chamber A, in the use of a mercury manometer instead of a steam-gauge, and in the use of a standard orifice, by means of which the throttling is done.

The most striking resemblance in form is, however, shown between Mr. Barrus' calorimeter, as shown in Fig. 317, and the original calorimeter, Fig. 333.

It is seen that the original calorimeter was provided with a drip, which we know now would have the same effect in removing the moisture as the especial drip box of Mr. Barrus' calorimeter.

The instrument lacks only one important accessory—a gauge glass; and there is strong evidence that the drip pipe C was used as a separator to remove some of the water from the steam before the patent had been granted to Mr. Barrus. It is much more accurate than the calorimeter of Mr. Barrus, as it is provided with an exhaust pressure gauge.

The form Fig. 249 is criticised because of the absence of a heat-insulating device. This change is, in my opinion, a very decided improvement, and was omitted after a series of trials with the Heisler instrument. That there is a flow of heat along

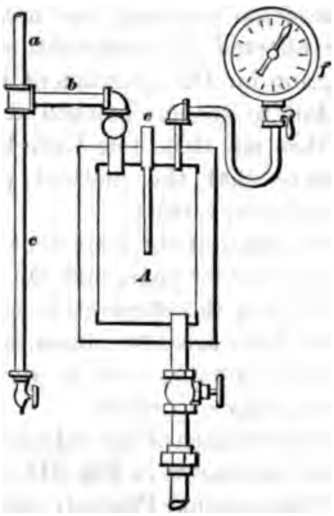


FIG. 333.

Peabody Throttling Calorimeter. Cut from *Journal of Franklin Institute*, August, 1888.

the walls sufficient to affect a thermometer immersed in steam in the interior does not appear in any experiments we could make, nor is it a reasonable thing to suppose.

The cuts of the nipples used to receive the steam, as given by Mr. Barrus, Fig. 313 and Fig. 314, show very clearly the superiority of the nipple with a few large holes, but Mr. Barrus fails to state that the reason for this fact is that the nipple with small holes has an action analogous to that of the steam separator, and, consequently, will not deliver a fair sample of steam.

CCCCLIII.*

THE EFFECT OF THE STEAM-JACKET ON CYLINDER CONDENSATION.

BY W. W. BIRD, CAMBRIDGEPORT, MASS.
(Junior Member of the Society.)

THE object of this paper is to show something of what is being done at the Mechanical Laboratory of the Worcester Polytechnic Institute, in regard to the effect of the steam-jacket on cylinder condensation and engine economy, and to add, if possible, something to the general information on these subjects.

The work here presented was done to a great extent by Mr. F. A. Gardner, of the class of 1890, and presented by him as his graduating thesis. The experiments were made on the upright compound engine at the Institute, a further description of which is given at the end of this paper, and the results and conclusions of these experiments are for this engine only.

It is a generally accepted fact that, aside from radiation, all the heat lost in a steam-engine is what goes out at exhaust, and, as a certain amount must go out, the real loss is the difference between what does and what should go out. Cylinder condensation is the great cause of this loss, as heat is taken up at a high temperature and given back at a lower, when the opportunity for useful work has nearly if not altogether passed. As all the heat taken up in the cylinder from the entering steam must be returned during the cycle, excepting what is lost by radiation, it is not the loss of heat as heat, but the loss of the opportunity of converting heat into work which affects so seriously the efficiency of the engine.

With these facts in mind it was proposed to investigate the action of the steam in the cylinder in regard to the exchanges of heat: the effect of the steam-jacket, and to see if it was a benefit, and if so, why.

Judging from the work done thus far, we conclude that for this engine the steam-jacket is a benefit, the steam saved more than

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. of the *Transactions.*

duced from the paper referred to and given below for convenience of reference.

I would criticise Prof. Carpenter's throttling instrument in providing no means for preventing the conduction of heat from the metal surrounding the high-pressure steam and that surrounding the low-pressure steam. This is an important point, and it tends to make the indication of the thermometer higher than it ought to be, and, consequently, the percentage of moisture lower than the actual. The orifice in the instrument is placed in a vertical plate, and there is opportunity for water to collect in the pocket which forms below, and, at times, vitiate the results.

I must also criticise the statement that the error caused by radiation is very difficult to estimate. For practical work this is not true. In my own tests, which are described in the paper referred to, a method is practised which entirely overcomes the effect of radiation, whatever this amounts to, and makes the reading of the instrument for moisture absolutely correct.

*Prof. R. C. Carpenter.**—In reply to the inquiry from Prof. Peabody, I would say that, as the paper was about to go to press, a formal demand for indemnity was made by the patentee for infringement of his patent No. 401,111, by the use of the throttling calorimeter of Heisler's design, and on various other forms used by the Sibley College laboratory. An examination of the patent claim elicited the facts as stated on page 832, and, as the amount demanded was small, it was paid, after considerable delay and discussion, and such right as the patentee may have rightly owned to use of the patent was obtained. It seemed to the author that this patent claim was a matter of vital importance in relation to the early invention of the throttling calorimeter, and, besides, it seems but justice that the public should know of the nature of the lien existing on the use of this simple instrument.

In regard to the discussion by Mr. Barrus, he has taken considerable space to throw doubt on a series of observations, and he makes them appear doubtful by assuming some circumstances which did not exist, and which had nothing to do with the experiment in question. The simple facts are that the nearest calorimeter to the boiler was more than 200 feet away, that the farthest one was nearest a drip, which acted in a

* Author's Closure

CCCCLIII.*

THE EFFECT OF THE STEAM-JACKET ON CYLINDER CONDENSATION.

BY W. W. BIRD, CAMBRIDGEPORT, MASS.
(Junior Member of the Society.)

THE object of this paper is to show something of what is being done at the Mechanical Laboratory of the Worcester Polytechnic Institute, in regard to the effect of the steam-jacket on cylinder condensation and engine economy, and to add, if possible, something to the general information on these subjects.

The work here presented was done to a great extent by Mr. F. A. Gardner, of the class of 1890, and presented by him as his graduating thesis. The experiments were made on the upright compound engine at the Institute, a further description of which is given at the end of this paper, and the results and conclusions of these experiments are for this engine only.

It is a generally accepted fact that, aside from radiation, all the heat lost in a steam-engine is what goes out at exhaust, and, as a certain amount must go out, the real loss is the difference between what does and what should go out. Cylinder condensation is the great cause of this loss, as heat is taken up at a high temperature and given back at a lower, when the opportunity for useful work has nearly if not altogether passed. As all the heat taken up in the cylinder from the entering steam must be returned during the cycle, excepting what is lost by radiation, it is not the loss of heat as heat, but the loss of the opportunity of converting heat into work which affects so seriously the efficiency of the engine.

With these facts in mind it was proposed to investigate the action of the steam in the cylinder in regard to the exchanges of heat: the effect of the steam-jacket, and to see if it was a benefit, and if so, why.

Judging from the work done thus far, we conclude that for this engine the steam-jacket is a benefit, the steam saved more than

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. of the *Transactions*.

making up for that used in the jacket directly. The surfaces of the cylinder are warmer when the jacket is used, thus decreasing condensation to begin with; the steam is dryer at the beginning of exhaust, so that less heat is given up during that period. The steam is probably not homogeneous in the cylinder; the condensation takes place where it is in contact with the surfaces, forming a film of moist steam, which takes up the heat returned after reëvaporation begins, and the more moisture in the steam the more heat there is taken from the surfaces. And since most of the heat is returned during exhaust, the condition of the steam at the beginning of that period is of great importance in determining the amount of heat to go out and the efficiency of the engine.

The accompanying diagram (Fig. 257) is presented to show

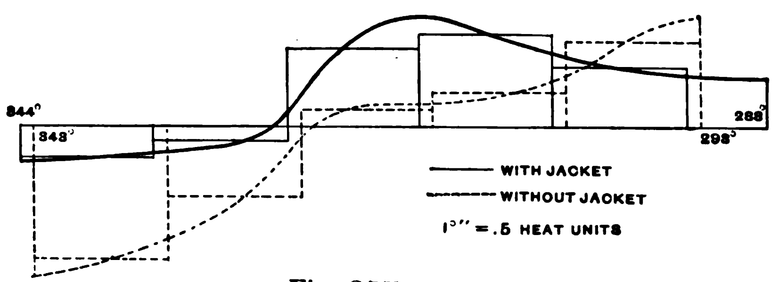
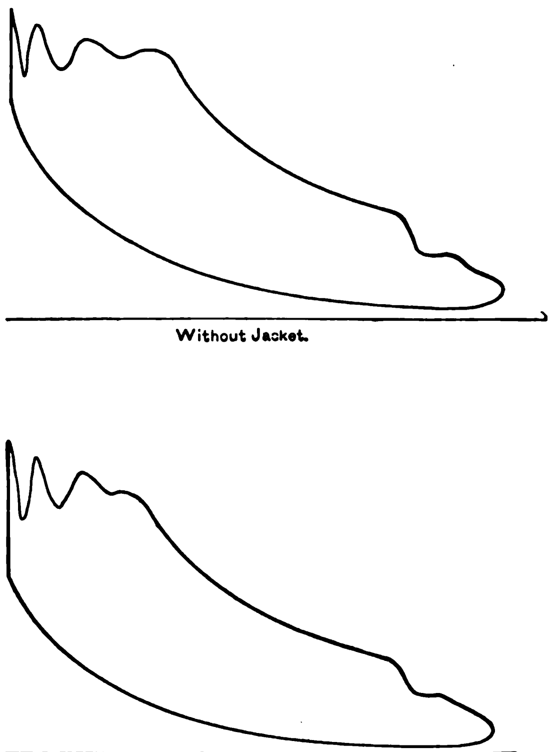


Fig. 257.

graphically the exchanges of heat between cut-off and exhaust, or during the expansion in the high-pressure cylinder. The areas below the base line show heat taken up after cut-off and before reëvaporation begins; the areas above, the heat given back. Horizontal distances represent temperatures, and vertical distances entropy. Thus, if there were no exchange of heat, the curve would coincide with the base line, and the entropy would be constant as it should be with adiabatic expansion.

The method of constructing the diagram is as follows: The range of temperature between cut-off and exhaust determines the length of base line, which is divided into several parts, and corresponding divisions made on the indicator card by taking the different pressures corresponding to the several temperatures. The heat accounted for at each of these temperatures, together with the heat equivalent of the external work of each division, gives the required data for finding the exchange of heat during each period, which is shown by the rectangles, the curve being sketched in afterward to enclose equal areas.

It will be seen by this and Table III. that the work is the same as in Hirn's analysis, only applied at more points in the cycle. To obtain the weight of steam and water in the cylinder during expansion the steam was taken as dry in each case at the end of



With Jacket.
80 Pound Spring
Clearance 38. 31. cu. in

Fig. 258.

compression, and, as a separator was used, the steam supplied was also taken as dry. These assumptions do not affect the curves relatively to any amount.

The diagram in this case shows that the curve with the jacket is nearer to a straight line than when unjacketed, therefore nearer to true adiabatic expansion, although heat is being received all the time from the steam in the jacket. It also shows that there is less condensation after cut-off when the jacket is used; that re-evaporation begins earlier, after which heat is rapidly returned, thus giving it more chance for useful work; that the amount returned decreases as the steam becomes dryer, so that less goes out during exhaust.

Without the jacket, after reëvaporation begins, the heat is given back apparently quite slowly; this is probably due to the steam coming in contact with new surfaces, as the piston advances, which are cooler; however, the rate increases and is greatest at the beginning of exhaust, and the steam is still moist enough to continue to take heat rapidly during this period, thus cooling the surfaces much more than in the other case.

These curves might be continued so as to show the action during the time the two cylinders are in communication or when the high-pressure is exhausting, and then during compression. Sufficient has been given to show the method. At some other time, it is hoped to present diagrams showing the effect of time on cylinder condensation and also the effect of superheated steam. Different combinations of these with and without the steam-jacket will give variety of work and doubtless lead to some valuable information. Fig. 258 shows reproductions of the cards taken with and without the jacket.

TABLE I.

	Average Boiler Press. Gauge.	Revolutions per Minute.	Average Cut-off.	Indicated Horse Power.	Steam per Hour to Engine.	Steam per Hour to Jacket.	Total.	Pounds per Indicated Horse-Power per Hour.
With Jacket....	120	288	.262	10.57	180	12	192	18.16
Without Jacket..	120	289	.389	10.53	224		224	21.27

Gain by jacketing, 14.6%.

TABLE II.

	Steam Admitted per Stroke. Pounds.	Steam Saved by Compression.	Total Weight at Cut-off.	Per Cent. of Steam at Cut-off.	Heat in Steam at End of Compression.	Heat Received per Stroke.	Heat Equivalent of Work Done Up to Cut-off.
With Jacket.....	.0104	.0036	.0140	87.59	3.9666	12.3577	.6140
Without Jacket..	.0129	.0039	.0168	84.60	4.2911	15.3280	.8007

	Total Heat Accounted for at Cut-off.	Actual Heat in Steam at Cut-off.	Heat Taken Up by Surfaces.	Per Cent. of Steam at Exhaust.	Heat in the Steam at Exhaust.	Heat Equivalent of External Work, Cut-off to Exhaust.	Total Heat Accounted for at Exhaust.	Heat Given Up by Surfaces and Jacket.
With Jacket. ...	15.7104	14.1120	1.5983	89.97	14.1291	.7791	13.3329	.7961
Without Jacket..	18.8184	16.5223	2.2961	80.00	15.5784	.6810	15.8413	−.2628

TABLE III.

	Range of Temperature.	Total Heat at Higher Temperature.	Per Cent. of Steam in Mixture at Lower Temperature.	Total Heat at Lower Temperature.	Heat Equiv. of External Works during the Range.	Total Heat Accounted for at Lower Temperature.	Heat to or from Surfaces and Jackets.
With Jacket.....	344.334	14.1120	85.93	13.8785	.1192	13.9928	−.1143
" "	334.324	13.8785	84.85	13.7063	.1173	13.7612	−.0529
" "	324.314	13.7063	86.54	13.8502	.1470	13.5613	+.2889
" "	314.304	13.8502	88.54	14.0334	.1576	13.6925	+.3408
" "	304.294	14.0334	89.53	14.1038	.1498	13.8837	+.2202
" "	294.288	14.1038	89.97	14.1291	.0881	14.0157	+.1134
Without Jacket..	343.333	16.5223	80.54	15.9176	.1020	16.4203	−.5027
" " ..	333.323	15.9176	78.27	15.5429	.1099	15.8077	−.2647
" " ..	323.313	15.5429	78.17	15.4595	.1409	15.4020	+.0576
" " ..	313.303	15.4595	78.41	15.4264	.1605	15.2991	+.1273
" " ..	303.293	15.4264	80.00	15.5784	.1679	15.2586	+.3198

The engine is an upright single-acting compound, $5\frac{1}{4}$ inch and 9 inch cylinders, 8 inch stroke, and about 10 I. H. P. It was built for the Mechanical Engineering Department of the Worcester Polytechnic Institute, and was installed in the Salisbury Laboratories in April, 1889. The engine is shown in vertical section by Fig. 259, and has the following peculiar features:

I. The cylinders are side by side, but the high-pressure cylinder has its lower end just below the upper end of the low-pressure cylinder, so as to give a direct passage across from one cylinder to the other.

II. The cranks are set 120° apart, the low-pressure crank being 120° behind the other. By this arrangement the high-pressure piston acts as a valve between the two cylinders, and is the only valve required between them.

III. The exhaust port is at the lower end of the L. P. cylinder, and the L. P. piston takes the place of an exhaust valve. By this arrangement the exhaust is performed, not in the usual way, by forcing the steam out during the larger part of the stroke, but by allowing the pressure in the cylinder to drop to that in the condenser. Early on the return stroke the exhaust closes, and the

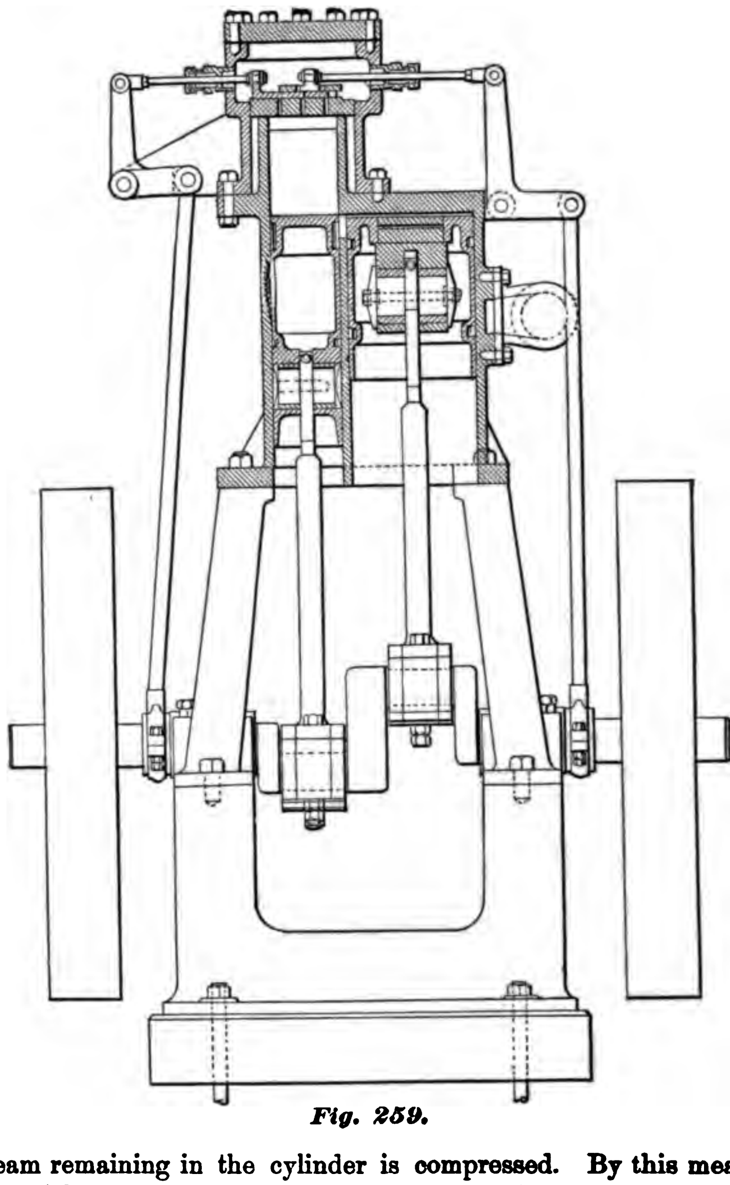


Fig. 259.

steam remaining in the cylinder is compressed. By this means the work, otherwise done upon the fluid in the condenser during exhaust, is retained in the engine as heat or as potential energy of the compressed steam.

IV. If the compression in the high-pressure cylinder were carried to initial pressure, there would be no loss in that cylinder from clearance on account of unused compressions, as the compressed steam would expand back to the same pressure from which compression begins. The cut referred to does not show the governor, which is an ordinary shaft governor, controlling the eccentric which operates the cut-off valve. The speed at which the engine has been driven thus far is about 290 revolutions per minute. The high-pressure cylinder is provided with a steam-jacket. It is clear that the engine is arranged for a high degree of compression in both cylinders.

DISCUSSION.

Prof. R. H. Thurston.—This paper contains two or three especially interesting points. It illustrates the fact that the sometimes disparaged analysis of Hirn and of Dwelshauvers-Dery is actually coming into general use in this country—for the papers of Professors Carpenter and Peabody also illustrate its application—and shows how well even the incomplete analysis as yet found practicable may be made to throw light on the workings of the steam engine. I note that the steam is *assumed* to be dry at compression. This corresponds with Hirn's deduction, and is confirmed by the direct experiments now frequently made in the laboratories of Sibley College on a small engine, showing by calorimetric test that this is true. If true on so small an engine—5 H. P. maximum—it may be taken as true for all ordinary practice. Prof. Carpenter explains this fully in his paper on Hirn's analysis, now before the society. Mr. Bird's assumption may thus, I think, be accepted as correct. His assumption of dryness at the commencement may be less valid; but if the separator was of good form and fairly large, it would be right. We find, in our experiments especially directed to this point, that, with ample capacity, we can insure sensibly "dry and saturated" steam, especially if a good trap or steam-loop gives insurance against intermittent priming. The facts here shown are already familiar in a general way; but it is interesting to see how well the entrophy-temperature diagram exhibits the method of action of the cylinder walls, with and without a jacket. It is easy to see why it is that we invariably find that a well-made jacket, under favorable conditions of action of engine, as to dryness of steam especially, saves more heat than it wastes, and how superheating a few degrees may save much more than the quantity of heat thus expended, and that the quantity lost by initial condensation may be reduced by a fraction of the same amount of heat expended in superheating. The promised work of Mr. Bird will be still more interesting, and I have no doubt even more instructive than this.

One fact of prime importance and interest in this case, as it seems to me, is the saving by jacketing on an engine making 290 revolutions per minute. It has been the general impression among engineers—an impression which I myself shared—that the high-speed engine, or any engine at moderately high speed of revolution, could not be expected to exhibit much gain by jacketing; though it is, I presume, a well-understood fact, that the jacket should, if equally well designed and operated, be more effective as the size of engine decreases, and, if its action is proportional to area covered and inversely as volume of steam acted on, in the inverse proportion of diameter of cylinder for similar forms of engine.

Taking the wastes of jacketed engines as varying inversely as the square root of speed of revolution, inversely as content of cylinder, and directly as area of internal surfaces and the action of the jacket as governed by similar laws, we should have in this engine the computed quantity very nearly that here given by test. The jacket saves the excess of wastes due to reduced size of engine, and it thus evidently tends to bring all sizes and speeds of engine to a common and high efficiency. This engine is itself a remarkably fine piece of designing, and its performance is most extraordinarily high for so small a machine. All things considered, it is, I am inclined to think, the best as yet recorded. I suspect that the secrets of its success are its high compression in both cylinders, and the peculiar arrangement for the avoidance of heat-transfer between steam and metal, for which Prof. Alden may, I think, claim great credit.

It is worthy of notice that here we have a supply of 6% of the steam by way of the jacket, giving a saving of 14% or more.

Prof. James E. Denton.—I think, with Prof. Thurston, that this instance of the action of a jacket is worth noting with care. Hearing of this case, I went to Worcester to examine the engine. It is a peculiar engine, in the arrangement of cylinders so very closely together, in its excessive cushion, and its separate exhaust and inlet parts, and we have yet to find parallel conditions in practice. But before we conclude that there is a useful discovery here regarding the value of jacketing, which can be used practically with large engines, we shall have to explain several discrepancies in the case of other tests where there has been no saving, such as are shown to have occurred in this case in the high cylinder. The Reynolds tests in England, on a triple expansion engine, which excited so much notice about three years ago, were made on an engine having small cylinders, and the saving due to jacketing was entirely due to the intermediate and low cylinders. There was a loss due to the jacket in the high-pressure cylinder by cylinder condensation; that is, there was more cylinder steam entered the high cylinder with the jacket than there was without it; more steam went through the engine; but there was a gain in M. E. P. in the intermediate and low cylinders. It is yet to be shown that the same combination as described embodied in a larger engine will have the same effect. The low-pressure cards should be added to make the data complete.

Prof. Geo. I. Alden.—What I wish to present does not relate particularly to the effect of jacketing, although, since more tests have been run since Mr. Bird prepared this paper, we are now quite sure that the jacket does save heat in this particular engine. As Mr. Bird has outlined a method which shows graphically for the high-pressure expansion curve the effect of the interchange of heat between the cylinder and the fluid, I wish to present a graphical method, which, in its study, gives an idea of what goes on in the cylinder, and which represents the interchanges of heat, and also enables one from the diagram to ascertain, in so far as the diagram is correct, the exact exchange of heat between the surfaces and the fluid for any part of the stroke for which the diagram is completed.

Mr. Bird's method of tracing the effects of cylinder condensation is a very accurate and complete solution of the problem undertaken, and, if applied to the complete cycle of the engine, would leave nothing to be desired in the way of completeness, clearness, and correctness. One great difficulty in applying this or any process to the expansion curve of the H.P. cylinder, and to the compression curves, is the impossibility of securing with absolute certainty the necessary data.

It is impossible to determine exactly the quality of the steam in the clearances of an engine without heat-conducting surfaces, that the quantity lost by initial condensation may be reduced by a fraction of the same amount of heat expended in superheating. The promised work of Mr. Bird will be still more interesting, and I have no doubt even more instructive than this.

One fact of prime importance and interest in this case, as it seems to me, is the saving by jacketing on an engine making 290 revolutions per minute. It has been the general impression among engineers—an impression which I myself shared—that the high-speed engine, or any engine at moderately high speed of revolution, could not be expected to exhibit much gain by jacketing; though it is, I presume, a well-understood fact, that the jacket should, if equally well designed and operated, be more effective as the size of engine decreases, and, if its action is proportional to area covered and inversely as volume of steam acted on, in the inverse proportion of diameter of cylinder for similar forms of engine.

Taking the wastes of jacketed engines as varying inversely as the square root of speed of revolution, inversely as content of cylinder, and directly as area of internal surfaces and the action of the jacket as governed by similar laws, we should have in this engine the computed quantity very nearly that here given by test. The jacket saves the excess of wastes due to reduced size of engine, and it thus evidently tends to bring all sizes and speeds of engine to a common and high efficiency. This engine is itself a remarkably fine piece of designing, and its performance is most extraordinarily high for so small a machine. All things considered, it is, I am inclined to think, the best as yet recorded. I suspect that the secrets of its success are its high compression in both cylinders, and the peculiar arrangement for the avoidance of heat-transfer between steam and metal, for which Prof. Alden may, I think, claim great credit.

It is worthy of notice that here we have a supply of 6% of the steam by way of the jacket, giving a saving of 14% or more.

Prof. James E. Denton.—I think, with Prof. Thurston, that this instance of the action of a jacket is worth noting with care. Hearing of this case, I went to Worcester to examine the engine. It is a peculiar engine, in the arrangement of cylinders so very closely together, in its excessive cushion, and its separate exhaust and inlet parts, and we have yet to find parallel conditions in practice. But before we conclude that there is a useful discovery here regarding the value of jacketing, which can be used practically with large engines, we shall have to explain several discrepancies in the case of other tests where there has been no saving, such as are shown to have occurred in this case in the high cylinder. The Reynolds tests in England, on a triple expansion engine, which excited so much notice about three years ago, were made on an engine having small cylinders, and the saving due to jacketing was entirely due to the intermediate and low cylinders. There was a loss due to the jacket in the high-pressure cylinder by cylinder condensation; that is, there was more cylinder steam entered the high cylinder with the jacket than there was without it; more steam went through the engine; but there was a gain in M. E. P. in the intermediate and low cylinders. It is yet to be shown that the same combination as described embodied in a larger engine will have the same effect. The low-pressure cards should be added to make the data complete.

Prof. Geo. I. Alden.—What I wish to present does not relate particularly to the effect of jacketing, although, since more tests have been run since Mr. Bird prepared this paper, we are now quite sure that the jacket does save heat in this particular engine. As Mr. Bird has outlined a method which shows graphically for the high-pressure expansion curve the effect of the interchange of heat between the cylinder and the fluid, I wish to present a graphical method, which, in its study, gives an idea of what goes on in the cylinder, and which represents the interchanges of heat, and also enables one from the diagram to ascertain, in so far as the diagram is correct, the exact exchange of heat between the surfaces and the fluid for any part of the stroke for which the diagram is completed.

Mr. Bird's method of tracing the effects of cylinder condensation is a very accurate and complete solution of the problem undertaken, and, if applied to the complete cycle of the engine, would leave nothing to be desired in the way of completeness, clearness, and correctness. One great difficulty in applying this or any process to the expansion curve of the H.P. cylinder, and to the compression curves, is the impossibility of securing with absolute certainty the necessary data.

It is impossible to determine exactly the quality of the steam in the clearances of an engine without heat-conducting surfaces, and, therefore, impossible to know exactly the weight of steam in the engine at any given piston position. It is quite certain that in most engines the quality of the steam in the clearances at the end of compression would be different in the actual and theoretical engines.

To carry on any investigation which has for its object to show the transference of heat between the cylinder surfaces and the working fluid, certain assumptions must be made, and if there is any way to check the correctness of these assumptions, such checks increase the value of the work. For the engine under consideration, the writer has attempted to give a graphical solution, which is presented in outline for the purpose of suggesting the method.

The indicator cards were taken during a trial made at the Salisbury Laboratories, by Messrs. Somerset and Alley, of the senior class at the Worcester Polytechnic Institute, and the diagrams in the accompanying plate were made by Mr. Somerset. They are not given on account of their completeness, but only to suggest the possible value of the method, which will be understood so that it may be applied to other engines, by any one familiar with steam tables, from the following outline of its application to the engine described by Mr. Bird.

First. Compute the weight of steam in the high-pressure cylinder at cut-off, assuming for this purpose the quality in clearance, and, if separator is used, assuming the entering steam dry.

Second. Assume or compute the quality of this steam at cut-off, supposing it to be working in an engine with non-conducting surfaces.

Third. Construct adiabatic expansion and compression lines for the high-pressure cylinder.

Fourth. Compute the weight of steam in the low-pressure clearance at the beginning of low-pressure compression, estimating its quality from such considerations and data as the engine and experiment may furnish.

Fifth. Construct the adiabatic compression line for the low-pressure cylinder.

Sixth. Having now the pressures and qualities of two masses of steam about to mix as the high-pressure cylinder opens into the low, compute the resulting pressure and quality of the mixture by the usual formula.

33

ıe

ıs

·l-
›n
k,
K
ıe
ıe
es
ıe

oe
rk
ıe-
be
al

ne
he

he
of
ıs,
of

li-

at
he
al
ce
er
nd
ne
he
en
he

he

Seventh. Construct now the adiabatic expansion line for the low-pressure cylinder.

Eighth. Next search for some check upon the assumptions thus far made.

In the present case, compression in the high-pressure cylinder begins when the low-pressure piston is at the position corresponding to the 120° crank-angle of the low-pressure crank, so that the point where the high-pressure compression line JK (Fig. 318) cuts the line CJ should theoretically be at the same distance from the zero line of pressures as the point where the low-pressure expansion DE curve cuts the line EH, as ordinates at these points theoretically represent the pressures in the same mass of steam at the same instant.

If this result is reached by the first trial, the work may be regarded as substantially correct. If not, a study of the work already done will very likely reveal the point where an erroneous assumption has been made, and, by trials, the work will be made to check, as above indicated. When it does the principal lines of the theoretical card are drawn.

Ninth. Plot the real indicator cards, referred to the same absolute pressure and clearance lines and to same scale as the theoretical cards have been constructed.

Tenth. Compute the total heat per pound of the steam in the theoretical and in the actual engine at any desired number of piston positions, and erect ordinates through these positions, upon which lay off distances proportional to the number of thermal units thus found.

Eleventh. Draw lines through the extremities of these ordinates, which curves may be called heat lines.

It is obvious that the vertical distance between the heat lines for a given expansion of compression curve measures the difference in total heat per pound of steam in the theoretical and the actual engine, and that the reason for this difference must be the effect of the heat-conducting surfaces of the cylinder and other parts coming in contact with the steam. To find approximately the number of heat units transferred from one medium to the other, between any two piston positions, take the difference of distances between the heat lines for the given piston positions, and multiply this distance by the scale of the ordinates and the weight in pounds of the expanding steam.

If this result is correct by adding the heat equivalent to the difference of work on the adiabatic and on the actual cards, between the two piston positions in question, the result will be the actual transfer taking place between the surfaces and the steam while the piston moves from the first to the second of the positions assumed. To illustrate the application of this method, the diagram is given.

A B C D E F G H I K is the theoretical card constructed by the method outlined above, all the lines being true adiabatics. *a b c d e f g h i k* is the indicator card. A_1 B_1 is the heat line for A B, and a_1 b_1 is the heat line for a b. Points on the part a_1 b_1 of this line cannot be constructed unless the rate of initial condensation is known, so that this part of the line is drawn straight in this case. The heat lines for each of the curves forming the theoretical and the actual cards are designated by the same letters as the lines on the cards, but with subscripts affixed.

If the steam in the actual engine had done as much work as is shown by the theoretical card, it would have had less heat per pound. The dotted lines b_1 c_2 d_2 e_2 show the correction to be made for difference of work between the theoretical and the actual expansion curves. This dotted line, which may be called for convenience the corrected heat line, is found by reducing the difference of work between the lines of the theoretical and actual cards and any two vertical ordinates to its heat equivalent, and dividing this heat equivalent by the scale of the ordinates and the weight of the expanding steam. For representing graphically the general effect of the heat-conducting surfaces the corrected heat lines need not usually be drawn, or the correction above referred to made. No attempt is made in Plate 1 to show the heat lines during the time the exhaust is open and for a little time just before exhaust opening and closure in the L. P. cylinder.

The steam-jacket on the H. P. cylinder in the engine described in Mr. Bird's paper effects a considerable saving of heat, as shown by repeated tests made under varying conditions. In each comparative test the engine was run under exactly the same brake load and at the same boiler pressure, the only element changed after a trial without the jacket being the admission of steam to the jacket. Comparing the indicator cards with the adiabatics for the same weight of dry or nearly dry steam cut off, it is found that jacketing considerably diminishes initial condensation.

In a series of four tests the average gain in efficiency reckoned from steam saved per I. H. P. per hour was 13.54%.

I might add in regard to the conditions under which the tests showing effect of jacket were made, that, when running without the jacket, the steam was shut off from the jacket and the jacket was in communication with the outer air through a small pipe. When the steam jacket was used, the steam was turned into the jacket direct from the supply pipe to the engine. There was no means of giving the steam circulation in the jacket, as there doubtless should be to get the best effect of the jacket, particularly for a long run.

Mr. John T. Hawkins.—What I have to say on this subject will be more in the nature of an inquiry than anything better. The prevention of loss of energy through the condensation in the cylinder seems to be sought in two opposite directions, or rather efforts are being made to ameliorate this loss in two directions. One is, as we have found in the suggestions of Prof. Thurston, Dr. Emery, and others, to procure in the cylinder non-conducting interior surfaces. This appears to be diametrically opposed to the other remedy, that of steam jacketing; the object of the latter being to maintain such a high temperature of the interior surfaces as shall prevent interchange of heat between the fluid and the cylinder—both of them seeking to minimize this interchange. Now, I have never heard it suggested in these discussions that it would be politic to make the lining or the interior of the cylinder of the best possible conducting material, so that the effect of the steam jacketing might be greater. I can readily conceive as a mechanic how a cylinder might be constructed in which we may have the expansive stresses received upon the jacketing itself, while having a very thin, highly conducting partition between the jacket and the steam by which the heat of the jacket might be conveyed to the working fluid very rapidly. It seems to me that we could thus best realize Watt's object, which was to have the walls of the cylinder kept at a temperature as high as, or higher than that of the entering steam, and there may be something in this idea as a means of reducing the loss of heat. Of course it would be quite antagonistic to attempt to put steam jackets on any sort of engine in which we had been contriving non-conducting interior surfaces. There may be some superior benefit to come from the introduction of a more highly conducting surface in connection with the steam jacket, since we cannot produce absolutely non-conducting surfaces for the interior of the cylinder for use without the jacket. It has been shown in late experiments of M. Dwelshauvers-Dery that the metal of a cast-iron cylinder undergoes fluctuations of temperature greatest at the interior and gradually decreasing toward the jacket to a point somewhere in the metal, where the temperature is constant, and also constant from that point to the surface adjoining the jacket, and that the loss of heat has been least as this neutral point, or point of constant temperature, became nearer the interior surface of the cylinder. It was that very statement of M. Dwelshauvers-Dery that led to the train of thought that, possibly (if it had not already been done), it might be worth some engineer's while to try some experiment in this direction and construct a cylinder which would offer the best and most rapid means of conducting the heat from the jacket to the interior surface.

It was my intention to prepare something of a more definite character in writing, as a discussion of this paper, on the lines I have pointed out; but, owing to the exceedingly short time to be had after receipt of the copies of the papers to be read, I was unable to do so; but I promise myself to prepare something of the kind for a future meeting.

Mr. H. H. Supplee.—In reply to Mr. Hawkins' suggestion: I recollect reading of a quadruple expansion engine built in England, in which the high-pressure cylinder is constructed with a bronze lining, supported at points in the cast-iron cylinder body, and the steam in the jacket is just outside the bronze. The original idea was not so much to make a good conducting cylinder as to avoid the action upon the cast iron which the very high-pressure steam would be apt to produce.

Prof. J. B. Webb.—I think it would be appropriate here to call attention to the difference between a steam turbine and an ordinary reciprocating engine in respect to condensation. In the latter the parts are exposed alternately to high and low temperatures, and in the compound engine that difficulty is somewhat remedied by cutting off the fall of temperature into several different falls; but nothing can avoid that fault in a reciprocating engine. The same metal is during every revolution exposed to that change of temperature. In the turbine it is exactly the opposite. The only loss of heat is the conduction made from one end of the turbine to the other end, which must occur in any metallic engine. So that this is an advantage inherent in the nature of the machine, which places the steam turbine above the reciprocating engine in that respect.

*Mr. W. W. Bird.**—The point of which Prof. Thurston speaks regarding the assumption of dry steam at the end of compression having been confirmed by direct experiments made by Prof. Carpenter, suggests the question whether or not the method employed gives a correct result. The steam in the cylinder is not probably homogeneous, and, consequently, the quality of the sample will depend on the point at which it is taken, and hence may not show the correct result for the steam as a whole.

In reply to Prof. Denton I would say that in the paper particular attention was called to the fact that the results were for this engine only, and nothing is said as to the effect of the jacket in other cases. The method of constructing the diagram would be the same if the jacket had not been a benefit, and would probably show why it was not. Whether or not the steam jacket is of value when all things are considered is a fair question, as is shown by the various results from engine tests. Direct experiment determines the results; further analysis shows the reason for the results. This paper was presented more for the purpose of showing the reasons for the results than the results themselves.

---

* Author's Closure.

CCCCLIV.*

PERFORMANCE OF A STEAM REACTION-WHEEL.

BY J. BURKITT WEBB, HOBOKEN, N. J.
(Member of the Society.)

At the Erie meeting, a short note, "Note on the Steam Turbine," was presented with the object of calling attention to and inviting discussion upon whatever progress is being made in that direction, and in doing so there were described, with the assistance of blackboard sketches, the latest and most successful form of a multiple steam turbine, and also a steam reaction-wheel, or Barker's mill, which did good service many years ago. The latter description was supplemented with an off-hand and very rough calculation of the performance of the wheel and the centrifugal force developed.

Owing to a delay on my part in returning the proof of the discussion and appendix, which I had carefully corrected, the matter went to press as it was. The first two paragraphs of the "Author's Closure" are all that belong to it, and were, with the appendix, written after the meeting. The remaining paragraphs belong to the verbal presentation above alluded to, and should precede the discussion.

The appendix was the result of a request that I would add to the paper an outline of the calculation which was put upon the blackboard. I did so reluctantly. Such rough calculations are of questionable value in print, and this one the more so from the errors remaining in it and the omission of the concluding paragraphs, in which the approximate horse-power was arrived at. To set the matter straight I promised a paper on the more exact calculation of such a wheel, which could not be prepared sooner from lack of time. The paper, with its corrected discussion and appendix, was also published at the time in *The Stevens Indicator*, Vol. VI., 1889, page 288.

In the calculation of a steam reaction-wheel there are two

* To be presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Vol. XII. of the *Transactions*.

things which threaten to make the computation of no value unless they are taken duly into consideration. These are the change in density and wetness of the steam as it escapes through, and the increase of pressure from centrifugal force as it approaches, the orifice. If, however, these things are properly treated, the remaining calculation is quite simple. We will discuss essentially the same problem as before, viz., a 15 H. P. reaction-wheel, of one foot diameter, with a speed of 4,000 revolutions per minute, and a boiler pressure of 70 lbs., or less, absolute.

CENTRIFUGAL FORCE.

To separate these two things, and thus be able to treat each by itself, we will suppose that the pressure of steam is 70 lbs. absolute and the steam saturated, just inside the orifice where the centrifugal force has produced its whole effect. The boiler pressure will then be less than 70 lbs. by the amount due to centrifugal force, and this will not be so great as to require more than a first, or at the most a second, approximation thereto.

Evidently the increase of pressure will be no more than the centrifugal force of a radial column of steam 6 inches long and 1 inch square, with a density at its outer end corresponding to 70 lbs. and at its inner end corresponding to the boiler pressure. For a first approximation suppose the steam to weigh $\frac{1}{6}$ lb. per cubic foot, which corresponds to 70 lbs. pressure, then the whole column, containing $\frac{1}{288}$ of a cubic foot, will weigh $\frac{1}{1728}$ lb., which, being divided by $g = 32.2$, gives .000018 of a unit of mass. For every 1,000 revolutions per minute that the wheel may make, the circumferential velocity will be about $52\frac{1}{2}$ feet and the angular velocity twice that, or 105, and, as the radius to the centre of the column of steam is $\frac{1}{4}$ foot, the centrifugal force of the column of steam is — cent. force = (ang. vel.)^2 × radius × mass = $105^2 \times .25 \times .000018 = \frac{1}{20}$ lb., and this would be somewhat greater than the increase of pressure because the density of the column of steam is less toward the centre.

For greater speeds we should have: For 2,000 per minute, $\frac{4}{20} = \frac{1}{5}$ lb.; for 4,000, $\frac{16}{20}$ or $\frac{4}{5}$ lb.; for 16,000, $\frac{256}{20}$, or say 12 lbs. Therefore at 4,000 revolutions the boiler pressure would not be a pound less than the nozzle pressure, and no more accurate estimate of the centrifugal force is needed, while at 16,000, if the effect of the decreasing density of the column toward the centre is to be allowed for, it will reduce the difference of pressure calculated above about 5%, leaving a difference of over 11 lbs. between the boiler and nozzle pressures.

We come now to a more important part of the calculation:

FLOW OF STEAM THROUGH A NOZZLE.

It is not enough to know from a formula how much steam will escape through a nozzle; the way in which it escapes has an important influence on the power developed.

Let Fig. 261 be a nozzle delivering steam of 70 lbs. pressure. In speaking of this pressure as existing at the entrance to the nozzle, say at the point *a*, a slight reservation must be made, so as to allow for the velocity past that point. This velocity is comparatively small; suppose, for instance, that the cross-section at *a* is ten square inches, while the smallest or "throat" section of the nozzle is one square inch, then about one pound of steam per second will pass *a** and its velocity will be only 86 feet per second, to produce which a fall of pressure of two ounces would be enough, or if the section at *a* was only five square inches there would be required a fall of a half pound to produce the necessary velocity of 172 feet. The 70 lbs. then is 70 lbs. for still steam.

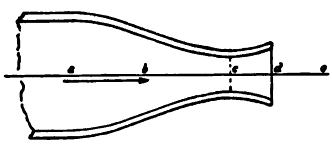


FIG. 261.

No exact solution of the problem can be made until we know the way in which the pressure decreases, as the steam flows from *a* to *e*, at which latter point we shall suppose it to have reached atmospheric pressure. If we had to do with the flow of water, or other non-compressible fluid, the variation of pressure along the route would have no effect upon the result, and we would need to know only the mass of the water delivered per second, and the section of the stream at its point of exit; from these we could at once calculate the momentum per second to which the reaction of the jet is equal. The previous rough calculation was in fact made in this way with an estimated allowance for the increase in velocity due to the expansion of the steam, and as the particular shape of the delivery nozzle was not then supposed to be known, it would not have been possible to arrive at a result without some such assumption. In fact,

* For the weight of steam escaping through a one-inch nozzle at the different pressures, see Rankine's *Steam Engine*, pages |.

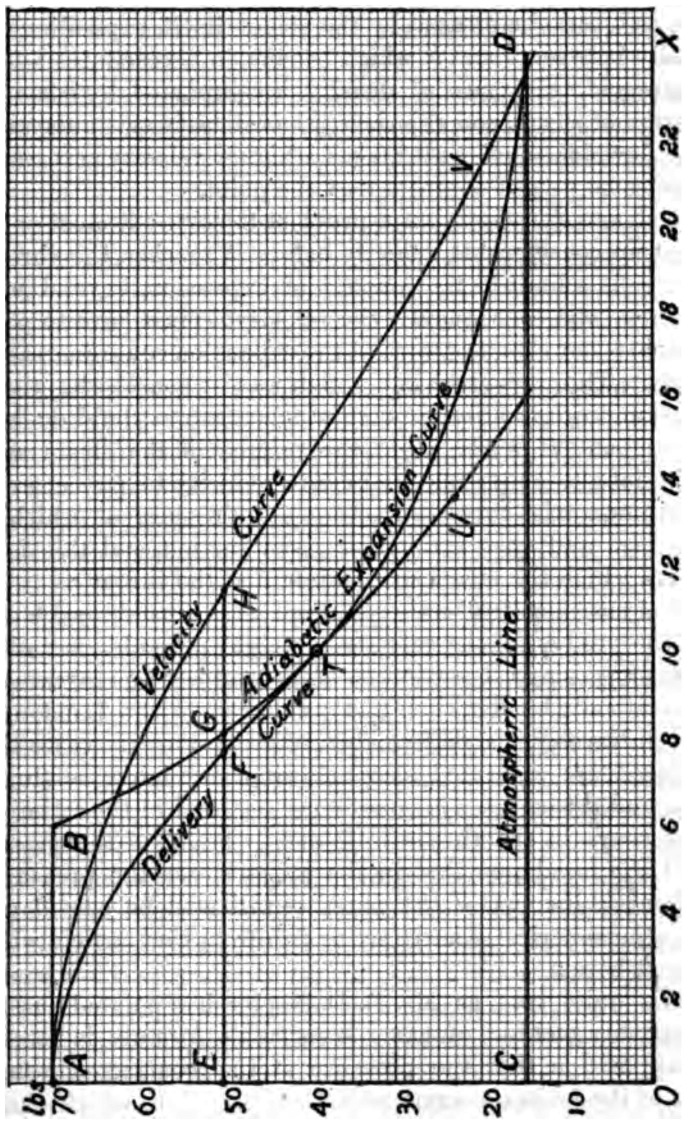


FIG. 263.

he form of the nozzle governs to a considerable extent the ower of the wheel, as will shortly appear.

If the steam passing a were to be forced through the throat c ithout any change of density, its velocity at c would be the ngth of a column of steam one inch square and containing six ubic feet nearly, or about 864 feet. But the velocity at any point, as b, is to be created by the steam itself, expanding adiabatically between a and b, which expansion necessitates a fall of pressure and a decrease of density, accompanied by some condensation of steam, and this decrease of density acts unfavorably on the flow, inasmuch as it makes a higher velocity necessary to get the same weight of steam past the point.

The steam, therefore, has a problem to solve; it must expand and get up considerable velocity before it reaches the throat c, but there is a limit to the amount of expansion, beyond which its volume, and consequently the velocity it must have to pass c, increases faster than the work of the expansion could increase its velocity. This is evident in a rough way by considering that an expansion down to absolute zero of temperature could do but a finite amount of work, while the volume of the steam would then be infinite, requiring an infinite velocity to pass c, so that a point must exist between 70 lbs. and 0 pressure, at which the most steam will pass through a given section, and this is the pressure which the steam will maintain at the throat c.

Stated mathematically, the pressure at c must be such as to make the product of the velocity × weight per cubic feet of the wet steam passing c a maximum. This subject was investigated mathematically by Rankine, and published in the London *Engineer*, in November and December 1869. Among the cases there considered are saturated steam flowing through a conducting nozzle, and through a non-conducting one, and in the latter case the pressure in the throat is found to be .58 of the pressure behind the nozzle, so that in the present case the pressure at c is about 40 lbs. At d the pressure will still be considerably above atmospheric. An extract or two from that article will be pertinent here:

"It is quite true, as Mr. R. D. Napier has pointed out, that the ordinary method of using Weisbach's formula is based on the supposition that the pressure at the contracted vein or throat of the jet is the same as in the space into which the gas is discharged"; also,

"Zeuner, near the end of the paper already referred to, considers it probable that the effective area of the jet is in general greater than the actual transverse area of its throat or narrowest part, in a proportion depending partly on the form of the outlet and partly on the pressure. He makes no attempt to determine from theoretical principles what laws that proportion may follow, and states that those laws are to be ascertained by experiment only."

The first extract shows that this is a subject comparatively new, quite new, I believe, to most engineers; the second, while showing the same thing, indicates that it is a new field for mathematicians.

I propose to deduce, by a graphical method, the law governing the reduction of the pressure in such a jet of steam, arriving thereby at the same result, namely: that with 70 lbs. at the entrance there will still be 40 lbs. in the throat of the jet; and I believe this graphical representation of the facts puts the whole matter in the clearest possible shape and must satisfy any one as to its correctness, although, at first, it may seem difficult to believe that the pressure in the throat and at the end of the nozzle can be so much higher than the atmospheric pressure.

Let *ABCD*, Fig. 262, be the theoretical indicator card for the adiabatic expansion of one pound of saturated steam from 70 lbs. pressure to atmospheric. *AB*, the volume of a pound at the highest pressure, can be taken directly from a table, but the remaining volumes must be calculated by the proper formulæ (see Rankine's *Steam Engine*, paragraphs 281 and 282, formulæ for u).

It will be simpler if we calculate first the amount of steam going through an area or opening of unity, *i. e.*, one square foot, and the line *AB* will then represent the velocity (6 feet per second) necessary to make 1 lb. of steam at 70 lbs. pressure, that is, to make 6 cubic feet of steam, go through a one-foot-square opening. In the same way any other line, as *EG*, represents, not only the volume of a pound of mixed water and steam at that pressure, but the velocity necessary for it to pass through an area of 1 square foot.

We must now inquire into the production of the necessary velocity by the expansion.

The work done in falling from 70 lbs. to any other pressure, as 50 lbs., is given by the area *ABEG;* thus, this area represents about 19,872 foot pounds, found by multiplying the fall of pressure per square foot by the average volume, or, $= 20 \times 144 \times 6.9 = 19{,}872$. Multiplying this by $2g$ and extracting the square root (see Rankine's *Steam Engine*, page 298, formula 1), we get 1,131 as the velocity produced by the expansion; *i. e.*, if a pound of steam were to fall by gravity through a height of

19,872 feet, 19,872 foot pounds of work would be done by gravity and a velocity of 1,131 feet produced as the result. This is the velocity corresponding to 50 lbs., and in the same way the velocity can be found for all pressures on the card. A more exact way is, by means of the formula (1 A), on page 387, Rankine's *Steam Engine*.

Let these velocities be laid off to a suitable scale opposite the corresponding pressures, and the velocity curve $A\ V$ drawn (the scale chosen is such that figures 2, 4, 6, etc., along $O\ X$, indicate 20, 40, 60, etc., feet per second).

A comparison of the two curves shows the following:

At 60 lbs. pressure the pound of steam has a velocity of 780 and a volume of about seven cubic feet; therefore it must have a sectional area of seven seven-hundred-and-eightieths of a square foot to pass through, because the area × velocity = volume. At 50 lbs. it has a velocity of $EH = 1{,}140$ feet and a volume $EG = 8.2$ cubic feet, and we get, in the same way, a sectional area of about .007 square foot; at 40 lbs. an area of less than .007 square foot; at 30 lbs. an area greater than .007 square foot; and for 20 lbs., etc, the area goes on increasing. Therefore the jet will naturally assume a shape with the smallest cross-section agreeing with the neighborhood of 40 lbs. pressure, and the shape will fit itself into the nozzle if the latter has a shape similar to that of the jet. By a more careful calculation the minimum sectional area will be found at about 40.6 lbs., and will be .0068 square foot, or .98 square inch.

Lay out a new curve of volumes $A\ U$, which we will call the "delivery curve," by multiplying all the velocities by the minimum sectional area, thus: EH multiplied by $.98 = EF$. This curve will be tangent to the adiabatic curve at T,* where the pressure is 40.6 lbs., and any volume given by it, as EF, is the volume of steam which will pass through a sectional area equal to the throat section, or .0068 square foot, at the corresponding pressure, or 50 lbs.; therefore, where the steam has this pressure the nozzle must have a sectional area larger than the throat area in the proportion of EG to EF. It does not come within the scope of this paper to show how to lay off a nozzle of the most correct form, but, given such a nozzle, we may mark the pressure at any point along it, as b, Fig. 261, by finding the

* The point T in the figure has, by mistake, been shown just below the 40 lb. line instead of above it.

pressure in Fig. 262, at which $EF : EG$ as the throat section : section at b.

The first use to be made of the above discussion is to obtain the size of the throat necessary to pass the quantity of steam desired, or, *vice versa*, when the throat is given to find the quantity. If one pound will flow through a throat section of .98 square inch, then a one-square-inch throat will pass 1.02 lbs.; but friction will certainly reduce the flow somewhat, so that we may as well allow 2%, and take one pound as the delivery through one-square-inch throat section, which agrees with Rankine's approximate delivery formula (see Rankine's *Steam Engine*, 11th edition, page 298, 3d line from the bottom).

The next use is a much more important one in its bearing on the power and efficiency of the reaction-wheel If the nozzle be terminated at the throat the steam will be abandoned at 40.6 lbs. pressure; but if it be flared beyond the throat to d, so that the section at d is about 10% greater than the throat section, then we shall hold on to the steam until it expands down to about 26 lbs., for which pressure EG is 10% greater than EF, and thereby increases the power and efficiency of the wheel.

To calculate the horse-power for these two forms of nozzle we proceed as follows:

In one second one pound of steam escapes, or $1 \div g = .031$ unit of mass, which, multiplied by the velocities for 40.6 and for 26 lbs., viz., 1,440 and 1,905, give 45 and 59 for the momentum per second, which are the reactions of the jets in pounds. In one minute the orifice d of the jet travels $4,000 \times \pi$ feet; the steam does not however issue tangentially, but at an angle α (Fig. 263), and therefore the work done is $45 \times \cos \alpha \times 4,000 \times \pi$ in the one case, and $59 \times \cos \alpha \times 4,000 \times \pi$ in the other, and if $\cos \alpha$ equals, say .9, we get respectively 509000 and 667000 foot pounds per minute, or about 15 and 20 H.P. In estimating the cosine of α an allowance should be made for the divergence of the outer parts of the jet from the direction of its axis.

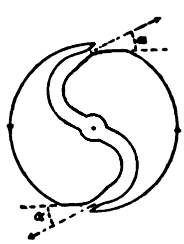


FIG. 263.

The same wheel run at 16,000 revolutions would develop four times the gross horse-power with a boiler pressure of about 59 lbs., as before shown, giving therefore about 62 and 80 H.P., according to the degree of expansion. By prolonging the nozzle until its outward end has about one and a half times the throat section, the steam can be expanded to atmospheric pressure with a velocity of 2,300 feet, which would increase the horse-power theoretically to about 24 for 4,000 revolutions, and to 96 for 1,600. That the full theoretical advantage of this complete expansion could be realized in an actual wheel cannot be assumed, as there may be drawbacks which would appear in an experimental test.

In the above calculation we have made no allowance for friction beyond using one pound per second, instead of the somewhat larger discharge indicated by our analysis. But, as in reducing the flow, the friction reduces the velocity, a further allowance should be made for it. This, however, will be different for different sizes, forms and smoothness of nozzles, and we cannot attempt to calculate its effect here.

The horse-powers calculated above may be called gross horse-powers, or those actually developed by the steam jets. From these must be made a deduction peculiar to a reaction-wheel, but which is avoided in a turbine by means of the entrance guide-blades. All of the steam arriving at the point a has been received by the wheel at its centre with but little velocity, and during its flow outward to a the wheel has given to it the velocity of its periphery, or 52½ feet per second for each 1,000 revolutions per minute. Thus the steam absorbs each second from the wheel an amount of energy equal to the mass per second into half the square of the velocity $= .03 \times .5\,(52.5)^2 = 43$ foot pounds, or about 2,600 foot pounds per minute. This deduction for the speeds of 4,000 and 16,000 per minute will be respectively 16 and 256 times greater, or 41,000 and 656,000; thus reducing the gross horse-powers per 4,000 revolutions, namely, 16, 20, and 24, to less than 15, 19, and 23; while for 16,000 the horse-powers fall from 64, 80, and 96, to 44, 60, and 76.

The water used, 3,600 lbs. per hour, divided by the horse-power, gives from 240 down to 60, and even 48 for the complete expansion.

A comparison of the above results, with those deduced in the rough calculation, sustains the general accuracy of the latter.

DISCUSSION.

Mr. Geo. H. Barrus.—It may be of interest, in connection with this paper, to give the results of tests which I made upon an engine which depended for its action on the impact of a steam jet striking against the blades of a wheel enclosed in a metal casing. The chamber in which the wheel revolved was about 15 inches in diameter and 1½ inches thick, these being the inside dimensions. The wheel was 14 inches in diameter at the edge of the blades, and the width of the blades was about 1 inch. The steam was admitted through a pipe placed in such a position that the steam struck the blades at a point near the end, and in a direction tangential to the circle. The orifice was ¼ inch in diameter. The quantity of steam used, which was constant, amounted at 100 lbs. pressure to 210 lbs. weight per hour. The amount of brake horse-power developed when the engine was run at its maximum speed, viz., 6,000 revolutions per minute, was 1.2 H.P. Under these circumstances the rate of consumption of steam per brake horse-power per hour was 175 lbs.

Mr. Strickland L. Kneass.—Although I cannot give at the present time any information in regard to experiments with the steam turbine, yet I would like to present some notes upon that part of Prof. Webb's paper bearing upon the discharge of a jet of steam.

The efficiency of a steam reaction-wheel depends upon two principal conditions:

(*a*) The amount of work converted into velocity by expansion of the steam.

(*b*) The application of the energy of the discharging jet to the rotation of the wheel.

The first condition requires, in order to produce the highest theoretical efficiency, that the expansion must be carried down to the pressure of the surrounding medium during its passage through the nozzle, and that with the least possible loss. The second condition has already been covered, and therefore will not be discussed.

The peculiar form assumed by a jet of steam after issuing from an orifice indicates that the laws that govern the discharge may be very complex, and that the shape of the nozzle to develop the maximum energy of the jet is entirely different from the form used for inelastic fluids. The sketches shown in the illustration are based upon observations and photographs of jets at 120 lbs. initial gauge pressure; four orifices of different shapes are represented, each having an individuality of its own.

1. A short cylinder or convergent nozzle. (Fig. 320.)
2. An aperture in a thin plate. (Fig. 321.)
3. A divergent straight taper. (Fig. 322.)
4. A divergent curve. (Fig. 323.)

For the sake of clearness the white part of the discharge is made with dark tones and the visible lines near the orifice are somewhat emphasized.

The first two nozzles permit an immediate diametral expansion and consequent loss of velocity-producing energy; an

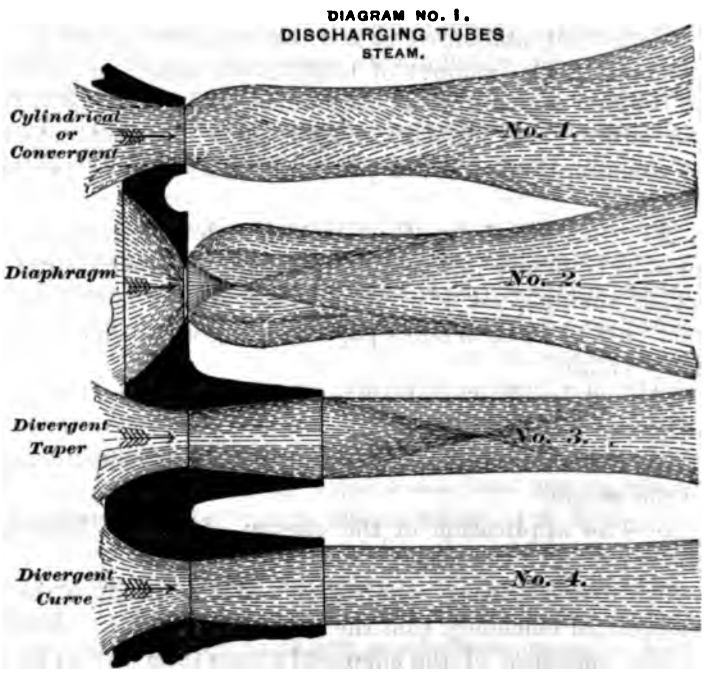


FIG. 320.
FIG. 321.
FIG. 322.
FIG. 323.

envelope of low-pressure steam is formed, through which the central jet is clearly seen; this swelling of the jet occurs whenever the internal pressure at the mouth of the nozzle is higher than that of the medium into which it discharges. The appearance of the envelope depends upon the pressure and the percentage of moisture in the steam; when very wet the discharge is almost perfectly white, while dry steam gives a clear, transparent blue with occasional flecks of white and changes of color, that indicate the boundaries of the internal and external jets.

The divergent nozzles No. 3 and No. 4, on the other hand, compel an axial expansion which utilizes the energy more fully, l if correctly proportioned will give the steam very nearly its oretical velocity.

Some experiments were made about four years ago upon the rnal condition and velocity of discharge of a jet during passage through the tubes of various shapes, and the results oodied in a paper presented before the Engineers' Club of

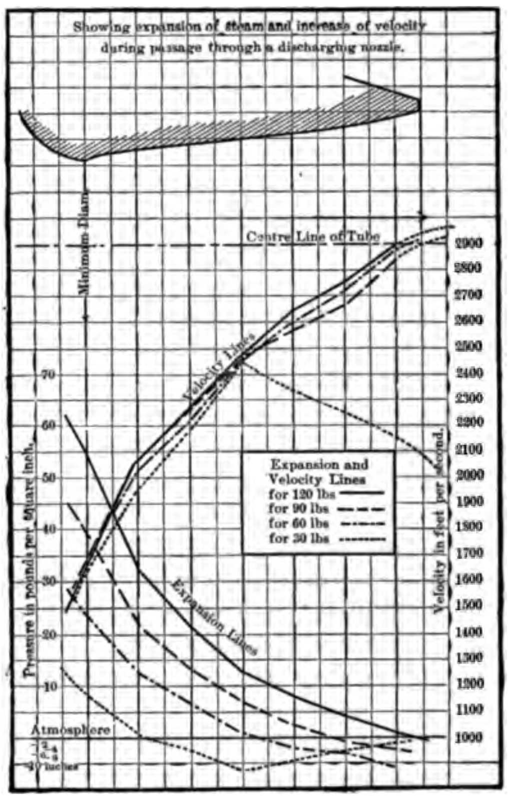


FIG. 324.

ladelphia. Therefore, without going into detail, I will state t the internal pressure during discharge was observed at eral different points of a tube, and the velocity of the steam ulated for each section. The results correspond closely those found theoretically by Prof. Webb, and, further to strate, I would present a diagram (Fig. 324) showing the ansion of the steam and increase of velocity in a nozzle h divergent curves; pressures and velocities at all points of the tube are shown for 30, 60, 90 and 120 lbs. initial gauge pressures, and the relative length of the tube for any degree of expansion is apparent. The upper lines indicate curves of velocity, and the lower expansion lines indicate the fall of pressure as the steam traverses the tube.

There will be noticed a curious fact to which I have never seen reference made, viz.: that the velocity is practically the same at any given section for all pressures from 30 to 120 lbs., making the weight discharged per second almost exactly proportional to the initial density of the steam.

In conclusion I would like to state that the ratio of the absolute pressure at the minimum diameter of the tube, to the absolute initial pressure, was not found to be constant in my experiment, but varied slightly with the pressure; the value of 58%, as given by Rankine and demonstrated graphically by Prof. Webb, would seem therefore to be a mean for all pressures.

Mr. Carleton W. Nason.—I notice there that the best efficiency is about 60 lbs. of water per horse-power per hour, and I believe there is also a considerable percentage of that which has been due to the condensation of steam within the wheel. Take, for example, the instance of the Avery engine, which is a pair of steel plates brazed together at the edges, and running at a very high velocity, the wheels being six feet in diameter. The condensation in that wheel, unless the surface was protected, must have been very large. Experiments that I have made in steam condensation on metallic surfaces show that as the velocity of air or water passing over them becomes high, they increase very rapidly in their transmission of heat and condensation of steam on them.

Prof. Sweet.—I would like the attention of the meeting for a few moments for the purpose of giving a few facts in respect to the Avery engine, and to ask whether the theories advanced by the theorists correspond with the facts as they are known to have existed in the engine. One of the original engines can be seen at the Cooper Institute in New York. It is a machine five feet across the arms, made of steel, with two orifices, each one-twelfth of an inch in thickness and one-sixth of an inch long. It was reported, and I believe it to be true, that those orifices would permit all of the steam that an ordinary boiler would make to pass through them when the engine was at its maximum speed.

The orifices would not permit all of the steam from the boiler to pass out when at rest, but when under motion they would pass over the steam, and it was for this reason, probably, that all the steam that the boiler could make would go through those two small holes. Probably a hundred engines of this kind were run for a number of years, driving saw-mills, and were at that time a successful machine, at least so the people thought who ran them; and one machinist, who took out an Avery engine and put in a stationary slide valve engine, said it took just as much wood to run the mill as it did before. The trouble with the Avery engine was, they ran so fast that the second arm came around into the steam as it left the first arm, cutting each other away, so that the arms would not last more than two or three months. While that would not trouble us at all at this day, it was what threw the Avery engines out at that time. The first slide valve engines that were put in to take the place of the Avery engines were not particularly more economical. In Mr. Avery's note book, which I own, he states that in his engine, which was on a locomotive, and started to run from Newark to New York and ran into the ditch, the arms were seven feet from end to end; that is, revolving in a circle seven feet in diameter, and the outer end of the arm ran at the enormous velocity of fourteen miles a minute.

Mr. Suplee.—I should like to refer to these instantaneous photographs of jets. I notice some similarity between them and some photographs of rifle bullets I saw not long ago, where the object was distinct, as well as the condensation produced in the air made by the velocity of the bullets through the air. The shapes of the curves were very similar to those of the jet, except that when the velocity reached a sufficiently high point the air was very noticeably condensed in front of the bullets. Of course the reaction in the gun would not be affected in the slightest by the subsequent resistance which the bullet might meet in the air, but the shape of the curves in the photographs was very similar to the sketches exhibited.

*Prof. Webb.**—These experiments with a steam jet seem to me to be most important. A thing which is used as much as steam is and in the economical use of which we are so much interested, should be used intelligently, and it would seem that large amounts of money might be advantageously spent in learning

* Author's Closure.

exactly how steam acts under various circumstances. There is one fact, which was not brought out sufficiently, to which I wish to call your attention, as to the escape of steam from the reservoir, with reference to the amount of steam escaping at different pressures and with different lengths of nozzle. Suppose we have a reservoir and a properly shaped nozzle which permits the expansion of the steam down to atmospheric pressure. Say we have 70 lbs. in the reservoir and 14 lbs. outside at the end of the nozzle. A certain amount of steam will escape. If you cut off this nozzle, so as to shorten it gradually, still, leaving out the effect of friction, the same amount of steam will escape; the surrounding air takes the place of the part cut off the nozzle as long as it is not cut away as far as the throat. Again, if you increase the outside pressure to anything less than the throat pressure, or 40 lbs., you will not affect the flow of steam.

The experiments undertaken by Mr. Kneass are of great interest and of considerable difficulty. Too great dependence should not be placed upon conclusions drawn from the appearance of the jets. Photographic views, even of jets, may require considerable ingenuity to interpret them properly, and may in some points be deceptive, and cuts engraved from them must be more so. I would suggest that the jets should be placed in front of a background ruled in small squares, and this background photographed as it appears through the jets, so as to get at their constitution by means of their distorting effect upon the system of squares. Precaution should also be taken to prevent rotation of the jets, a very natural explanation of the form of such jets as Figs. 320 and, especially, 321 being that the steam escapes in paths approximating to the elements of a hyperboloid of revolution. The "curious fact" referred to is a well-known one, and lies at the bottom of Rankine's rule for getting the delivery in pounds per second of a jet by dividing the pressure by 70 and multiplying by the section of the jet in square inches.

Some gentleman stated that in a steam reaction-wheel the periphery ought to run at the same speed as the steam escaped, and Prof. Sweet questions whether theory and practice ought to agree when the steam escapes from a revolving wheel.

Theory takes into account the motion of the wheel, and, indeed, the effect of the motion is very simple. Steam will escape just as fast whether the wheel revolves or stands still, provided the pressure back of the orifice remains the same, but the effect of the motion of the wheel is to increase that pressure by the centrifugal force of the steam in the wheel; this makes the steam denser, with more pressure to force it through the orifice, so that the same orifice delivers more when the wheel is in motion. There is nothing in the suggestion that the wheel moves over or past the steam, thus increasing the flow, because the steam before it escapes is going round with the speed of the wheel; centrifugal force fully accounts for any increased flow. As to running the wheel as fast as the steam escapes, it is impossible to do so with a reaction-wheel, because increasing the speed of the wheel increases the centrifugal force, and therefore the speed of the escaping steam, and makes it impossible for the former to catch up with the latter; and this is the fundamental reason why a reaction-wheel cannot be highly efficient, as a turbine can. In a turbine you can make the wheel revolve as fast as the steam escapes. In the calculations I have made in the paper, the greatest difference of pressure due to centrifugal force is 11 lbs. In a steam turbine a fall of pressure of 4 or 5% produces as high a velocity in the steam as the wheel can safely have to keep up with it. Multiple turbines are built which run, say, 9,000 revolutions per minute for a 30 H. P. machine. In these the steam is economized by repeated expansions; there are forty-five expansions in the turbine referred to. In calculating a steam turbine, all these points are taken into account in theory except the radiation; and these machines are so much smaller than other engines, that the radiation might be very rapid, and yet but little heat be lost.

It has been suggested that the gyroscopic action of the Avery wheel prevented the locomotive from freely following the curves of the track; of course, to avoid such an action, all that would be needed would be to place the axis of the wheel vertical, in which position it would have a beneficial effect in opposing pitching and rolling of the locomotive.

Mr. Barrus's experiment is very interesting, as it falls in so well with calculation. Of course 6,000 revolutions was nothing like the speed the wheel ought to have had to do its best.

CCCCLV.*

JET PROPULSION.

BY J. BURKITT WEBB, HOBOKEN, N. J.
(Member of the Society.)

THE following contribution to the subject will, I hope, be of interest. It needs but little introduction, since, to a great extent, it explains itself. A year or two since, when the subject was a live one in certain quarters, several students took special interest in the mechanical principles involved, when we came to them in the course of instruction. I laid down the law, and, as usual, enforced it by illustration and argument, but it so happens that these principles, like many other good ones, are unacceptable to the unregenerate mind. When, therefore, I learned that some had gone away and quietly tried the experiment, I enjoyed one of those moments of satisfaction which come to every successful laborer in a good cause; I was glad that Nature had had the chance to vindicate her own laws. The subject is not one which is, ordinarily, well understood, in fact many engineers are totally wrong in their conception of the way in which a jet acts, and it will be a satisfaction to me if this letter contributes towards a more correct knowledge of the facts.

April 12, 1891.

DEAR SIR:

The following is a description of some experiments which I made with water jets in the Fall of '89, to prove that the reaction of a jet is the same in air as in water.

These trials were made to disprove a statement that a stream of water issuing from a nozzle under water reacts upon the surrounding water and tends to drive the nozzle in the opposite direction to that of the jet, in the same way that a solid rod, issuing from the same nozzle, would react upon a stone wall.

I felt certain as to the results of the experiment because *you* had told me what they would be beforehand.

The arrangement for measuring the reaction is shown in the accompanying sketch (Fig. 260).

The tests were made in a large tank and the water was supplied by a steam-

* To be presented at the Providence Meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. of the *Transactions*.

pump at a constant pressure of 100 lbs. per square inch. The pump had a steam-cylinder of four inches and a water-cylinder of two and three-sixteenths inches diameter, and a stroke of two and a half inches.

The nozzle was fastened to the foot of a pendulum four feet long and connected to the pump by a flexible half-inch rubber hose. The hose was hung by strings so that the stiffness was inappreciable. The pendulum at the nozzle was connected to a spring balance by a cord about four feet long. When the jet was flowing the pendulum was brought to the vertical by moving the balance, and in this way every force due to the deflection of the pendulum was eliminated. The pendulum was first hung so that the nozzle was a little above the water and the reaction found to be four pounds. The pendulum was then lowered so that the nozzle was about four inches below the surface of the water, all other conditions being the same, and the reaction was found to be the same—four pounds.

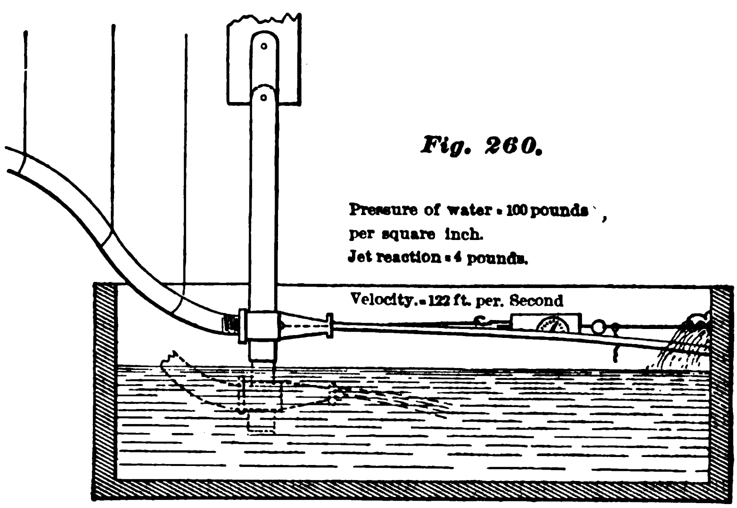


Fig. 260.

The only difference was that when the nozzle was under water the balance hand vibrated a little more.

When these results were reported to the party making the statement, the rejoinder was made that "the reason was that the jet struck the opposite side of the tank"—five feet distant—"and so pushed back on the nozzle."

I therefore measured the reaction in air again with the pendulum hung from the roof of the house, twenty-five feet above the ground, so that the jet broke into spray before it struck. All the conditions were the same as before and the reaction was the same—four pounds. I then put a flat plate on the pendulum and let the jet impinge upon it, and the pull was the same as the reaction—four pounds. I afterwards used a similar apparatus (but more delicate) with a constant head of water—about 10 lbs. per square inch—from an elevated tank. The only difference from the first trial was that the pendulum was held in a vertical position by a delicate spring balance when the jet discharged in the air, and then nothing was altered, but the water allowed to rise in the tank until the nozzle was two inches below the surface, and the balance reading remained *absolutely the same.*

I hope these results may be of some service to you, and thanking you for your kindness and instruction while I was at Stevens Institute, I remain,

Truly yours,

CHAS. J. EVERETT, JR.

DISCUSSION.

Mr. Carleton W. Nason.—I would like to ask Prof. Webb, in regard to the duty of the engine or pump, whether there is anything more in it than merely the pressure at the point of discharge multiplied by the area of the opening in inches, as giving the "pull" in any one direction?

Mr. G. C. Henning.—The experiments described in this paper are so clear and to the point, having eliminated all confusing or conflicting conditions, that even the ordinary inventor can understand them. The one fact demonstrated, "that the effective principle" is the reaction upon the orifice, irrespective of the surrounding medium, is clearly brought out and deserves particular attention.

Recently I had occasion to look into the matter of jet propulsion because an inventor was sent to me to explain his new invention, for which he wanted $40,000 for experimental purposes to prove that he was correct. His plan was to eject the water from the boat with a velocity equal to that of the boat, claiming thus to obtain the maximum efficiency. At the same time this inventor condemned Dr. Jackson's scheme and statement that a jet impinging upon a surrounding mass of water would act precisely like a steel rod pushing against a stone wall. Although he believed that the boat was propelled merely by the reaction of the issuing jet of water on the orifice he could or would not keep this in view, but was misled by other conditions. Had he been able to make as simple an experiment as detailed in the paper before us, he would not have spent time and money in the attempt to accomplish an impossibility.

This paper deserves to be generally read for its clearness and simplicity and directness.

Mr. Henry H. Suplee.—It was my fortune to witness some experiments made to increase the efficiency of a screw propeller by surrounding it with a converging nozzle. The experiments were made on a little naphtha launch about 12 to 15 feet long, and a propeller about 15 inches in diameter. The nozzle was made with a bell-mouth entrance, so that the water would readily flow into it. It fitted entirely around the propeller, so that the water was drawn in by the propeller and discharged through the tube. When the boat was moored fast to the wharf and a spring-balance inserted, similar to the experiment described by Prof. Webb, it showed a marked increase in the pull on the spring-balance when the tube was used. It seemed as if the propelling power of the propeller was decidedly increased. In some cases it ran up as high as 25% to 30%. When the boat was loosed and we endeavored to see the increase in the speed by running over a measured course there was a marked loss, as great as 10% or 15%, which was apparently due to the increased resistance of the braces, etc., by which the tube was attached, in the water. So that while it gave an increased pull it gave also a marked reduction in velocity.

Mr. F. Meriam Wheeler.—Some reference was made to a boat constructed in South Brooklyn using the principle of the jet for propulsion, and in which the inventor was more than sanguine in regard to the success of it. I always contended that, even if he secured the effect which he expected with his jet, he never could practically apply it to vessels, for the simple reason that he has got to use a *pumping* engine. It is a well-known fact that the piston speed of a pumping engine is very limited. In ordinary direct-acting pumps about one hundred feet piston travel per minute is generally considered the standard rate of speed. In the fly-wheel type of pumping engines, it is sometimes as high as 350 to 400 feet. But what is that speed compared to the modern marine engine that makes from 850 to 1000 feet piston travelled per minute? The application of the jet system would require the engine to be very bulky and heavy, to say nothing about its lower rate of economy in steam. In other words, the room and weight required for a pumping engine for jet propulsion (providing the jet was a success) would be so great that you could not practically apply it with any success. You would require more boiler capacity, your vessel would be full of pumping machinery and boilers, with little or no room left for coal, crew, or provisions.

Mr. W. W. Bird.—I should like to ask one question: If a jet is thrown into a vacuum what is the reaction then?

Prof. Webb.—The object of the paper is, as Mr. Henning has said, to put the matter in a shape that can be understood.

If a jet is thrown into a vacuum its reaction is the same, viz.: its momentum—equal to its mass per second multiplied by the velocity of exit with respect to the nozzle. The reduction of the pressure against which the jet issues, will, however, other things being equal, increase the velocity of the jet and thus increase its momentum. With the same energy expended in pumping, the velocity of the jet will be the same, however, whether it issues just beneath the surface or on a line with the keel, and it might be delivered equally well *at* the surface into the air. If I understood Mr. Henning's remarks aright in reference to this inventor's plan, it was that he was to have the jet escape through the nozzle at the speed of the vessel.

Mr. Henning.—Yes, sir; the water moved back at the same speed.

Prof. Webb.—That is a most excellent plan. That will give perfect efficiency, and the machinery may be the simplest in the world. All you have to do is to put a straight tube, with no machinery whatever, right through the vessel, and then the water will naturally enter and leave at the same speed, and there will be no loss except friction—you can get a canal horse or something else to pull the vessel along. (Laughter.)

Mr. Henning.—The only objection would be that the orifice would have to be exactly the same as the midship section, because the resistance of the boat is equal to so much per square foot of reverse midship section.

Prof. Webb.—I did not suppose this jet was to propel the vessel at all. (Laughter.) That is the way in which all these schemes have been carried out; they are propelled by high-speed or high-pressure talking, mostly, and these rhetorical jets stir up the surrounding sea of dollars and cents, so that it doesn't come to rest again without much money disappearing in heat or otherwise.

In reply to Mr. Nason: If the area of the nozzle is A square feet and the water is forced through it with a velocity V feet, then the volume of water delivered per second is VA. Also, if w is the weight of a cubic foot of water and g the intensity of gravity, then VAw is the weight discharged per second, and $VAw \div g$ the mass. Also let F be the force in pounds which propels the vessel with the velocity v through the water.

The energy developed by the engine or pump, neglecting friction, is reckoned thus:

The useful work per second is that employed in propelling the vessel, and is equal to Fv. The wasted work is the energy left in the water, which gradually changes into heat as the water comes to rest again. It is equal to half the mass multiplied by the square of the velocity with which the escaping water goes through the surrounding water, equal to $\frac{1}{2} VAw \div g\,(V-v)^2$. The sum of these two is the whole energy developed. Now $F =$ the momentum of the jet with respect to the vessel less the momentum of the feed-water with respect to the vessel, or $F = VAw \div g \times V - VAw \div g \times v = VAw \div g \times (V-v)$, which being multiplied by v gives $VAw \div g \times v\,(V-v)$ for the useful work.

Adding the useful and wasted work, we get $\frac{1}{2}\ VAw \div g \times (V^2-v^2)$. This is equal, as it should be, to the difference of the energies (reckoned with reference to the vessel) of the water leaving and entering the vessel.

The pounds pressure pushing the vessel ahead is equal to the difference of the reactions (or momenta) of the out-going and in-coming jets, regarding the feed-water as a jet entering the vessel in front. There is an interesting fact with reference to the reaction of a jet. Suppose at first a tank is closed up and you have a pressure of 100 per square inch near the bottom inside. Now let a hole be opened of 1 square inch section so that a jet of water flows out, this will make a difference of 200 pounds on that wall of the tank in which the hole is made; 100 pounds because the wall is cut away, and another 100 pounds because the pressure on the rest of the wall near the hole decreases.

In reply to Mr. Suplee: If the propeller in an ordinary ship is surrounded by a tube—we won't say a conical tube, but a parallel one—it will prevent the propeller giving the water any radial velocity. Therefore, if we could put a tube around the propeller, and have no friction in that tube, we would gain something; but the friction in the tube would probably be the element which would bring the speed of the vessel down. In reference to the remarks as to the impossibility of running a vessel by a jet on account of using a pumping engine, why, every propeller boat is run by a jet; it has simply a different kind of a pump. A propeller is a pump, then, to give the velocity backwards, and the only difference between ordinary propellers and these schemes for running a small jet is, that the propeller is large enough to propel the boat economically and the jet is not, and the propeller runs at a small velocity and therefore gets a higher efficiency. But further, as to the discharge into a vacuum. It does not matter what the jet discharges into, any more than when you shoot a gun it matters what you shoot at; if the gun kicks, it will kick, and reaction is only another name for kicking. (Laughter.) At the same time, if you plug the gun up it is not likely to let the ball go out. Consequently, reduce the pressure outside to a vacuum, and the more you reduce it the greater the velocity of escape into the vacuum. The whole propelling action is obtained in giving the velocity to the water; after the jet leaves the vessel it makes no difference what you do with it; you can put a board right up close to the back of the vessel, provided you don't put it near enough to choke the jet, which would be analogous to plugging up the gun. The principle of jet propulsion occurred to me when the *Charleston* was sent after the *Itata*. If the two vessels were able to steam at about the same speed and the former had nearly caught the latter, and then they commenced firing at each other, the reaction of the guns would be a case of "jet propulsion;" that is, the reaction or kick of the *Itata's* guns, firing backward, would increase her speed, while the forward firing on the *Charleston* would retard her and prevent her from overtaking the *Itata*.

CCCCLVI.*

FLEXURE OF ELASTIC RINGS.

BY DE VOLSON WOOD, HOBOKEN, N. J.

(Member of the Society.)

WE first establish the general equation of flexure for a piece initially bent. Let IO be the neutral line of the piece $AJFE$, whose length IO is infinitesimal; ρ_1 the initial radius of curvature at O, ρ the radius of curvature after it is bent by an external force. Let the moment of the bending stress at O M, AJ the transverse section of the beam normal to the axis IO, BD parallel to AJ, EF a section originally normal to the axis IO, and GH a normal after final bending.

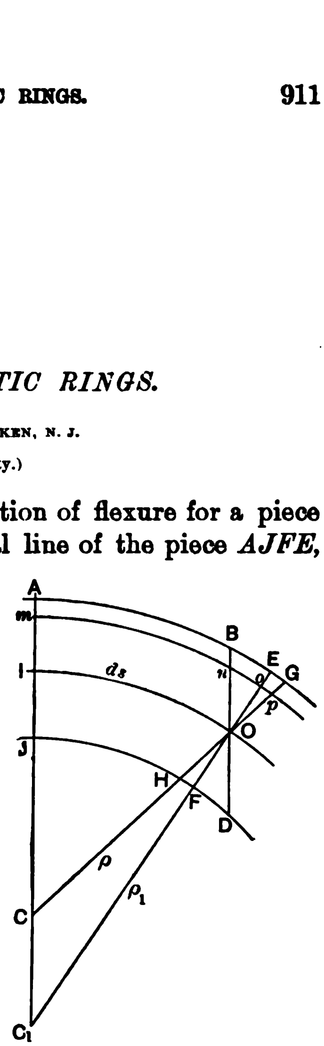


FIG. 264.

Let mo be any element of the initially bent beam at a distance $y = Oo$ from the neutral axis, then will $op = \lambda$ be the elongation due to the moment M, and no the excess of the initial length mo over that of IO.

We have
$$\frac{ds}{\rho_1} = \frac{no}{y},$$

$$\frac{ds}{\rho} = \frac{np}{y};$$

$$\therefore np - no = \lambda = \left(\frac{1}{\rho} - \frac{1}{\rho_1}\right) yds \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Vol. XII. of the *Transactions*.

The stress that will produce an elongation λ of an element whose section is k, will be

$$p = \frac{Ek\lambda}{ds} = Eky\left(\frac{1}{\rho} - \frac{1}{\rho_1}\right), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where E is the modulus of elasticity. The moment will be

$$py = Eky^2\left(\frac{1}{\rho} - \frac{1}{\rho_1}\right); \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

hence the formula for the flexure of a piece initially bent is the same as for one initially straight, excepting that $\frac{1}{\rho} - \frac{1}{\rho_1}$ is substituted for $\frac{1}{\rho}$ in the latter. If I be the principal moment of inertia of the cross section at O, we have, by taking the sum of all the internal moments of stress,

$$M = \Sigma py = EI\left(\frac{1}{\rho} - \frac{1}{\rho_1}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Deflections are not easily computed by this formula for the cases we chiefly desire to consider in this paper. The change of direction of the neutral line will equal the angle $COC_1 = EOG$, which call $d\varphi$. Then from Fig. 264 and equation (1)

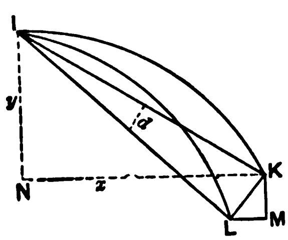


FIG. 265.

$$d\varphi = \frac{op}{Oo} = \frac{\lambda}{y} = \left(\frac{1}{\rho} - \frac{1}{\rho}\right)ds = \frac{M}{EI}ds \quad (5)$$

In Fig. 265, let the curve IK be the axis before flexure, and IL after, K and O representing the same points in the preceding figures, $IK = ds$, the displacement of a point K in reference to I will be KL, the two components of which will be KM vertically and LM horizontally. Through I draw the straight lines IK and IL, then will the change of direction of the axis at I be

$$LIK = d\varphi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Let the coördinates of K in reference to I be

$$KN = x,\ NI = y,\ IK = s,\ NIK = \theta = LKM;$$

then $\quad KL = zd\varphi,\ KM = z \sin \theta d\varphi,\ LM = z \cos \theta d\varphi, \quad . \quad . \quad (7)$

$$x = z \sin \theta,\ y = z \cos \theta;$$

$$\therefore KM = xd\varphi,\ LM = yd\varphi. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Let δ_x be the x-component of the displacement, and similarly δ_y the y-component, due to the moment M; then, if the change be infinitesimal, equations (5) and (8) give

$$LM = d\delta_x = \frac{My}{EI} ds,\ KM = d\delta_y = \frac{M_x}{EI} ds \quad . \quad . \quad . \quad (9)$$

The displacement of K due to several bending moments between K and I, if expressed by a continuous function, will be, equations (5), (7), and (9),

$$\varphi = \int \frac{Mds}{EI} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

$$KL = \int \frac{Mzds}{EI} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

$$\delta_x = \int \frac{Myds}{EI} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

$$\delta_y = \int \frac{Mxds}{EI} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

These equations are also applicable to beams initially straight.

APPLICATIONS.

1. Let the beam be prismatic, initially straight and horizontal, fixed at one end and loaded at the free end.

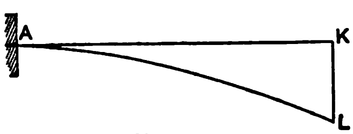


FIG. 266.

Let P represent the load, l the length of the beam; then, the origin being at K,

$$ds = dx,\ M = Px,\ y = o,\ z = x,$$

and the preceding equations give

$$\varphi = \frac{P}{EI}\int_o^l xdx = \tfrac{1}{2}\frac{Pl^2}{EI};\ KL = \delta_y = \tfrac{1}{3}\frac{Pl^3}{EI};\ \delta_x = o \quad . \quad . \quad . \quad (14)$$

These are well-known results.

If the origin of coördinates be at A, x in the preceding equations must be changed to $l - x$.

2. Let the beam be initially curved, being that produced by the weight P in the preceding exercise; then add another weight P.

To find y, let the origin be at L; we have, by means of rectangular coördinates, the well-known equation,

$$EI\frac{d^2y}{dx^2} = Px = M,$$

which integrated twice will give

$$y = \frac{P}{6EI}(x^3 - 3l^2x),$$

which substituted in equation (12) above, and still considering $ds = dx$, gives

$$\delta_x = \frac{P^2}{6E^2I^2}\left(\frac{x^5}{5} - l^2x^3\right),$$

$$= -\frac{2}{15}\frac{P^2l^5}{E^2I^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

when $x = l$.

The *increased* deflection due to the second load P will be

$$\delta_y = \frac{1}{3}\frac{Pl^3}{EI}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

the same as for the first load P, the new value of KL being $2\delta_y$.

3. If a thin, uniform, circular, elastic ring be hung on a pin, and carry a weight P at its lower end, required the form after distortion.

The ring after distortion will be symmetrical in reference to its vertical diameter, and its highest and lowest elements will be horizontal; and the problem will be the same as that of half the ring with its upper end firmly fixed, and at its lower end a moment equal to that existing internally in the solid ring, and also a horizontal force acting in the same sense as in the solid ring, retaining the lower end in the vertical diameter, and carrying $\frac{1}{2}P$. There will be no shearing stress at A or B. Let Fig. 267 represent the case, and

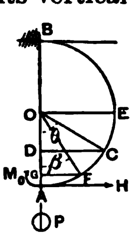


FIG. 267.

M_0 be the moment of internal stress at A.
H, the horizontal force at A.

w, the weight of unity of length of the ring.
r, the radius of the centre line of the ring $= OB$.
θ, the angle AOC.
β, the angle AOF.
ρ, the radius of curvature of the ring after distortion.
φ, the change of direction of the tangent at any point due to distortion.
s, the arc measured from A.
I, the moment of inertia of a transverse section.
E, the modulus of elasticity.
M, the moment of internal stress at any point as C.

Then

$$rd\theta = ds = rd\beta \quad \ldots \ldots \quad (17)$$

$$y = DG = r(\cos\beta - \cos\theta) \quad \ldots \ldots \quad (18)$$

$$x = DC - GF = r(\sin\theta - \sin\beta) \quad \ldots \ldots \quad (19)$$

The moment of the weight of the ring between A and C in reference to C will be

$$\int_o^\theta r\,(\sin\theta - \sin\beta)\,wrd\beta = wr^2(\theta\sin\theta + \cos\theta - 1) \,. \quad (20)$$

The moment of all the forces between A and C will be, considering those moments as positive which increase the radius of curvature,

$$M = -M_0 + rH(1-\cos\theta) + \tfrac{1}{2}P.\,r\sin\theta + wr^2(\theta\sin\theta + \cos\theta - 1) \quad \ldots \ldots \quad (21)$$

Equations (10) and (21) give, for the change of direction of the curve from F to C:

$$EI\varphi = r\big[(-M_0 + rH)(\theta - \beta) - rH(\sin\theta - \sin\beta) - \tfrac{1}{2}Pr(\cos\theta - \cos\beta) + wr^2(-\theta\cos\theta + \beta\cos\beta + 2\sin\theta - 2\sin\beta - (\theta - \beta)\big] \quad \ldots \ldots \quad (22)$$

Reckoning from the highest point, where the curve necessarily remains horizontal, θ becomes π, and the preceding equation becomes

$$EI\varphi_\beta = r\big[(-M_0 + rH)(\pi - \beta) + rH\sin\beta + \tfrac{1}{2}Pr(1 + \cos\beta) + wr^2(\beta\cos\beta - \sin\beta + \beta)\big] \quad \ldots \ldots \quad (23)$$

If the origin of coördinates be at A, x in the preceding equations must be changed to $l - x$.

2. Let the beam be initially curved, being that produced by the weight P in the preceding exercise; then add another weight P.

To find y, let the origin be at L; we have, by means of rectangular coördinates, the well-known equation,

$$EI\frac{d^2y}{dx^2} = Px = M,$$

which integrated twice will give

$$y = \frac{P}{6EI}(x^3 - 3l^2x),$$

which substituted in equation (12) above, and still considering $ds = dx$, gives

$$\delta_x = \frac{P^2}{6E^2I^2}\left(\frac{x^5}{5} - l^2x^3\right),$$

$$= -\frac{2}{15}\frac{P^2l^5}{E^2I^2}. \qquad (15)$$

when $x = l$.

The *increased* deflection due to the second load P will be

$$\delta_y = \frac{1}{3}\frac{Pl^3}{EI}. \qquad (16)$$

the same as for the first load P, the new value of KL being $2\delta_y$.

3. If a thin, uniform, circular, elastic ring be hung on a pin, and carry a weight P at its lower end, required the form after distortion.

The ring after distortion will be symmetrical in reference to its vertical diameter, and its highest and lowest elements will be horizontal; and the problem will be the same as that of half the ring with its upper end firmly fixed, and at its lower end a moment equal to that existing internally in the solid ring, and also a horizontal force acting in the same sense as in the solid ring, retaining the lower end in the vertical diameter, and carrying $\frac{1}{2}P$. There will be no shearing stress at A or B. Let Fig. 267 represent the case, and

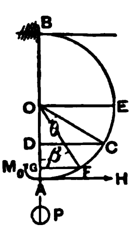


FIG. 267.

M_0 be the moment of internal stress at A.

H, the horizontal force at A.

w, the weight of unity of length of the ring.
r, the radius of the centre line of the ring $= OB$.
θ, the angle AOC.
β, the angle AOF.
ρ, the radius of curvature of the ring after distortion.
φ, the change of direction of the tangent at any point due to distortion.
s, the arc measured from A.
I, the moment of inertia of a transverse section.
E, the modulus of elasticity.
M, the moment of internal stress at any point as C.

Then

$$rd\theta = ds = rd\beta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

$$y = DG = r\,(\cos\beta - \cos\theta) \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

$$x = DC - GF = r\,(\sin\theta - \sin\beta) \quad . \quad . \quad . \quad . \quad . \quad (19)$$

The moment of the weight of the ring between A and C in reference to C will be

$$\int_0^\theta r\;(\sin\theta - \sin\beta)\;wrd\beta = wr^2\,(\theta\sin\theta + \cos\theta - 1)\,. \quad (20)$$

The moment of all the forces between A and C will be, considering those moments as positive which increase the radius of curvature,

$$M = -M_0 + rH(1-\cos\theta) + \tfrac{1}{2}P.\,r\sin\theta + wr^2\,(\theta\sin\theta + \cos\theta - 1) \quad . \quad . \quad . \quad (21)$$

Equations (10) and (21) give, for the change of direction of the curve from F to C:

$$EI\varphi = r\Big[\,(-M_0 + rH)\,(\theta - \beta) - rH(\sin\theta - \sin\beta) - \tfrac{1}{2}Pr\,(\cos\theta - \cos\beta) + wr^2(-\theta\cos\theta + \beta\cos\beta + 2\sin\theta - 2\sin\beta - (\theta - \beta)\Big] \quad . \quad . \quad . \quad (22)$$

Reckoning from the highest point, where the curve necessarily remains horizontal, θ becomes π, and the preceding equation becomes

$$EI\varphi_\beta = r\Big[(-M_0 + rH)\,(\pi - \beta) + rH\sin\beta + \tfrac{1}{2}Pr\,(1 + \cos\beta) + wr^2\,(\beta\cos\beta - \sin\beta + \beta)\Big] \quad . \quad . \quad . \quad (23)$$

If the origin of coördinates be at A, x in the preceding equations must be changed to $l - x$.

2. Let the beam be initially curved, being that produced by the weight P in the preceding exercise; then add another weight P.

To find y, let the origin be at L; we have, by means of rectangular coördinates, the well-known equation,

$$EI\frac{d^2y}{dx^2} = Px = M,$$

which integrated twice will give

$$y = \frac{P}{6EI}(x^3 - 3l^2x),$$

which substituted in equation (12) above, and still considering $ds = dx$, gives

$$\delta_x = \frac{P^2}{6E^2I^2}\left(\frac{x^5}{5} - l^2x^3\right),$$

$$= -\frac{2}{15}\frac{P^2l^5}{E^2I^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

when $x = l$.

The *increased* deflection due to the second load P will be

$$\delta_y = \frac{1}{3}\frac{Pl^3}{EI}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

the same as for the first load P, the new value of KL being $2\delta_y$

3. If a thin, uniform, circular, elastic ring be hung on a pin, and carry a weight P at its lower end, required the form after distortion.

The ring after distortion will be symmetrical in reference to its vertical diameter, and its highest and lowest elements will be horizontal; and the problem will be the same as that of half the ring with its upper end firmly fixed, and at its lower end a moment equal to that existing internally in the solid ring, and also a horizontal force acting in the same sense as in the solid ring, retaining the lower end in the vertical diameter, and carrying $\frac{1}{2}P$. There will be no shearing stress at A or B. Let Fig. 267 represent the case,

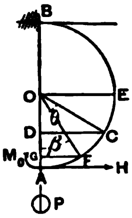


FIG. 267.

and

M_0 be the moment of internal stress at A.

H, the horizontal force at A.

w, the weight of unity of length of the ring.
r, the radius of the centre line of the ring $= OB$.
θ, the angle AOC.
β, the angle AOF.
ρ, the radius of curvature of the ring after distortion.
φ, the change of direction of the tangent at any point due to distortion.
s, the arc measured from A.
I, the moment of inertia of a transverse section.
E, the modulus of elasticity.
M, the moment of internal stress at any point as C.

Then

$$rd\theta = ds = rd\beta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

$$y = DG = r(\cos\beta - \cos\theta) \quad . \quad . \quad . \quad . \quad . \quad (18)$$

$$x = DC - GF = r(\sin\theta - \sin\beta) \quad . \quad . \quad . \quad . \quad . \quad (19)$$

The moment of the weight of the ring between A and C in reference to C will be

$$\int_o^\theta r(\sin\theta - \sin\beta)\, wrd\beta = wr^2(\theta\sin\theta + \cos\theta - 1) \,. \quad (20)$$

The moment of all the forces between A and C will be, considering those moments as positive which increase the radius of curvature,

$$M = -M_0 + rH(1-\cos\theta) + \tfrac{1}{2}P.\, r\sin\theta + wr^2(\theta\sin\theta + \cos\theta - 1) \quad . \quad . \quad . \quad . \quad (21)$$

Equations (10) and (21) give, for the change of direction of the curve from F to C:

$$EI\varphi = r\Big[(-M_0 + rH)(\theta - \beta) - rH(\sin\theta - \sin\beta) - \tfrac{1}{2}Pr(\cos\theta - \cos\beta) + wr^2(-\theta\cos\theta + \beta\cos\beta + 2\sin\theta - 2\sin\beta - (\theta - \beta))\Big] \quad . \quad . \quad . \quad (22)$$

Reckoning from the highest point, where the curve necessarily remains horizontal, θ becomes π, and the preceding equation becomes

$$EI\varphi_\beta = r\Big[(-M_0 + rH)(\pi - \beta) + rH\sin\beta + \tfrac{1}{2}Pr(1 + \cos\beta) + wr^2(\beta\cos\beta - \sin\beta + \beta)\Big] \quad . \quad . \quad . \quad (23)$$

The conditions of the problem require that the sum of all the changes of direction from B to A shall be zero, since the curve remains horizontal at A, hence making $\beta = o$, in the preceding equation, there results

$$(-M_0 + rH)\,\pi + Pr = o \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (24)$$

$$\therefore M_o = r\left(H + \frac{P}{\pi}\right)$$

Equations (12) and (18) give

$$EI\delta_x = \int_\beta^\theta Mr\,(\cos\beta - \cos\theta)\,rd\theta \quad . \; . \; . \quad (25)$$

But the distortion δ_x will, in practice, be computed from the fixed end B, for which point $\theta = \pi$. Substituting from (21), integrating, making π the superior limit, we find,

$$EI\delta_x = r^2 \left\{ \begin{array}{l} (-\pi M_o + \pi r H + \frac{1}{2} Pr)\cos\beta \\ + (M_o - r H)(\beta\cos\beta - \sin\beta) \\ + \frac{1}{2} rH(\pi - \beta + \frac{1}{2}\sin 2\beta) \\ + \frac{1}{4} Pr\,(2\cos^2\beta + \sin^2\beta) \\ - wr^2\,[\frac{5}{8}\sin 2\beta - \beta\,(1 - \frac{1}{2}\sin^2\beta) \\ \qquad - \beta\cos\beta + \sin\beta + \frac{1}{4}(\pi - \beta)] \end{array} \right\} . \quad (26)$$

Since the lowest end remains in the same vertical, δ_x will be zero for $\beta = o$, giving

$$-\pi M_o + \frac{3}{2}\pi rH + Pr - \frac{1}{4}\pi wr^2 = o \; . \; . \; . \quad (27)$$

Equations (24) and (27) give

$$H = \frac{1}{2} wr \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (28)$$

$$M_o = \frac{1}{2} wr^2 + P\frac{r}{\pi} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (29)$$

These values substituted in (21), changing θ to β, (23) and (26) give

$$M = wr^2\,[\beta\sin\beta + \frac{1}{2}\cos\beta - 1] + \frac{1}{2} Pr\left[\sin\beta - \frac{2}{\pi}\right] \quad (30)$$

$$EI\,\varphi_\beta = wr^3\,[\sin\beta - \beta\cos\beta + \frac{1}{2}\sin\beta - \beta] + \frac{1}{2} Pr^2$$
$$\left[\cos\beta - 1 + 2\frac{\beta}{\pi}\right] \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (31)$$

$$EI\,\delta_x = w\,r^4\,[\beta\cos\beta\,(1 + \frac{1}{2}\cos\beta) - \sin\beta\,(1 + \cos\beta) + \frac{1}{2}\beta]$$
$$+ {}^1 Pr^3\,[1 - \cos\beta + \frac{2}{\pi}(\beta\cos\beta - \sin\beta) - \frac{1}{2}\sin 2\beta] \; . \quad (32)$$

Similarly,

$$\left.\begin{aligned} EI\delta_y = wr^4 \,&[\tfrac{1}{4}(\pi^2 - \beta^2) - \cos\beta\,(1 + \cos\beta) \\ &- \beta \sin\beta\,(1 + \tfrac{1}{2}\cos\beta)] \\ &+ \tfrac{1}{2} Pr^3\, [{}^1 (\pi - \beta) + \tfrac{1}{2}\sin 2\beta \\ &+ \sin\beta - \frac{2}{\pi}(1 + \cos\beta + \beta \sin\beta)] \end{aligned}\right\} \quad . \; . \; . \quad (33)$$

If P acted upward its moment would be negative, and the signs of the terms containing P would be changed.

If the lower end of the ring were fixed, and the weight P were placed at the upper end of the vertical diameter, the preceding equations would be applicable to compression by changing all the algebraic signs.

The problem naturally divides itself into two parts: one in which P is zero, and the other in which the weight of the ring is neglected.

4. Let $P = 0$, and $B = 0$;

Then (33) becomes

$$\delta_y = 0.4674 \frac{wr^4}{EI} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (34)$$

which will be a practical formula for determining the elongation of the vertical diameter when the ring is hung on a pin; or of the shortening of that diameter when it stands on a plane.

5. Let the ring be a cast-iron cylinder, $E = 19{,}900{,}000$ lbs., thickness t, length b, then $I = \frac{1}{12} bt^3$, and $w = 0.26\, bt$; and equation (34) becomes

$$\delta_y = 0.000{,}000{,}071{,}48 \frac{r^4}{t^3} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (35)$$

If the ring, or cylinder, be 100 inches in diameter, and one inch thick, the elongation will be

$$\delta_y = 0.447 \text{ of an inch} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (36)$$

if hung on a pin; or, if it rests on a plane, the vertical diameter will be shortened that amount.

6. Let the weight of the ring be neglected. Then

$$w = o,$$

and equations (30) to (33) become

$$M = \tfrac{1}{2} Pr \left[\sin\beta - \frac{2}{\pi}\right] \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (37)$$

$$EI\varphi_\beta = \tfrac{1}{2}\, Pr^2\left[\cos\beta - 1 + 2\,\frac{\beta}{\pi}\right]. \quad . \quad . \quad . \quad (38)$$

$$EI\delta_x = \tfrac{1}{2}\, Pr^3\Big[1 - \cos\beta + \frac{2}{\pi}$$
$$(\beta\cos\beta - \sin\beta) - \tfrac{1}{2}\sin 2\beta\Big] \quad . \quad . \quad . \quad (39)$$

$$EI\delta_y = \tfrac{1}{2}\, Pr^3\Big[\tfrac{1}{2}(\pi - \beta) + \tfrac{1}{4}\sin 2\beta$$
$$+ \sin\beta - \frac{2}{\pi}(1 + \cos\beta + \beta\sin\beta)\Big]. \quad . \quad . \quad (40)$$

At the lowest point $\beta = o$, for which we have

$$M = -P\,\frac{r}{\pi}$$

$$\varphi_o = 0.$$

$$EI\delta_x = 0.$$

$$EI\delta_y = 0.14877\; Pr^3. \quad . \quad . \quad . \quad . \quad . \quad (41)$$

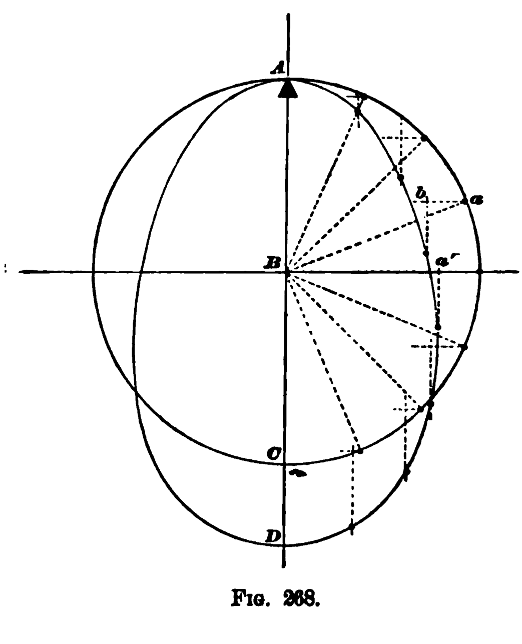


FIG. 268.

This determines the elongation of the vertical diameter, or the shortening of the same when the ring rests on a horizontal plane and the load P rests on the ring at the upper end of the diameter; and, hence, is a solution of a problem proposed some years since by Professor Sweet, in which it was proposed to find the depression of a ring partly supporting a valve.

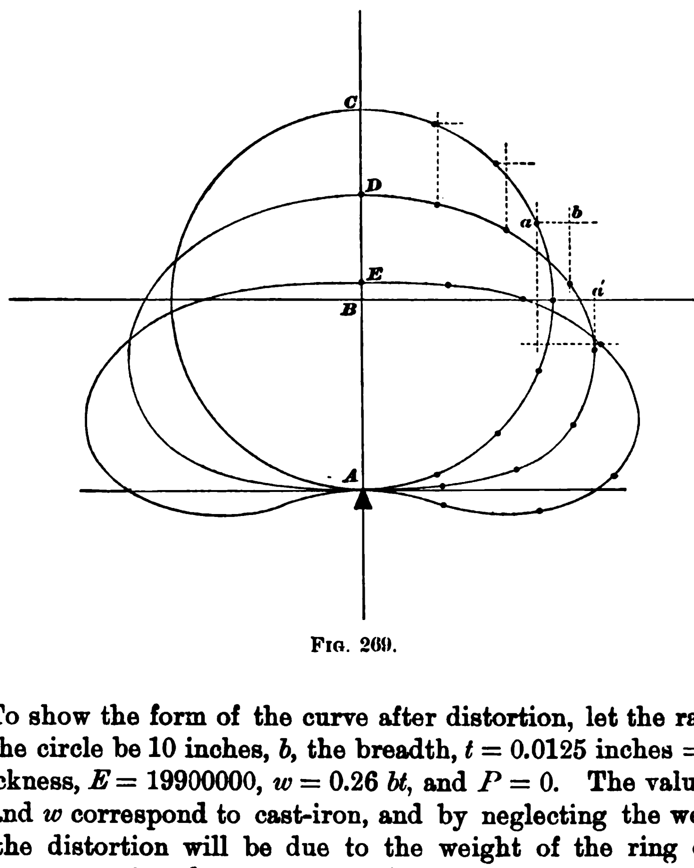


FIG. 269.

To show the form of the curve after distortion, let the radius of the circle be 10 inches, b, the breadth, $t = 0.0125$ inches = the thickness, $E = 19900000$, $w = 0.26\ bt$, and $P = 0$. The values of E and w correspond to cast-iron, and by neglecting the weight P, the distortion will be due to the weight of the ring only. Also assume that the equations hold true for this case although the vertical diameter will be elongated 4.674 inches. The coefficients of functions β in equations (32) and (33) become so nearly 10 we will call them 10, and the equations will become

$$\delta_x = 10\left[\beta \cos\beta\,(1 + \tfrac{1}{2}\cos\beta) - \sin\beta\,(1 + \cos\beta) + \tfrac{1}{2}\beta\right]$$
$$\delta_y = 10\left[\tfrac{1}{4}(\pi^2 - \beta^2) - \cos\beta\,(1 + \cos\beta) - \beta\sin\beta\,(1 + \tfrac{1}{2}\cos\beta)\right]$$

By means of these equations the following table has been computed:

β	0	$\frac{1}{8}\pi$	$\frac{1}{4}\pi$	$\frac{3}{8}\pi$	$\frac{1}{2}\pi$	$\frac{5}{8}\pi$	$\frac{3}{4}\pi$	$\frac{7}{8}\pi$	π
δ_x	0	−0.098	−0.630	−1.510	−2.146	−1.962	−1.060	−0.207	0
δ_y	4.674	4.323	3.549	3.116	2.804	2.743	2.088	0.841	0

In Fig. 268, let AEC be the circle whose radius is 10 inches, and $CBa = \frac{5}{8}\pi$; then will

$$ab = -1.962, \quad ba' = 2.743,$$

giving the point a' after distortion; and similarly for other points, by means of which the oval AD may be constructed.

Changing the signs of δ_x and δ_y in the preceding table, makes it applicable to the case of compression, as shown in Fig. 269, in which ab will be laid off to the right of a, and ba' will be downwards towards the fixed point A. We will have

$$ab = 2.146, \quad ba' = -2.804,$$

and similarly for other points.

In Fig. 269, the curve AE is the depression of a ring of half the thickness of AD.

The useful work per second is that employed in propelling the vessel, and is equal to Fv. The wasted work is the energy left in the water, which gradually changes into heat as the water comes to rest again. It is equal to half the mass multiplied by the square of the velocity with which the escaping water goes through the surrounding water, equal to $\frac{1}{2}\, VAw \div g\, (V-v)^2$. The sum of these two is the whole energy developed. Now $F =$ the momentum of the jet with respect to the vessel less the momentum of the feed-water with respect to the vessel, or $F = VAw \div g \times V - VAw \div g \times v = VAw \div g \times (V-v)$, which being multiplied by v gives $VAw \div g \times v\, (V-v)$ for the useful work.

Adding the useful and wasted work, we get $\frac{1}{2}\, VAw \div g \times (V^2-v^2)$. This is equal, as it should be, to the difference of the energies (reckoned with reference to the vessel) of the water leaving and entering the vessel.

The pounds pressure pushing the vessel ahead is equal to the difference of the reactions (or momenta) of the out-going and in-coming jets, regarding the feed-water as a jet entering the vessel in front. There is an interesting fact with reference to the reaction of a jet. Suppose at first a tank is closed up and you have a pressure of 100 per square inch near the bottom inside. Now let a hole be opened of 1 square inch section so that a jet of water flows out, this will make a difference of 200 pounds on that wall of the tank in which the hole is made; 100 pounds because the wall is cut away, and another 100 pounds because the pressure on the rest of the wall near the hole decreases.

In reply to Mr. Suplee: If the propeller in an ordinary ship is surrounded by a tube—we won't say a conical tube, but a parallel one—it will prevent the propeller giving the water any radial velocity. Therefore, if we could put a tube around the propeller, and have no friction in that tube, we would gain something; but the friction in the tube would probably be the element which would bring the speed of the vessel down. In reference to the remarks as to the impossibility of running a vessel by a jet on account of using a pumping engine, why, every propeller boat is run by a jet; it has simply a different kind of a pump. A propeller is a pump, then, to give the velocity backwards, and the only difference between ordinary propellers and these schemes for running a small jet is, that the propeller is large enough to propel the boat economically and the jet is not, Clark's valuable work, *The Steam Engine*, has been published.* In the chapter on Mechanical Stokers, the author has so fully covered the ground, as far as he goes, that, not to follow too closely in his footsteps, but brief reference will be made to the leading types of English stokers. The expression "as far as he goes" is used advisedly, for the reason that, although he has given by far the most comprehensive treatise yet published on this subject, still he furnishes no information regarding the improvements in mechanical stokers later than 1881, and makes no reference whatever to American stokers which have been brought out since that date—a period which covers substantially the history of successful mechanical stoking in this country.

The earliest form of mechanical stoker was probably the one patented in 1785, by James Watt, the inventor of the steam-engine. It was a simple device to push the coal, after it was coked at the front end of the grate, back towards the bridge. It was worked intermittently by levers, and was designed primarily to prevent smoke from bituminous coal.

The next mechanical stoker of which there is any record was invented by Wm. Brunton, of Birmingham, Eng., and which he patented June 29, 1819. It consisted of a circular grate revolving on a central vertical spindle, and a hopper with a toothed roller so placed as to discharge the coal upon the slowly-revolving grate. This device is said to have worked quite satisfactorily.

Soon after, in 1822, John Stanley patented a stoker consisting of a hopper on the front of the boiler, with crushing rollers, and rapidly-revolving fans for distributing the coal over the grate. Later Mr. J. G. Bodmer patented, May 24, 1834, and October 5, 1843, a form of mechanical stoker in which the coal was fed from a hopper on to an ingeniously designed grate, in which the fire bars were slowly moved inward, and on reaching the bridge were dropped in sections and returned to the front of the furnace by means of return screws. This device, though ingenious, was complicated and too liable to get out of order to be practicable.

After the year 1810 many styles of mechanical stokers were

---

* *The Steam Engine: A Treatise on Steam Engines and Boilers.* By Daniel Kinnear Clark, Hon. Mem. A. S. M. E. Blackie & Son, London, Glasgow, Edinburgh, and New York, 1890.

CCCCLVI.*

FLEXURE OF ELASTIC RINGS.

BY DE VOLSON WOOD, HOBOKEN, N. J.
(Member of the Society.)

WE first establish the general equation of flexure for a piece initially bent. Let IO be the neutral line of the piece $AJFE$, whose length IO is infinitesimal; ρ_1 the initial radius of curvature at O, ρ the radius of curvature after it is bent by an external force. Let the moment of the bending stress at O M, AJ the transverse section of the beam normal to the axis IO, BD parallel to AJ, EF a section originally normal to the axis IO, and GH a normal after final bending.

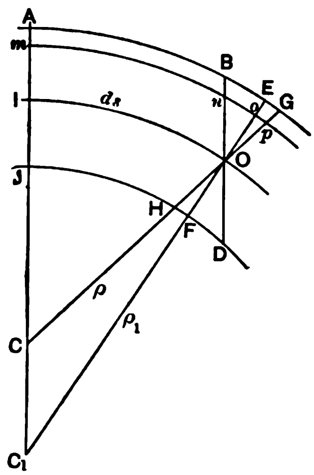


FIG. 264.

Let mo be any element of the initially bent beam at a distance $y = Oo$ from the neutral axis, then will $op = \lambda$ be the elongation due to the moment M, and no the excess of the initial length mo over that of IO.

We have

$$\frac{ds}{\rho_1} = \frac{no}{y},$$

$$\frac{ds}{\rho} = \frac{np}{y};$$

$$\therefore np - no = \lambda = \left(\frac{1}{\rho} - \frac{1}{\rho_1}\right) yds \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Vol. XII. of the *Transactions*.

some thirty patents taken out, but were all abandoned on account of inherent difficulties.*

In the class of mechanical stokers of which the Henderson stoker is an example, the coal was scattered over the fire by revolving fans or discs, or by shovels actuated by springs, which were put in tension by cams on a shaft across the front of the boiler. The other leading stokers of this type are, the Bennis, the Proctor, the Hodgkinson, the Barker, and the Whittaker & Newton, and were all patented between 1841 and 1875.

The Henderson stoker consists primarily of two horizontal fans revolving on vertical spindles, which scatter the coal over the fire. The coal is supplied to these fans by a toothed roller which crushes the larger lumps, and delivers it through an opening between this roller and a plate forming the front of the stoker hopper. The size of this opening is regulated by a screw, thereby controlling the feed of coal to the furnace. The Bennis and Proctor stokers use shovels actuated by springs and cams, so arranged as to give different degrees of tension to the springs, and different velocities to the successive strokes or throws of the shovel. The purpose of this variation in the throw of the shovel is to distribute the coal uniformly over the fire. The grate bars in the Henderson and Bennis stokers have a longitudinal reciprocating motion, for the purpose of breaking up the clinker, and carrying it forward to the bridge. The Proctor stoker accomplishes the same result in a measure, by an up-and-down movement of the front ends of each alternate grate bar. With the Bennis stoker the coal is fed to the shovels by a crushing roller, operating in a similar manner to the Henderson, the only practical difference between these two stokers being in the method by which the coal is distributed over the fire. The Barker, and the Whittaker & Newton stokers are later modifications of the Henderson stoker, and differ but little from it. The Hodgkinson stoker crushes and distributes the coal over the fire by means of a ribbed roller in the bottom of a hopper on the boiler front. This roller, running at a high rate of speed—800 revolutions and upwards per minute—is intended to scatter the coal in a fine shower over the fire. The mechanical objections to a shaft running at this high speed on the front of a boiler, and in the dust of a fire-room, are self-evident.

James Newton invented, in 1879, a stoker, by which the slack

* See *The Steam Engine*, Vol. I., page 338.

then $$KL = zd\varphi,\ KM = z \sin\theta d\varphi,\ LM = z\cos\theta d\varphi, \quad . \quad . \quad (7)$$

$$x = z\sin\theta,\ y = z\cos\theta;$$

$$\therefore\ KM = xd\varphi,\ LM = yd\varphi. \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Let δ_x be the x-component of the displacement, and similarly δ_y the y-component, due to the moment M; then, if the change be infinitesimal, equations (5) and (8) give

$$LM = d\delta_x = \frac{My}{EI}ds,\ KM = d\delta_y = \frac{M_x}{EI}ds \quad . \quad . \quad . \quad (9)$$

The displacement of K due to several bending moments between K and I, if expressed by a continuous function, will be, equations (5), (7), and (9),

$$\varphi = \int\frac{Mds}{EI} \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

$$KL = \int\frac{Mzds}{EI} \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

$$\delta_x = \int\frac{Myds}{EI} \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

$$\delta_y = \int\frac{Mxds}{EI} \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

These equations are also applicable to beams initially straight.

APPLICATIONS.

1. Let the beam be prismatic, initially straight and horizontal, fixed at one end and loaded at the free end.

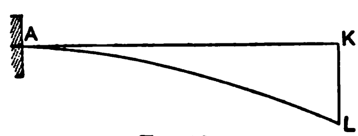


FIG. 266.

Let P represent the load, l the length of the beam; then, the origin being at K,

$$ds = dx,\ M = Px,\ y = o,\ z = x,$$

and the preceding equations give

$$\varphi = \frac{P}{EI}\int_o^l xdx = \tfrac{1}{2}\frac{Pl^2}{EI};\ KL = \delta_y = \tfrac{1}{3}\frac{Pl^3}{EI};\ \delta_x = o \quad . \quad . \quad . \quad (14)$$

These are well-known results.

If the origin of coördinates be at A, x in the preceding equations must be changed to $l - x$.

2. Let the beam be initially curved, being that produced by the weight P in the preceding exercise; then add another weight P.

To find y, let the origin be at L; we have, by means of rectangular coördinates, the well-known equation,

$$EI\frac{d^2y}{dx^2} = Px = M,$$

which integrated twice will give

$$y = \frac{P}{6EI}(x^3 - 3l^2x),$$

which substituted in equation (12) above, and still considering $ds = dx$, gives

$$\delta_x = \frac{P^2}{6E^2I^2}\left(\frac{x^5}{5} - l^2x^3\right),$$

$$= -\frac{2}{15}\frac{P^2l^5}{E^2I^2}. \qquad (15)$$

when $x = l$.

The *increased* deflection due to the second load P will be

$$\delta_y = \frac{1}{3}\frac{Pl^3}{EI}. \qquad (16)$$

the same as for the first load P, the new value of KL being $2\delta_y$.

3. If a thin, uniform, circular, elastic ring be hung on a pin, and carry a weight P at its lower end, required the form after distortion.

The ring after distortion will be symmetrical in reference to its vertical diameter, and its highest and lowest elements will be horizontal; and the problem will be the same as that of half the ring with its upper end firmly fixed, and at its lower end a moment equal to that existing internally in the solid ring, and also a horizontal force acting in the same sense as in the solid ring, retaining the lower end in the vertical diameter, and carrying $\frac{1}{2}P$. There will be no shearing stress at A or B. Let Fig. 267 represent the case, and

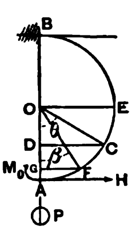


FIG. 267.

M_0 be the moment of internal stress at A.

H, the horizontal force at A.

w, the weight of unity of length of the ring.
r, the radius of the centre line of the ring $= OB$.
θ, the angle AOC.
β, the angle AOF.
ρ, the radius of curvature of the ring after distortion.
φ, the change of direction of the tangent at any point due to distortion.
s, the arc measured from A.
I, the moment of inertia of a transverse section.
E, the modulus of elasticity.
M, the moment of internal stress at any point as C.

Then

$$rd\theta = ds = rd\beta \quad \ldots \quad (17)$$

$$y = DG = r(\cos\beta - \cos\theta) \quad \ldots \quad (18)$$

$$x = DC - GF = r(\sin\theta - \sin\beta) \quad \ldots \quad (19)$$

The moment of the weight of the ring between A and C in reference to C will be

$$\int_0^\theta r(\sin\theta - \sin\beta)\, wrd\beta = wr^2(\theta\sin\theta + \cos\theta - 1)\,. \quad (20)$$

The moment of all the forces between A and C will be, considering those moments as positive which increase the radius of curvature,

$$M = -M_0 + rH(1-\cos\theta) + \tfrac{1}{2}P.\, r\sin\theta + wr^2(\theta\sin\theta + \cos\theta - 1) \quad \ldots \quad (21)$$

Equations (10) and (21) give, for the change of direction of the curve from F to C:

$$EI\varphi = r\big[(-M_0 + rH)(\theta - \beta) - rH(\sin\theta - \sin\beta) - \tfrac{1}{2}Pr(\cos\theta - \cos\beta) + wr^2(-\theta\cos\theta + \beta\cos\beta + 2\sin\theta - 2\sin\beta - (\theta - \beta))\big] \quad \ldots \quad (22)$$

Reckoning from the highest point, where the curve necessarily remains horizontal, θ becomes π, and the preceding equation becomes

$$EI\varphi_\beta = r\big[(-M_0 + rH)(\pi - \beta) + rH\sin\beta + \tfrac{1}{2}Pr(1 + \cos\beta) + wr^2(\beta\cos\beta - \sin\beta + \beta)\big] \quad \ldots \quad (23)$$

The conditions of the problem require that the sum of all the changes of direction from B to A shall be zero, since the curve remains horizontal at A, hence making $\beta = o$, in the preceding equation, there results

$$(-M_0 + rH)\,\pi + Pr = o \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

$$\therefore M_o = r\left(H + \frac{P}{\pi}\right)$$

Equations (12) and (18) give

$$EI\delta_x = \int_\beta^\theta Mr\,(\cos\beta - \cos\theta)\,rd\theta \quad . \quad . \quad . \quad (25)$$

But the distortion δ_x will, in practice, be computed from the fixed end B, for which point $\theta = \pi$. Substituting from (21), integrating, making π the superior limit, we find,

$$EI\delta_x = r^2 \left\{ \begin{array}{l} (-\pi M_o + \pi r H + \tfrac{1}{2} Pr)\cos\beta \\ + (M_o - r H)(\beta\cos\beta - \sin\beta) \\ + \tfrac{1}{2} rH(\pi - \beta + \tfrac{1}{2}\sin 2\beta) \\ + \tfrac{1}{4} Pr\,(2\cos^2\beta + \sin^2\beta) \\ - wr^2\,[\tfrac{5}{8}\sin 2\beta - \beta\,(1 - \tfrac{1}{2}\sin^2\beta) \\ \qquad - \beta\cos\beta + \sin\beta + \tfrac{1}{4}(\pi - \beta)] \end{array} \right\} . \quad (26)$$

Since the lowest end remains in the same vertical, δ_x will be zero for $\beta = o$, giving

$$-\pi M_o + \tfrac{3}{2}\pi rH + Pr - \tfrac{1}{4}\pi wr^2 = o \quad . \quad . \quad . \quad (27)$$

Equations (24) and (27) give

$$H = \tfrac{1}{2}wr \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

$$M_o = \tfrac{1}{2}wr^2 + P\frac{r}{\pi} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

These values substituted in (21), changing θ to β, (23) and (26) give

$$M = wr^2\,[\beta\sin\beta + \tfrac{1}{2}\cos\beta - 1] + \tfrac{1}{2}Pr\,\left[\sin\beta - \frac{2}{\pi}\right] \quad (30)$$

$$EI\,\varphi_\beta = wr^3\,[\sin\beta - \beta\cos\beta + \tfrac{1}{2}\sin\beta - \beta] + \tfrac{1}{2}Pr^2 \left[\cos\beta - 1 + 2\frac{\beta}{\pi}\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

$$EI\,\delta_x = w\,r^4\,[\beta\cos\beta\,(1 + \tfrac{1}{2}\cos\beta) - \sin\beta\,(1 + \cos\beta) + \tfrac{1}{2}\beta] + {}^1 Pr^3\,\left[1 - \cos\beta + \frac{2}{\pi}(\beta\cos\beta - \sin\beta) - \tfrac{1}{2}\sin 2\beta\right] \quad . \quad (32)$$

Similarly,

$$\left.\begin{aligned} EI\delta_y = wr^4 & [\tfrac{1}{4}(\pi^2 - \beta) - \cos\beta\,(1 + \cos\beta) \\ & - \beta\sin\beta\,(1 + \tfrac{1}{2}\cos\beta)] \\ & + \tfrac{1}{2} Pr^3 [{}^1(\pi - \beta) + \tfrac{1}{4}\sin 2\beta \\ & + \sin\beta - \frac{2}{\pi}(1 + \cos\beta + \beta\sin\beta)] \end{aligned}\right\} \quad . \quad . \quad (33)$$

If P acted upward its moment would be negative, and the signs of the terms containing P would be changed.

If the lower end of the ring were fixed, and the weight P were placed at the upper end of the vertical diameter, the preceding equations would be applicable to compression by changing all the algebraic signs.

The problem naturally divides itself into two parts: one in which P is zero, and the other in which the weight of the ring is neglected.

4. Let $P = 0$, and $B = 0$;

Then (33) becomes

$$\delta_y = 0.4674\,\frac{wr^4}{EI} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

which will be a practical formula for determining the elongation of the vertical diameter when the ring is hung on a pin; or of the shortening of that diameter when it stands on a plane.

5. Let the ring be a cast-iron cylinder, $E = 19{,}900{,}000$ lbs., thickness t, length b, then $I = \frac{1}{12}bt^3$, and $w = 0.26\,bt$; and equation (34) becomes

$$\delta_y = 0.000{,}000{,}071{,}48\,\frac{r_4}{t_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

If the ring, or cylinder, be 100 inches in diameter, and one inch thick, the elongation will be

$$\delta_y = 0.447 \text{ of an inch} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (36)$$

if hung on a pin; or, if it rests on a plane, the vertical diameter will be shortened that amount.

6. Let the weight of the ring be neglected. Then

$$w = o,$$

and equations (30) to (33) become

$$M = \tfrac{1}{2} Pr\left[\sin\beta - \frac{2}{\pi}\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

$$EI\varphi_\beta = \tfrac{1}{2} Pr^2\left[\cos\beta - 1 + 2\frac{\beta}{\pi}\right]. \quad . \quad . \quad . \quad .(38)$$

$$EI\delta_x = \tfrac{1}{2} Pr^3\left[1 - \cos\beta + \frac{2}{\pi}\right.$$
$$\left.(\beta\cos\beta - \sin\beta) - \tfrac{1}{2}\sin 2\beta\right] \quad . \quad . \quad . \quad (39)$$

$$EI\delta_y = \tfrac{1}{2} Pr^3\left[\tfrac{1}{2}(\pi - \beta) + \tfrac{1}{4}\sin 2\beta\right.$$
$$\left. + \sin\beta - \frac{2}{\pi}(1 + \cos\beta + \beta\sin\beta)\right]. \quad . \quad . \quad (40)$$

At the lowest point $\beta = o$, for which we have

$$M = -P\frac{r}{\pi}$$

$$\varphi_o = 0.$$

$$EI\delta_x = 0.$$

$$EI\delta_y = 0.14877\ Pr^3. \quad . \quad . \quad . \quad . \quad . \quad (41)$$

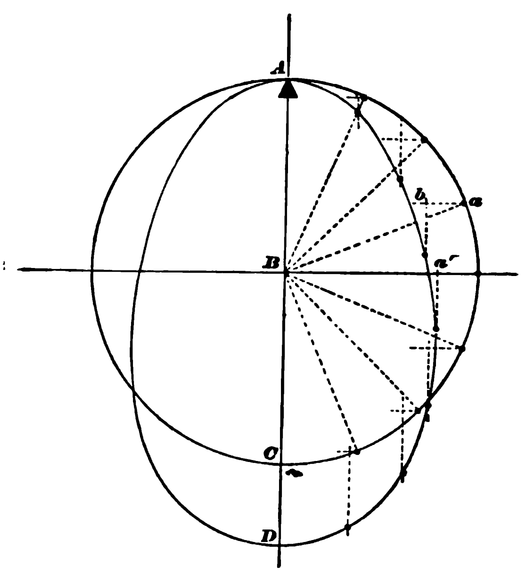


FIG. 268.

This determines the elongation of the vertical diameter, or the shortening of the same when the ring rests on a horizontal plane and the load P rests on the ring at the upper end of the diameter; and, hence, is a solution of a problem proposed some years since by Professor Sweet, in which it was proposed to find the depression of a ring partly supporting a valve.

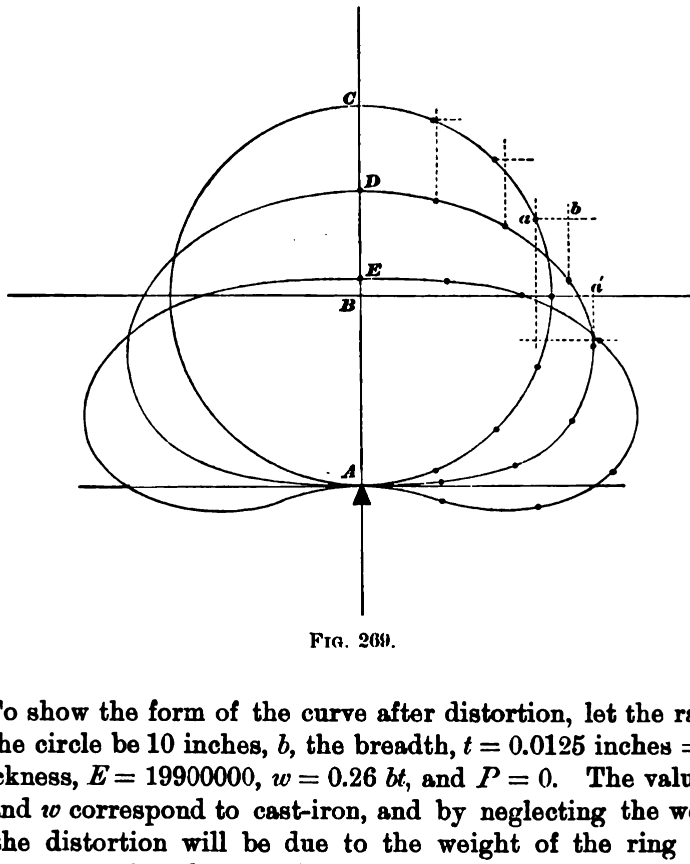


FIG. 269.

To show the form of the curve after distortion, let the radius of the circle be 10 inches, b, the breadth, $t = 0.0125$ inches = the thickness, $E = 19900000$, $w = 0.26\,bt$, and $P = 0$. The values of E and w correspond to cast-iron, and by neglecting the weight P, the distortion will be due to the weight of the ring only. Also assume that the equations hold true for this case although the vertical diameter will be elongated 4.674 inches. The coefficients of functions β in equations (32) and (33) become so nearly 10 we will call them 10, and the equations will become

$$\delta_x = 10\left[\beta \cos\beta\,(1 + \tfrac{1}{2}\cos\beta) - \sin\beta\,(1 + \cos\beta) + \tfrac{1}{2}\beta\right]$$
$$\delta_y = 10\left[\tfrac{1}{4}(\pi^2 - \beta^2) - \cos\beta\,(1 + \cos\beta) - \beta\sin\beta\,(1 + \tfrac{1}{2}\cos\beta)\right]$$

By means of these equations the following table has been computed:

β	0	$\frac{1}{8}\pi$	$\frac{1}{4}\pi$	$\frac{3}{8}\pi$	$\frac{1}{2}\pi$	$\frac{5}{8}\pi$	$\frac{3}{4}\pi$	$\frac{7}{8}\pi$	π
δ_x	0	−0.098	−0.630	−1.510	−2.146	−1.962	−1.060	−0.207	0
δ_y	4.674	4.323	3.549	3.116	2.804	2.743	2.088	0.841	0

In Fig. 268, let AEC be the circle whose radius is 10 inches, and $CBa = \frac{5}{8}\pi$; then will

$$ab = -1.962, \; ba' = 2.743,$$

giving the point a' after distortion; and similarly for other points, by means of which the oval AD may be constructed.

Changing the signs of δ_x and δ_y in the preceding table, makes it applicable to the case of compression, as shown in Fig. 269, in which ab will be laid off to the right of a, and ba' will be downwards towards the fixed point A. We will have

$$ab = 2.146, \; ba' = -2.804,$$

and similarly for other points.

In Fig. 269, the curve AE is the depression of a ring of half the thickness of AD.

CCCCLVII.*

MECHANICAL STOKERS.

BY WILLIAM R. RONEY, CHICAGO, ILL.
(Member of the Society.)

THE use of machinery for stoking fuel under boilers is quite old, and there have been many inventions in this direction.

The most extended use has been reached in England, where thousands are in service, built in many forms, but all particularly adapted to the stoking of coal on the grates of internally-fired boilers, where the grates are usually flat, long, and narrow. The most common forms of English stokers embody the features of throwing or pushing the coal from a recess in the boiler front over the grate, the mechanism in this recess being fed by a hopper. Travelling chain grates, or rocking or reciprocating parallel bars actuated by means of various devices, carry the coal from the feeding mechanism at the front end to an ash chamber in the rear.

Numerous modifications of these devices have been used with fair success; but, being designed especially for internally-fired boilers, these stokers where imported have thus far not been successful when applied to our externally-fired boilers.

We, as American engineers, are naturally more interested in the development and progress which have been made in this important department of engineering in this country than in what has been done abroad, since we have constantly to meet the conditions under which this progress has been made. But a paper on mechanical stokers would not be complete without some reference being made to what the pioneers in this field have accomplished; and I will therefore first consider chronologically the history of mechanical stokers across the water.

Since the writer began collecting data for this paper concerning the various forms of English and American stokers, D. K.

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Vol. XII. of the *Transactions*.

Clark's valuable work, *The Steam Engine*, has been published.* In the chapter on Mechanical Stokers, the author has so fully covered the ground, as far as he goes, that, not to follow too closely in his footsteps, but brief reference will be made to the leading types of English stokers. The expression "as far as he goes" is used advisedly, for the reason that, although he has given by far the most comprehensive treatise yet published on this subject, still he furnishes no information regarding the improvements in mechanical stokers later than 1881, and makes no reference whatever to American stokers which have been brought out since that date—a period which covers substantially the history of successful mechanical stoking in this country.

The earliest form of mechanical stoker was probably the one patented in 1785, by James Watt, the inventor of the steam-engine. It was a simple device to push the coal, after it was coked at the front end of the grate, back towards the bridge. It was worked intermittently by levers, and was designed primarily to prevent smoke from bituminous coal.

The next mechanical stoker of which there is any record was invented by Wm. Brunton, of Birmingham, Eng., and which he patented June 29, 1819. It consisted of a circular grate revolving on a central vertical spindle, and a hopper with a toothed roller so placed as to discharge the coal upon the slowly-revolving grate. This device is said to have worked quite satisfactorily.

Soon after, in 1822, John Stanley patented a stoker consisting of a hopper on the front of the boiler, with crushing rollers, and rapidly-revolving fans for distributing the coal over the grate. Later Mr. J. G. Bodmer patented, May 24, 1834, and October 5, 1843, a form of mechanical stoker in which the coal was fed from a hopper on to an ingeniously designed grate, in which the fire bars were slowly moved inward, and on reaching the bridge were dropped in sections and returned to the front of the furnace by means of return screws. This device, though ingenious, was complicated and too liable to get out of order to be practicable.

After the year 1840 many styles of mechanical stokers were

* *The Steam Engine: A Treatise on Steam Engines and Boilers.* By Daniel Kinnear Clark, Hon. Mem. A. S. M. E. Blackie & Son, London, Glasgow, Edinburgh, and New York, 1890.

patented in England, but nearly all were merely variations and modifications of the two forms of stokers patented by Mr. John Jukes in 1841, and by Mr. E. Henderson in 1843. The Jukes and Henderson stokers were, however, really the development of the crude ideas embodied in the Stanley and the Bodmer stokers already referred to.

The Jukes stoker consisted of longitudinal fire bars, connected by links, so as to form an endless chain, similar to the familiar tread-mill horse power. The small coal was delivered from a hopper on the front of the boiler, on to the grate, which, slowly moving from front to rear, gradually advanced the fuel into the furnace and discharged the ash and clinker at the back.

A large number of stokers were subsequently patented, designed to accomplish the same result, viz., to receive the fresh fuel at the front of the furnace and discharge the burnt-out residue at the back. In the most of them the grate bars extended lengthwise from front to back of the furnace, instead of crosswise as in the Jukes stoker, and the fuel was advanced by a slow reciprocating movement of the fire bars. This movement was produced in various ways by the different inventors, the usual method being by means of cams or cranks on a transverse shaft on the boiler front. The most successful of this type of stokers were the *Vicar's Mechanical Stoker*, patented in 1867, and *Knapp's Mechanical Stoker*, *McDougall's Mechanical Stoker*, and *Holroyd-Smith's Mechanical Stoker*—all patented subsequently to 1868.

The last-named stoker used, instead of reciprocating horizontal fire bars, three longitudinal troughs placed at right angles with another trough across the boiler front, from which they received the coal. The fuel was moved towards the back of the furnace by double threaded screws, one in each trough. These screws tapered from 5 to 2½ inches in diameter, and revolved at varying speeds, from 2 revolutions per minute upwards, as required.*

Somewhat similar in its operation to the Holroyd-Smith was an under-feed stoker, patented by James Frisbie, in 1844. In this machine the coal was pushed up through the center of the fire by means of a movable section of the grate working like a plunger. A number of modifications of this idea were tried, and

* See *The Steam Engine*, Vol. I., page 338.

some thirty patents taken out, but were all abandoned on account of inherent difficulties.*

In the class of mechanical stokers of which the Henderson stoker is an example, the coal was scattered over the fire by revolving fans or discs, or by shovels actuated by springs, which were put in tension by cams on a shaft across the front of the boiler. The other leading stokers of this type are, the Bennis, the Proctor, the Hodgkinson, the Barker, and the Whittaker & Newton, and were all patented between 1841 and 1875.

The Henderson stoker consists primarily of two horizontal fans revolving on vertical spindles, which scatter the coal over the fire. The coal is supplied to these fans by a toothed roller which crushes the larger lumps, and delivers it through an opening between this roller and a plate forming the front of the stoker hopper. The size of this opening is regulated by a screw, thereby controlling the feed of coal to the furnace. The Bennis and Proctor stokers use shovels actuated by springs and cams, so arranged as to give different degrees of tension to the springs, and different velocities to the successive strokes or throws of the shovel. The purpose of this variation in the throw of the shovel is to distribute the coal uniformly over the fire. The grate bars in the Henderson and Bennis stokers have a longitudinal reciprocating motion, for the purpose of breaking up the clinker, and carrying it forward to the bridge. The Proctor stoker accomplishes the same result in a measure, by an up-and-down movement of the front ends of each alternate grate bar. With the Bennis stoker the coal is fed to the shovels by a crushing roller, operating in a similar manner to the Henderson, the only practical difference between these two stokers being in the method by which the coal is distributed over the fire. The Barker, and the Whittaker & Newton stokers are later modifications of the Henderson stoker, and differ but little from it. The Hodgkinson stoker crushes and distributes the coal over the fire by means of a ribbed roller in the bottom of a hopper on the boiler front. This roller, running at a high rate of speed—800 revolutions and upwards per minute—is intended to scatter the coal in a fine shower over the fire. The mechanical objections to a shaft running at this high speed on the front of a boiler, and in the dust of a fire-room, are self-evident.

James Newton invented, in 1879, a stoker, by which the slack

* See *The Steam Engine*, Vol. I., page 338.

coal was propelled into the furnace by a blast of heated air, delivered at intervals, by means of a fan. This device cannot, however, be strictly classed among mechanical stokers as the term is ordinarily accepted.

A number of German mechanical stokers have been patented since the year 1850, but they are so similar in principle to the English stokers already described that it is hardly worth while to refer to them individually.

Although, in their progressive development, the English stokers already described were a great improvement over hand firing, still, from the standpoint of our present experience, they show faults, aside from their lack of adaptability to the types of boilers most used in this country.

These faults are, briefly:

First.—The mechanism is either of too complicated a nature, or running at too high a rate of speed, to keep in order long. In the case of the shovel stokers, they are dependent upon springs, which in time lose their tension, and the nice adjustment of the variable throw of the shovels fails, and as a result the coal is unevenly piled on the grates. Even when the springs and shovels are in order, a difference in size of the coal causes the dust to fall in one place and the lumps in another, and prevents uniform firing. In the case of stokers which distribute the coal by means of rapidly-revolving fans, discs, or drums, there are still more serious objections. Namely, the great wear consequent on running at such high speeds in coal dust and ashes, and where the high temperature of the bearings attached to the boiler front often precludes lubrication. Secondly, the continuous rain of fine coal, seeking the lowest level, falls into the air holes in the fire, and impedes the draft; while the larger and heavier pieces pile up at the front of the grate. Thirdly, the fan-like draft drives great quantities of dust and ashes into the tubes, thereby impeding the flow of gases, and reducing the efficiency of the boilers.

Second.—The grates are usually so placed as to be almost inaccessible, either for examining the condition of the fire, or slicing it, or for renewing worn-out parts when necessary. Being horizontal, the fire cannot be stirred from beneath, and the bars once in place, the grate openings cannot be changed according as the coal is coarse or fine, nor can the air supply be varied to suit different grades of coal. The mechanism of the grates and feeding devices is, in most cases, complicated and expensive to install and to keep in repair, especially where the machinery is moving in the fire. This complication is especially objectionable, considering the nature of the service and the place the stokers have to work in, as well as the character of labor usually employed in the fire-room. The simplest and most durable stoker which can be designed will receive hard usage in thehands of the average fireman.

After 1873 there were no further important improvements in mechanical stokers until 1878, when the Murphy Patent Furnace was brought out in this country by Thomas Murphy, of Detroit, Mich. This furnace, which he further improved upon in 1881 by the addition of a small engine for operating it, was the first distinctively American stoker, and is worthy of note on account of its differing so radically from the English types of internal-firing stokers, and because of its being adapted only to externally-fired boilers. The Murphy stoker consists of two coal magazines placed in the side walls of the boiler furnace, and extending back from the boiler front 6 or 7 feet. In the bottom of these magazines are rectangular iron boxes, which are moved from side to side by means of a rack and pinion, and serve to push the coal upon the grates, which incline at an angle of about 35° from the inner edge of the coal magazines, forming a V-shaped receptacle for the burning coal. The grates are composed of narrow parallel bars, so arranged that each alternate bar lifts about an inch at the lower end, while at the bottom of the V, and filling the space between the ends of the grate bars, is placed a cast-iron toothed bar, arranged to be turned by a crank. The purpose of this bar is to grind the clinker coming in contact with it. Any large pieces of clinker and cinder which the teeth will not reach are beaten down by a poker through a door in the front opposite the middle of the furnace, or hauled out on the floor in front of the boiler by means of hooks, and quenched with water. This door is also used when slicing the fire to keep the grate open, and in stirring and raking the coal when it fails to move down the grates, on account of burning up too closely to the magazines, or because of an accumulation of clinker in the V-shaped receptacle. Over this receptacle is sprung a fire-brick arch with each foot resting on one of the cast-iron coal magazines. This arch extends back under the boiler about one-third its length, and by thus preventing the hot gases from striking the shell, a practically smokeless combustion is obtained, except at such times when it is necessary to slice the fire or rake the coal as referred to above. A bituminous caking coal is best suited for this stoker, to prevent the sifting of fine coal between the moving grate bars, which takes place with semi-bituminous and non-caking coals.

In 1885 the Brightman Mechanical Stoker was patented by Mr. J. A. Brightman, of Cleveland, Ohio; and the same year the Roney Mechanical Stoker was brought out by the author of this paper, after a series of experiments extending over some five years. Just one hundred years after the first stoker was patented by Watt.

These two stokers, together with the Murphy, comprised the three types of mechanical stokers invented and now in use in this country.

The Brightman stoker differs from the Murphy in having the coal magazine or hopper on the front of the boiler, and in the form of the grate, which inclines from the inner edge of the coal hopper to the back part of the furnace at an angle of 34°. The grate is composed of parallel bars extending lengthwise the furnace, with projecting overlapping shelves, and each alternate bar having a reciprocating swinging motion for the purpose of moving the fuel down the grate. The coal is fed into the furnace by a plunger working in the bottom of the hopper, and when necessary the motion of the bars is supplemented by pushing the fuel down the grate by a poker through an opening provided in the hopper. The ash and clinker is drawn into the ash pit by means of a hook through an opening in the lower part of the grate, the larger masses of clinker being first broken up with a slice bar. The Brightman stoker possesses some advantages over the Murphy in having a simpler construction, but the long fire brick arch over the furnace enables the Murphy to maintain a more intense heat, and consequently is more nearly smokeless.

Before describing the other American stoker referred to above it may be interesting to consider briefly the principles of combustion, and the important part which mechanical stoking plays in producing the conditions necessary for the most economical combustion of coal.

The combustible elements of fuel are hydrogen, carbon, and sulphur. When coal is burned the oxygen of the air combines with the combustible matter, and the nitrogen, being neutral, merely serves to dilute the resultant heat. The products of this combustion are carbonic acid, water or steam, sulphurous acid, and nitrogen; and the quantity of heat evolved depends upon the proportions in which the combustible elements exist in the coal, and upon the completeness of the combustion.

To secure complete combustion the first condition necessary is a sufficient quantity of air; second, that the air and fuel, both gaseous and solid, shall be thoroughly mixed; and third, that the air and the combustible gases shall be maintained at a sufficiently high temperature—the higher the better for good combustion. Incomplete combustion, and consequently loss of heat, may result from the failure of any of these three conditions.

Generally speaking, a ton of good bituminous coal requires for complete combustion about 314,000 cubic feet of air, or 140 cubic feet per pound of coal. If the supply of air is too great, loss results from the increased weight of heated gases sent up the chimney, as well as from the cooling effect upon the boiler. If, on the other hand, the supply of air is insufficient, a much greater loss in heat results, due to incomplete combustion.

One pound of carbon with 11.6 lbs. of air, which is the exact quantity required for perfect combustion, will develop 14,500 B. T. U., producing a temperature of 4877 above initial temperature of 62° Fahr., the product being carbonic acid. But 1 lb. of carbon with half this weight of air will produce an invisible gas, carbonic oxide, developing 4452 B. T. U., and producing a temperature of about 2700° Fahr., assuming the specific heat of the gaseous products of combustion to be .:5 B. T. U. One pound of hydrogen with 34.8 lbs. of air, which contains the oxygen necessary for its combustion, the product being steam, will develop 62,000 B. T. U., resulting in a temperature of 5741°, above 62 Fahr.; but in consequence of the greater weight of products, the evaporative power of each pound of hydrogen is four times that of 1 lb. of carbon.

Assuming average good bituminous coal to contain—

Carbon	Hydrogen	Oxygen	Nitrogen	Sulphur	Ash
80	5	8	1 2	1.25	4

which is very nearly the actual composition of Virginia Pocahontas coal. If then, instead of 314,000 cubic feet of air necessary to consume perfectly one ton of good bituminous coal, we supply only 235,000, one-half of the carbon will only be burned to carbonic oxide, and instead of developing 14,700 B T. U. per pound of coal, there will be but 10,680 B. T. U., a loss of 27.3% in efficiency, although the same weight of fuel will be consumed. The evaporative power of the coal being taken at 15 lbs. and 4.09 lbs. deducted for this cause, the efficiency will be only 10.91 lbs. If from this we deduct the heat lost in the chimney, by radiation and absorbed by brickwork, etc., we have left 9 to 10 lbs. of evaporation from and at 212°, which is about that obtained in general practice.

That the air and gases should be thoroughly mixed is almost as important a condition for complete combustion as that there be a sufficient quantity of air present in the furnace. This fact is so well known that it is only necessary to mention it. The often-referred-to argand burner is probably the best known example of the beneficial effect of a thorough mixing of air with the gases of combustion. The very simple experiment of closing the small perforations around the wick and admitting the same quantity of air through one or two large openings, readily demonstrates the importance of this condition, and also shows that an excess of air is more easily prevented when the air and gases are brought together in such a manner that they are thoroughly mixed. Or if the exact quantity of air which is supplied to a coal fire in a given time through the perforations in the grate could be passed through a single large opening in the same length of time, the efficiency of the combustion would be reduced over one-half, due to the gases escaping up the chimney only partially consumed. This is what happens every time the fireman opens the furna e door to supply fresh fuel or clean the grates, to say nothing of the great volume of cold air which, rushing in, lowers the temperature of furnace and boiler.

From the results of careful observations it has been shown that in addition to the usual large surplus of air in the furnace, fully 38% *excess* of air over that required for combustion enters the furnace during the time the fire-door is opened for firing and cleaning. The effect of such an inrush of cold air over the flame is to lower the temperature of the gases below the point at which combustion takes place, viz., 1000°, when any unburned gases will pass into the flues and up the chimney, *invisible* but lost, exactly as if, should one gas-tap be turned on

in the bearer, and is capable of a rocking motion through an adjustable angle. All the grate-bars are coupled together by a "rocker bar," the notches of which engage with a lug on the lower rib of each grate-bar, pin connections being made with two of the grate-bars only, for the purpose of holding the

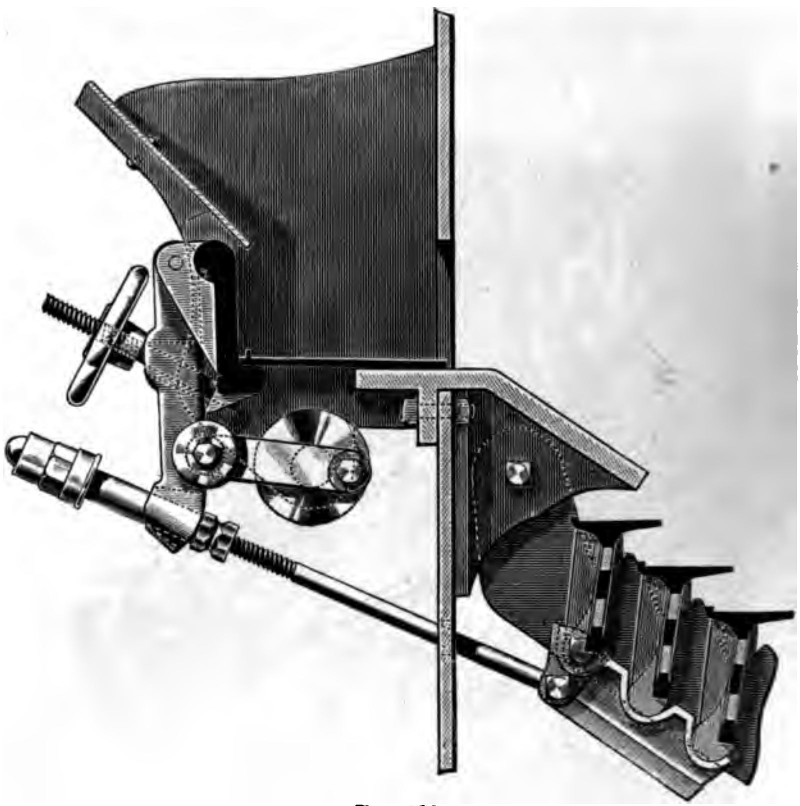


FIG. 273.

Stepped Position. Details of Feed and Grate Movement.

"rocker bar" in position. A variable back-and-forth motion being given to the "rocker bar," through a connecting rod, by a device to be hereafter described, the grate-bars necessarily rock in unison, now forming a series of steps (see Fig. 273), and now approximating to an inclined plane, with the grates partly overlapping like shingles on a roof. (See Fig. 272.)

This variable, shutter-like movement furnishes a simple means of regulating the quantity of air passing through the grate,

facturing demands greater economy in the production of power. Until quite recently the boiler-room has been too much neglected in the effort to improve the engine duty. How frequently we find a magnificent plant of engines supplied with steam from boilers which do not work within 25% of their possible economy. Too often the engineer thinks only of having sufficient steam to enable his engines to work economically, regardless of how the steam is produced; while the fireman, with his eye on the steam gauge, shovels in the coal, "knowing little and caring less" whether he is burning 2 lbs. or 4 lbs. of coal per horse-power. It is all right for engine-designers to multiply cylinders and raise steam pressures, but we must go back of the throttle valve and stop the waste in the fire-room, if we would realize the highest economy in the production of power.

Mechanical stoking naturally goes with improved engine practice. The compound engine and the mechanical stoker logically belong together. As boiler pressures increase, the character of the firing must improve to obtain the best results and enable high pressures to be carried safely. Not only does the fluctuating pressure seriously affect the engine economy, but the frequent and extreme changes in the temperature of the furnace, which are unavoidable in hand-firing, must in time render high pressures unsafe. A properly constructed mechanical stoker meets both these requirements: by supplying the coal automatically and regularly, and by avoiding all opening of doors for firing or cleaning, maintains an even high temperature without injury to the boiler.

In the application of stokers, a proper conception of their scope, benefits, and limitations should be clearly in the minds of all. The stoker is not a device to be applied for the purpose of producing fine results under bad conditions, nor a machine for turning the tide of a manufacturer's business from loss to profit. It is not a remedy for existing evils, such as too small boiler power, deficient draught, bad coal, and careless attendance. It is not built for the purpose of revolutionizing the processes of combustion, or introducing any new laws of nature. But give the stoker good, fair conditions as to boilers, draught, coal, etc., and it will give good results; and the better the conditions, the better will be the results.

The growing popularity of mechanical stoking is largely due to the fact that a good mechanical stoker not only lengthens the

in the bearer, and is capable of a rocking motion through an adjustable angle. All the grate-bars are coupled together by a "rocker bar," the notches of which engage with a lug on the lower rib of each grate-bar, pin connections being made with two of the grate-bars only, for the purpose of holding the

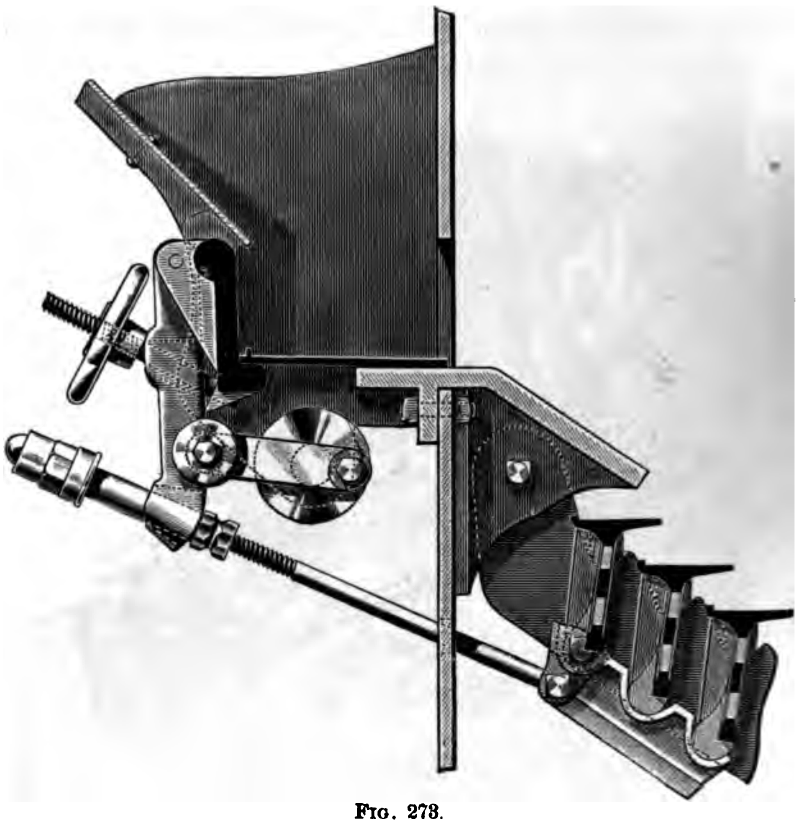


FIG. 273.

Stepped Position. Details of Feed and Grate Movement.

"rocker bar" in position. A variable back-and-forth motion being given to the "rocker bar," through a connecting rod, by a device to be hereafter described, the grate-bars necessarily rock in unison, now forming a series of steps (see Fig. 273), and now approximating to an inclined plane, with the grates partly overlapping like shingles on a roof. (See Fig. 272.)

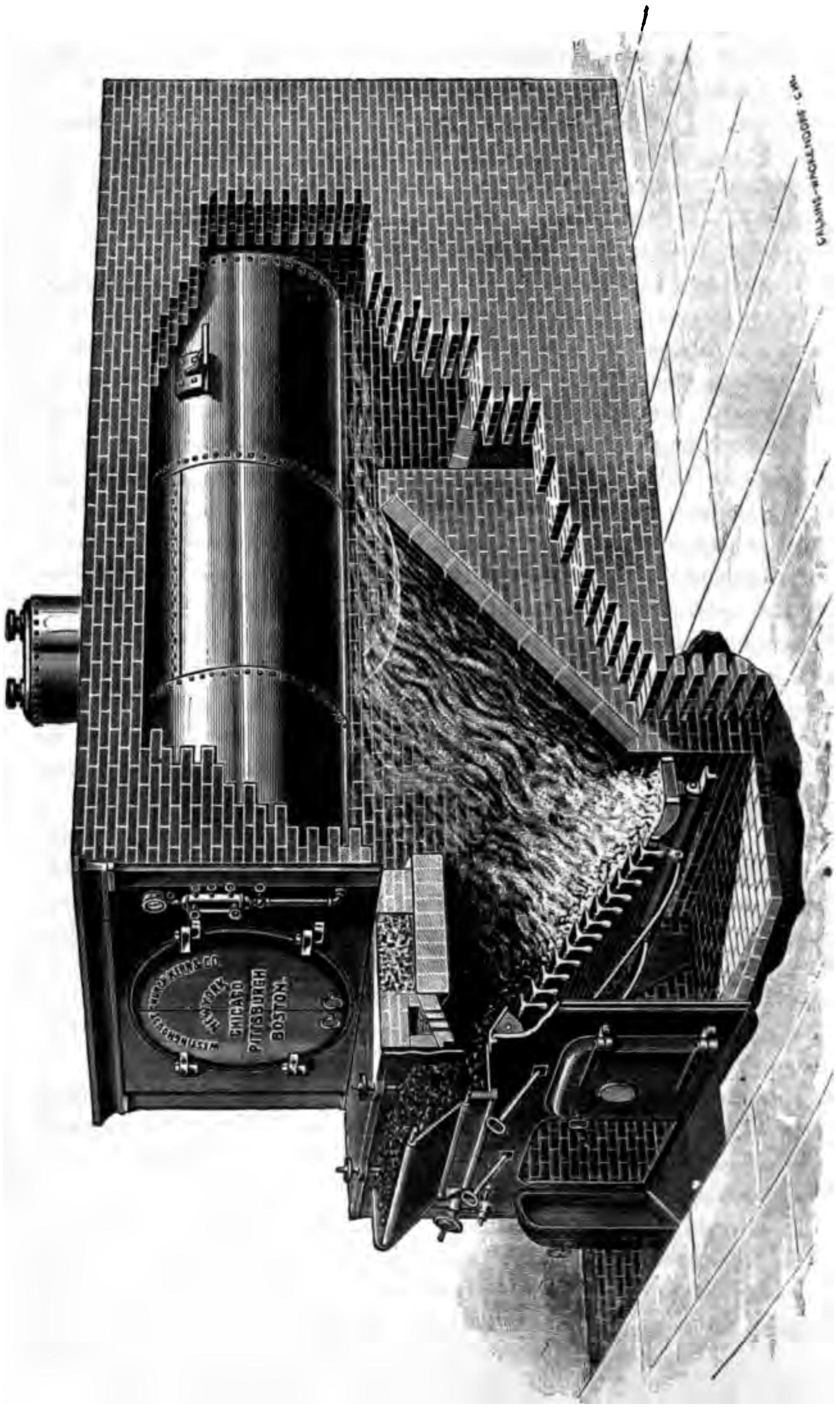
This variable, shutter-like movement furnishes a simple means of regulating the quantity of air passing through the grate,

The objection to the use of bituminous coal in cities, on account of smoke, is entirely removed by a good mechanical stoker, and the additional economy of smokeless combustion obtained. The results of a large number of tests at Newcastle-upon-Tyne, showed that, when dense smoke was produced, the evaporative efficiency was but 9.19 lbs. per pound of coal, as against 11.05 lbs., when smoke was entirely prevented, a difference of 17% in favor of smokelessness.

In addition to the saving in fuel, a properly constructed mechanical stoker reduces the number of men required in the fire-room, the amount of this reduction depending upon the number and arrangement of the boilers. When the size of a steam plant reaches the point where one man can no longer handle it, then the stoker begins to save labor; and ordinarily, one man with the stoker will easily do the work of two or more, hand-firing. The greatest labor-saving, however, appears when a boiler plant is large enough to warrant the use of coal and ash handling machinery, in connection with mechanical stokers. Probably the best and most complete illustration of what it is possible to accomplish in the reduction of labor by means of stokers and coal and ash machinery, is the well-known steam plant at Claus Spreckels' sugar refinery, Philadelphia, Pa. Here, four firemen on each turn take the entire care of a battery of Babcock & Wilcox boilers, equipped with Roney mechanical stokers, that are regularly developing an average capacity of 8000 H.P.;—and do it with less effort than 16 men would exert firing a similar battery by hand. One man is sufficient to remove the ash produced during the entire 24 hours. This plant is developing an average economy of over 11 lbs. of water from and at 212° Fahr. per pound of George's Creek semi-bituminous coal; which, considering the size of the plant, and the great irregularity in the steam consumption, incident to the business, must be regarded as good work.

In considering the subject of mechanical stoking, the question naturally arises: How should a stoker be constructed, to meet all the varied conditions, or in other words, what should be the characteristics of the ideal stoker? I would answer:

It should combine simplicity of construction and operation with efficiency, smokelessness, mechanical durability, and large capacity, when necessary. It should work automatically, supplying the coal to the furnace as fast as may be required, and


WESTINGHOUSE CHURCH KERR & CO.
NEW YORK
CHICAGO
PITTSBURGH
BOSTON.

maintaining a uniform fire on the grate. It should be so constructed as to permit free passage through the grates of a large amount of air when required, and this should be accomplished without the aid of a slice-bar. It should be so constructed as to make possible the easy application of a system of coal and ash handling machinery, for the further reduction of labor. It should discharge the ash and clinker quickly and easily, and without the admission of cold air to the furnace. It should be so constructed that the coal magazines and all the mechanism for feeding the coal shall be entirely outside the furnace, and where they will be unaffected by the intense heat. It should be capable of burning successfully all grades of bituminous, semi-bituminous, and lignite coals. The grate should be easily accessible from above and below, and so arranged that every part can be seen at all times, and the condition of the fire upon the grate ascertained without opening the firing doors. It should be so constructed that repairs can be reduced to a minimum, and worn-out parts be replaced without disturbing the remainder. It must be so simple as to be within comprehension of the most ordinary mind, and yet at the same time, in reality, compete with the work of a skilled hand-fireman, and do it with the accuracy of a machine, which will never "get tired" and never "strike."

It was with some such ideal stoker in mind, and with a conviction that this ideal had not yet been realized, that the author of this paper undertook some years ago to design a mechanical stoker adapted to the types of boilers and kinds of coal most generally used in this country. How well he succeeded, I will not say, but will leave it to the verdict of that large and intelligent body of engineers, who, by precept and practice, have done so much to encourage the development and use of mechanical stokers in this country.

A brief description of the construction and operation of this stoker may be of interest to those who have never had an opportunity to see one in operation.

The Roney Mechanical Stoker is a simple apparatus, which, when attached to steam boilers, receives the fuel in bulk, and thereafter, without further handling, feeds it continuously, and at any desired rate, to the furnace; burns the combustible portion, and deposits the ash and cinder in the ash pit, ready for removal.

Fig. 271.—Sectional Perspective.

The fuel to be burned is dumped into a hopper on the boiler front. In small plants it may be shoveled in by hand. In large plants it is usually handled, direct from the car to the hopper, by elevators and conveyors. Set in the lower part of the hopper is a "pusher" (see Figs. 271, 272, and 273), to which is attached, by a flexible connection, the "feed plate" forming the bottom of the hopper. The "pusher," by a vibratory mo-

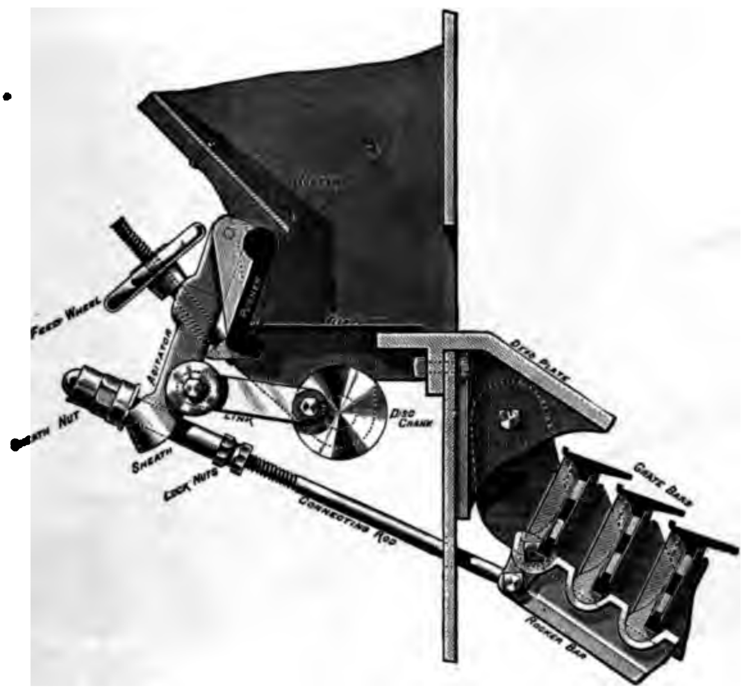


FIG. 272.

Inclined Position. Details of Feed and Grate Movement

tion, carrying with it the "feed plate," gradually forces the fuel over the "dead plate" and on the grate. The grate consists of horizontal, flat-surfaced bars, reaching from side to side of the furnace, carried on inclined side bearers, extending from the throat of the hopper to the rear and bottom of the ash pit. The grate-bars, therefore, in their normal condition form a series of steps, to the top step of which coal is fed from the "dead plate." These steps at the inclination given would, however, prevent the free descent of the coal. But each bar rests in a concave seat

in the bearer, and is capable of a rocking motion through an adjustable angle. All the grate-bars are coupled together by a "rocker bar," the notches of which engage with a lug on the lower rib of each grate-bar, pin connections being made with two of the grate-bars only, for the purpose of holding the

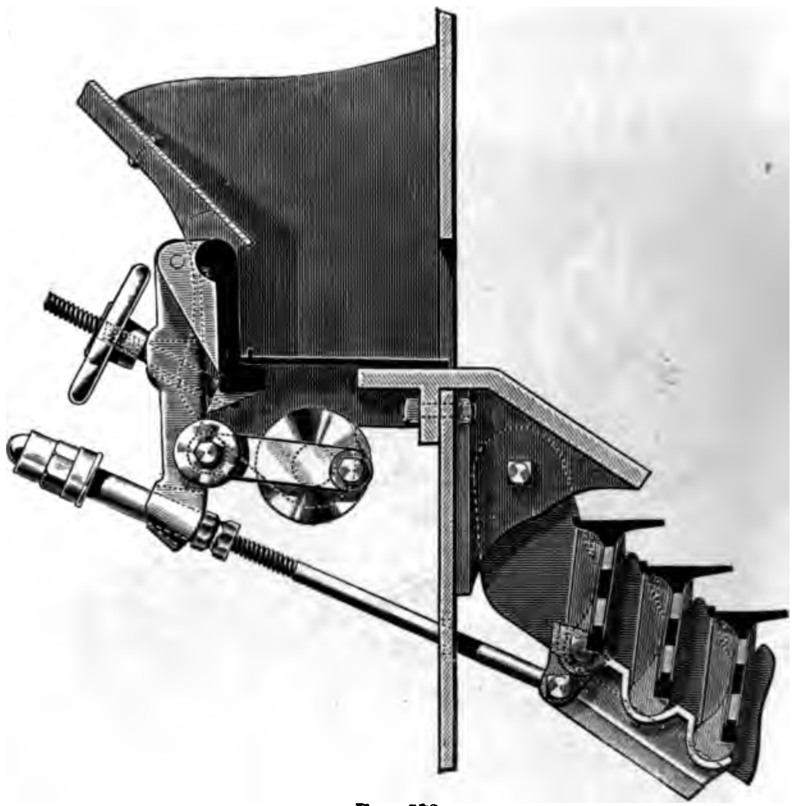


FIG. 273.

Stepped Position. Details of Feed and Grate Movement.

"rocker bar" in position. A variable back-and-forth motion being given to the "rocker bar," through a connecting rod, by a device to be hereafter described, the grate-bars necessarily rock in unison, now forming a series of steps (see Fig. 273), and now approximating to an inclined plane, with the grates partly overlapping like shingles on a roof. (See Fig. 272.)

This variable, shutter-like movement furnishes a simple means of regulating the quantity of air passing through the grate,

according to the amount of coal to be burned to supply the demand for steam. The depending webs of the grate-bars are perforated with longitudinal slots, so placed that the condition of the fire can be easily seen at all times, and free access had to all parts of the grate to assist, when necessary, the removal of clinker; thus avoiding all opening of doors into the furnace. These slots also serve an important purpose in furnishing an abundant supply of air for combustion. The sectional-perspec-

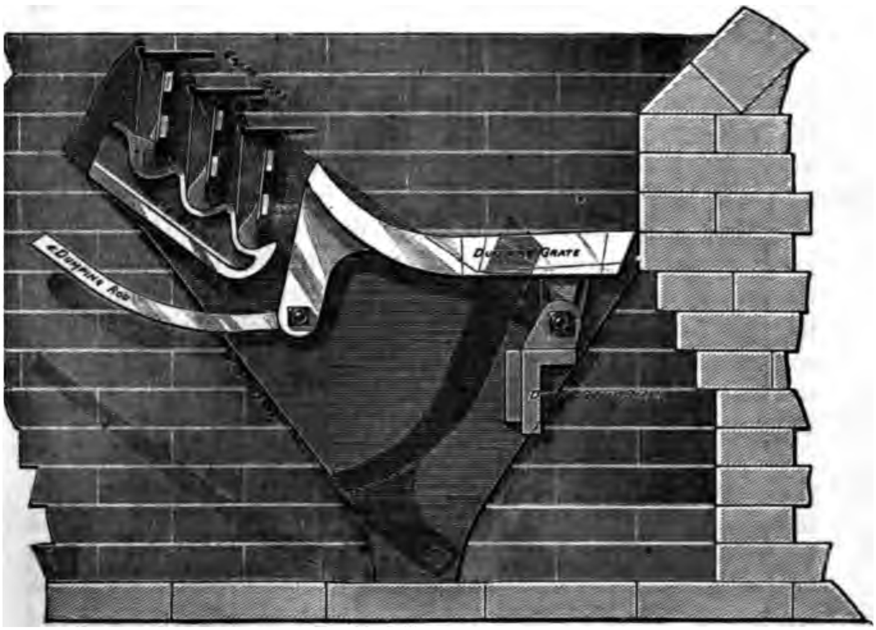


FIG. 274.

Detail of Dumping Grate. Shaded outline shows position of Grate when dropped.

tive cut, Fig. 271, illustrates clearly the form of the grate and the advantages of this construction.

Assuming the grate to be covered by a bed of coal, and fresh fuel being fed in at the top, it is obvious that when the grate-bars rock forward the fire will tend to work down in a body. But before the coal can move too far the bars rock back to the stepped position, checking the downward motion, breaking up the cake thoroughly over the whole surface, and admitting a free volume of air through the fire. The rocking motion is slow, being from 7 to 10 strokes per minute, according to the kind of coal. This alternate starting and checking motion being continuous, keeps the fire constantly stirred and broken up from underneath, and finally lands the cinder and ash on the "dumping grate" below. By releasing the "dumping rod," the "dumping grate" tilts forward (see Fig. 274), throwing the cinder into the ash pit, after which it is again closed ready for further operation. The "dumping grate" is made in two parts, so that each half can be dumped separately. The operation of the stoker, therefore, consists of a slow but continuous feed, a constant stirring of the fire, and an automatic rejection of the cinder, all performed without the opening of fire doors.

For the purpose of facilitating the smokeless combustion of the coal, provision is made for supplying heated air to the furnace. When the stoker is erected an air-space is left between the fire-brick lining and the red brick wall of the furnace. Air is admitted to this space through openings in the stoker front, and, after becoming highly heated, is discharged into a hollow brick chamber immediately over the throat of the hopper, whence it issues through small perforations to mingle with the gas from the fresh fuel. A coking arch of fire-brick is sprung across the front of the furnace, covering a part or all the grate, according to the kind of boiler to which it is applied. This arch forms a reverberatory furnace whose action is to thoroughly coke the fresh fuel and release its gases before ignition. In fact, that portion of the furnace under the arch may be likened to a gas producer, from the throat of which issues a huge volume of heated gas, already partially mingled with the heated air from the side flues, to be quickly burned in the large combustion chamber by contact with the incandescent body of coke on the lower portion of the grates.

Not the least important element in the smokeless combustion is the steady supply of the coal in small but continuous quantity on to the coldest portion of the grate, where, after coking, it is steadily carried down to the hottest portion of the fire at the bottom and finally consumed.

The actuating mechanism of the stoker is simple. All motion is taken from one driving shaft. In a single stoker this shaft is driven from a small engine attached to the stoker front and consuming a hardly measurable fraction of a horse-power. In large batteries of boilers, the driving shaft is extended across all the boiler fronts, delivering power to each stoker, and is driven by a small independent engine. The largest stoker can easily be turned over by the hand, indicating the nominal power consumed. The driving shaft carries a disc and wrist-pin or a crank, from which a link couples to the "agitator" (see Figs. 272 and 273). Through the eye of the "agitator" passes a stud screwed into the "pusher," on which stud is a "feed-wheel" by which the stroke of the "pusher," and consequently the amount of feed, is regulated. The "agitator" having a fixed stroke, it is apparent that if the "feed-wheel" is run down against it in the position shown in Figs. 272 and 273, the "pusher" will be given its full traverse and the greatest feed. If run back to clear the travel of the "agitator," the "pusher" will of course have no motion, and the feed will stop. Between these extremes any desired rate of feed can be given. In like manner, the rock of the grate-bars can be adjusted between any limiting angles, and over a range of motion from no movement to full throw, by means of the "sheath nut" and "lock nuts" on the "connecting rod," and the amount of stirring which the fire receives regulated according to the demand for steam. By these two simple adjustments, within the comprehension of the ordinary fireman, the whole action of the stoker is controlled, and the fires forced, checked, or banked at will. There are small doors in the front on each side of the hopper, giving access to the furnace above the grate when necessary.

The stoker is strong and well built, having in view the conditions of its operation and the usual nature of its attendance. There are few pin connections or finished parts. The strains are exceedingly light, and the motion is so slow as to be hardly perceptible to the casual observer. Any single grate-bar can be picked out and replaced even easier than in the ordinary flat grate; and in case it is necessary, the lower bar, where the greatest wear takes place, can be replaced without cooling down the furnace, thus avoiding loss of time and fuel.

The following table gives the results of six tests made for the purpose of ascertaining the comparative capacity and economy of horizontal return tubular boilers when fired by hand and by mechanical stoker. The conditions were as far as possible identical on all six tests, except that the first two extended over two weeks, while the remaining four were of one week duration each; and also that 4 boilers were used on the first test, 3 on the second, and 1 boiler on each of the four succeeding tests. In each test the boilers started cold Monday morning, and the fires were banked each night except Saturday. The record shows the total consumption of coal and water for the entire time in each case. Coal and water were weighed, and temperature of feed and steam pressure recorded each hour.

RECORD OF SIX TESTS TO DETERMINE THE COMPARATIVE ECONOMY OF THE RONEY MECHANICAL STOKER AND HAND FIRING ON H. R. T. BOILERS, 60 INS. x 20 FT., BURNING CUMBERLAND COAL WITH NATURAL DRAFT. RATING OF BOILER AT 12.5 SQ. FT.—105 H. P.

NOTE.—The same man fired on all six tests. First three tests, hand fired; last three tests, stoker.

Duration of Test. *Hours.*	Tempt. of Feed Water. *Degs. Fahr.*	Steam Pressure. *Lbs.*	Total Coal, plus Wood at 40%. *Lbs.*	Total Water Evaporated *Lbs.*	Evaporat'n per lb. dry Coal. Actual. *Lbs.*	Evaporat'n per lb. Dry Coal from and at 212°. *Lbs.*	H. P. Developed Above Rating of Boiler.
123.5	145.7	107.5	134459	1256249	9.34	10.36	—5.84%
132.0	143.2	104.6	.135338	1270758	9.39	10.44	13.52%
64.25	152.2	66.1	31224	310966	10.02	11.00	68.00%
65.5	145.4	63.1	28121	288781	10.31	11.89	54.65%
64.5	146.0	68.0	29794	303887	11.06	12.25	66.66%
65.5	145.2	65.2	29000	320034	11.35	12.54	84.26%

CCCCLVIII.*

COMPARISON OF THE ECONOMY OF COMPOUND AND SINGLE CYLINDER CORLISS CONDENSING ENGINES, EACH EXPANDING ABOUT SIXTEEN TIMES.

BY D. S. JACOBUS, HOBOKEN, N. J.
(Member of the Society.)

THE engines used in obtaining comparative results are located at Stations I. and II. of the Pawtucket Water Co. The results for the compound engine at Station I. have been abstracted from papers already presented to the Society,† so that the present paper is limited to tests made on the single cylinder engine at Station II., and to a comparison of the results with those already obtained on the compound engine.

The engines are very well adapted for obtaining comparative results, as they develop the same power and expand the steam about the same number of times.

The tests show that the compound engine is about 30% more economical than the single cylinder engine. The dimensions of the two engines are as follows:

	Single.	Compound.
Bore in Inches	20	15x30½
Stroke "	48	30
Diameter of Piston Rod in Inches	3	2¾

Both of the engines are steam-jacketed, practically on the barrels only, with steam at full boiler pressure.

A mean of two tests is taken for the compound engine, and a test made November 4, for the single cylinder engine. A comparison of the results obtained is as follows:

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Vol. XII. of the *Transactions*.

† *Transactions of American Society of Mechanical Engineers*, Vol. XI., pp. 328 and 1038.

	Single.	Compound.
Total Horse Power	141.5	145.9
Revolutions per minute	71.82	49.23
Ratios of Expansion with Initial Steam Pressure of 127 lbs.	15	15¼
Pressure of Steam in lbs. above Atmosphere	106.3	127.5
Number of Degrees that Steam is Superheated	5.5	3.0
Vacuum in Inches of Mercury	27.5	27.9
Steam per Hour per Horse Power	20.35	13.73

The steam pressure in the case of the compound engine is 21 lbs. higher than for the single engine. If the steam pressure be raised this amount in the case of the single engine, and the card be increased as represented by the shaded areas in Fig. 275, the consumption for the single cylinder engine would be 19.97

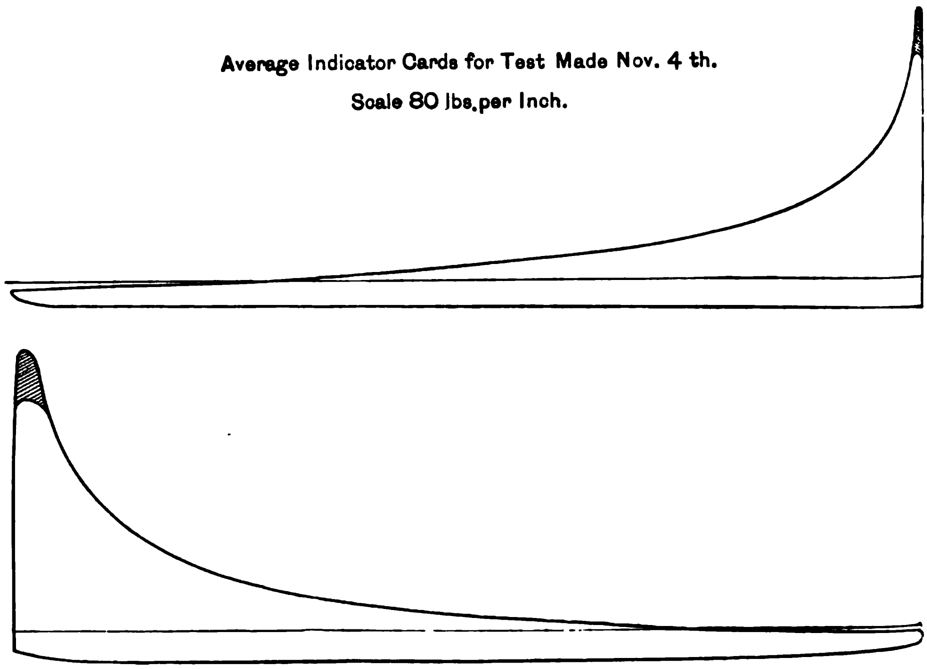


Fig. 275.

lbs. per hour per horse-power. This figure, which is used in obtaining the 30% difference in economy, is probably slightly too small, for on increasing the range of pressures in the cylinder we would increase the initial condensation, whereas this is not allowed for in the calculation. The correct figure must, therefore, be between 19.97 and 20.35 lbs. per hour per horse power, or the entire difference involved is only .38 lbs. The error in the correction applied is only a fractional part of this difference, and is therefore so small that it may be disregarded. The ratio of expansion given in the table for the cards taken from the single cylinder engine is measured from the point in the prolongation of the expansion line at 127 lbs. pressure, so as to be comparable with the figure given for the compound engine.

The detailed results of the tests are given in Table I. Tables

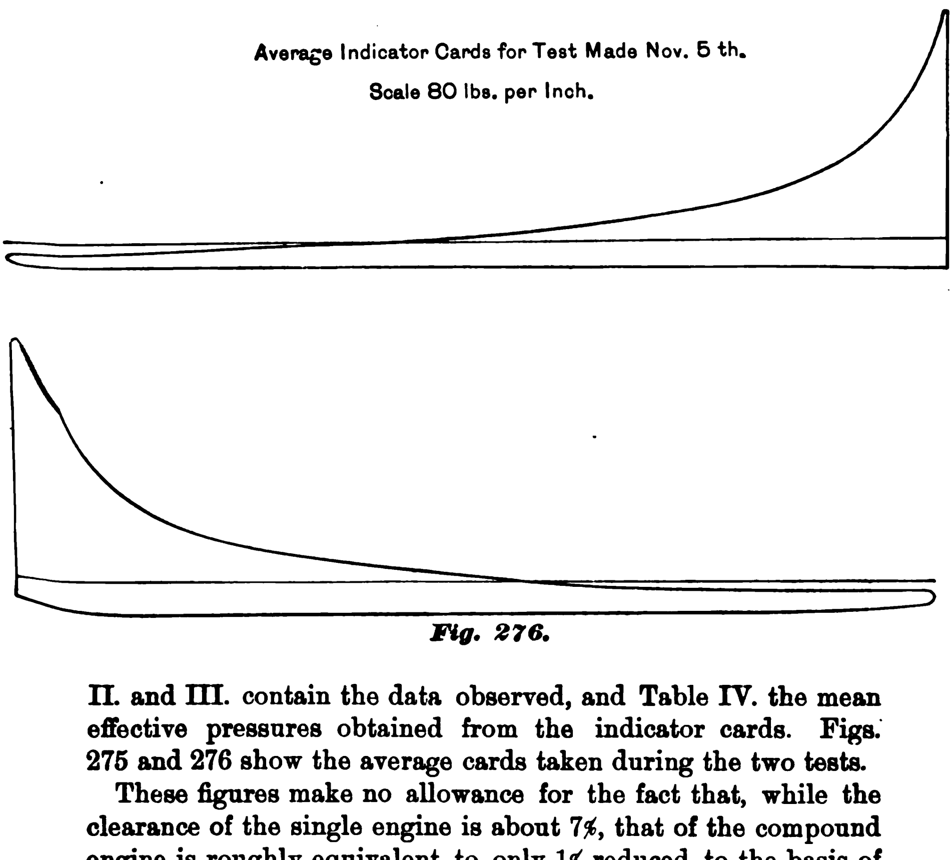


Fig. 276.

II. and III. contain the data observed, and Table IV. the mean effective pressures obtained from the indicator cards. Figs. 275 and 276 show the average cards taken during the two tests.

These figures make no allowance for the fact that, while the clearance of the single engine is about 7%, that of the compound engine is roughly equivalent to only 1% reduced to the basis of its total range of expansion taking place in the low cylinder.

The small range of superheating available with the boilers (about 30°) is liable to be almost entirely destroyed by small changes in the water-level, or in boiler pressure, or of chimney temperature, for instance, with a constant chimney temperature, an increase of boiler pressure from 90 to 105 lbs. may reduce the superheating from 20° to 5°, simply because the amount of steam generated is the same in both cases, but the higher temperature of the steam prevents as much heat being abstracted from the chimney gases by the outgoing steam.

TABLE I.

TEST OF STEAM CONSUMPTION OF THE SINGLE CYLINDER PAWTUCKET PUMPING ENGINE. BORE OF STEAM CYLINDER, 20"; STROKE, 48"; DIAMETER OF ROD, 3".

Date of test	Nov. 4.	Nov. 5.
Duration of test in hours	5½	6
AVERAGE PRESSURES.		
Boiler in lbs. per sq. inch above atmosphere	106.3	104.5
Water " " "	104.5	102.6
Vacuum in inches of Mercury	27.49	27.65
Suction in feet	13.77	13.62
Barometrical pressure in inches of Mercury	29.85	29.70
AVERAGE TEMPERATURE IN DEGS. FAHR.		
Steam in pipe 2 feet from steam chest	347.3	364.7
Of saturated steam at boiler pressure	341.8	340.6
Degrees of superheating	5 5	23.9
MEAN EFFECTIVE PRESSURE IN STEAM CYL.		
Head end cards	27.53	28.69
Crank " "	24.81	22.15
CALCULATED QUANTITIES.		
Revolutions per minute	71.818	71.895
Total horse-power	141.50	124.08
Steam used per hour	2880.2	2591.2
Steam per hour per horse-power	20.35	20.88

TABLE II.

DATA OBSERVED IN TESTS MADE NOVEMBER 4, 1890.

Time.	Height of Water Below Top of Gauge Glass in Inches.		Reading of Meter in Cubic Feet.			Pressure in Pounds per Square Inch.		Vacuum in Inches of Mercury.	Height that Suction Water is Lifted in Feet.		Temperature of Steam Before Entering Steam Chest.	Barometer in Inches of Mercury.	Revolution Counter Revolutions = Reading × 1¼.	
	Boiler No. 1.	Boiler No. 2.	Actual.	Corrected for Height of Water in Boilers.	Approximate Number of Cubic Feet Fed each Hour.	Steam.	Water.		Time.	Reading.			Time.	Reading.
A. M.												7.30 A. M.	A. M.	
9.15	0.6	2.5	372.3	372.3		105	105	27.5	9	13.77	350	29.83	9.20	291967
9.45	2.5	1.4	392.2		41.5	103	105	27.5			347		9.50	292403
10.15	1.8	1.7	413.9	413.8		106	104	27.7	10	13.75	349		10.20	294837
10.45	1.5	1.5	434.2		41.8	107	105	27.7			348	11 A. M.	10.50	296273
11.15	2.4	1.9	454.4	455.6		105	105	27.7	11	13.73	349	29.82	11.20	297710
11.45	2.0	0.9	478.5		41.5	108	105	27.7			348		11.50	299146
P. M.													P. M.	
12.15	1.1	1.2	497.9	497.1		108	105	27.4	12	13.73	346		12.20	300583
12.45	0.5	3.1	517.6		41.4	106	104	27.3			345		12.50	302019
1.15	3.0	0.9	537.7	538.5		107	103	27.3	1	13.83	347	4.45 P. M.	1.20	303456
1.45	1.7	1.2	553.7		41.5	107	105	27.3			347	29.89	1.50	304893
2.15	2.4	1.1	579.6	580.0		109	104	27.4	2	13.70	346		2.20	306330
2.45	0.6	2.5	602.0			105	104	27.4			346		2.50	307767
Sum or Difference.	19.6	19.9	229.7			1276	1254	329.9		82.60	4168	89.54		15800
Average and per hour.	1.63	1.66	41.76			106.3	104.5	27.49		13.77	347.3	29.85		4309.1

To obtain the weight of water used, in pounds, multiply the amount registered by the meter in cubic feet by 68.97.
The corrections in the meter readings for the height of water in the boilers are made by allowing one cubic foot per inch of water level.

TABLE III.

DATA OBSERVED IN TEST MADE NOVEMBER 5, 1890.

Time.	Height of Water in Gauge Glasses in Inches.		Reading of Meter in Cubic Feet.			Pressure in Pounds per Square Inch.		Vacuum in Inches of Mercury.	Height that Suction Water is Lifted in Feet.	Temperature of Steam Before Entering Steam Chest.	Barometer in Inches of Mercury.	Revolution Counter Revolutions = Reading × 1¼.	
	Boiler No. 1.	Boiler No. 2.	Actual.	Corrected for Height of Water in Boilers.	Approximate Number of Cubic Feet Fed each Hour.	Steam.	Water.					Time.	Reading.
A. M.												A. M.	
11.00	2.7	1.4	1856.1	1856.1		103	102	27.8	13.58	368	8.30 A. M. 29.67	11.05	325069
11.25	3.0	3.4	1874.8		39.4	103	102	27.8	13.58	365		11.35	326506
12.00	5.3	3.8	1900.5	1895.5		105	102	27.8	13.60	366	11 A. M. 29.78		
P. M.												P. M.	
												12.05	327944
12.30	6.3	5.5	1921.6		37.4	107	104	27.7	13.66	367		12.35	329381
1.00	4.4	5.8	1939.0	1932.9		101	102	27.6	13.70	364		1.05	330808
1.30	3.4	3.1	1954.4		36.3	106	102	27.6	13.68	364		1.35	332248
2.00	3.0	3.7	1971.8	1969.2		103	103	27.6	13.68	363		2.05	333683
2.30	5.5	3.4	1991.8		36.8	107	103	27.6	13.66	365	4 P. M. 29.65	2.35	335124
3.00	4.6	5.5	2012.0	2006.0		107	103	27.6	13.58	367		3.05	336571
3.30	6.0	4.4	2031.8		38.0	104	103	27.6	13.58	367		3.35	338008
4.00	5.4	4.9	2050.2	2044.0		105	103	27.6	13.58	364		4.05	339447
4.30	2.8	2.8	2061.9		37.5	108	103	27.6	13.58	364		4.35	340886
5.00	2.0	2.1	2081.5	2081.5		100	102	27.6	13.58				
Sum or Difference.	54.4	49.8	225.4			1359	1334	259.5	177.04	4379	89.10		15817
Average and per hour.	4.2	3.8	37.57			104.5	102.6	27.65	13.62	364.9	29.70		4313.7

TABLE IV.

MEAN EFFECTIVE PRESSURES FROM INDICATOR CARDS.

Test Made November 4, 1890.			Test made November 5, 1890.		
Time.	Mean Effective Pressure in Lbs. per Sq. Inch.		Time.	Mean Effective Pressure in Lbs. per Sq. Inch.	
A. M.	Head End.	Crank End.	A. M.	Head End.	Crank End.
9.25	25.6	26.9	11.10	25.1	22.9
9.50	27.3	24.4	11.45	23.6	22.5
10.10	27.9	24.4	11.45	23.8	22.4
10.43	27.1	23.8			
11.00	27.2	24.7	P. M.		
11.30	23.5	23.5			
11.42	27.1	24.8	12.33	23.8	22.8
11.47	28.5	25.3	12.43	23.4	23.1
			1.18	22.4	22.2
P. M.			1.40	23.6	21.3
			2.10	23.5	23.1
12.10	28.0	24.8	2.35	23.2	21.2
12.27	27.7	24.7	2.49	23.8	22.8
12.50	27.0	25.5	3.13	23.5	21.5
1.25	26.5	24.7	3.27	24.0	21.7
2.13	28.2	25.1	3.48	23.3	21.6
2.50	28.8	24.7	3.55	23.5	21.9
			4.18	23.8	22.5
			4.37	24.7	21.9
Sum........	385.4	347.3	Sum	379.0	354.4
Average	27.53	24.81	Average	23.69	22.15

ADDED IN PRESENTATION.

At the time of preparing this paper there had been no opportunity offered to measure correctly the clearance of the engine, and for this reason all deductions depending on the clearance were omitted from the test. This portion of the work is now complete. The clearance of the engine was determined with accuracy to be 7.65%. Making use of this figure, the ratio of expansion for the test made November 5 is the same as for the compound.

We deduce for this test:

Water per hour per horse-power, calculated from indicator cards.	Near cut-off....................	13.38 lbs.
	Near release....................	17.15 "

Water not accounted for by indicator	Near cut-off .... 35.9%.
	Near release .... 17.9%.

The similar quantities for the compound engine are :

Water per hour per horse-power, calculated from indicator cards.	Near cut-off. High press. cyl..... 9.86 lbs.
	Near release. Low press. cyl..... 9.76 "
Water not accounted for by indicator	Near cut-off. High press. cyl. ...28.2%.
	Near release. High press. cyl.....28.9%.

The indicator cards of the second test made on the same engine are compared with the combined diagram of the compound in Fig. 334.

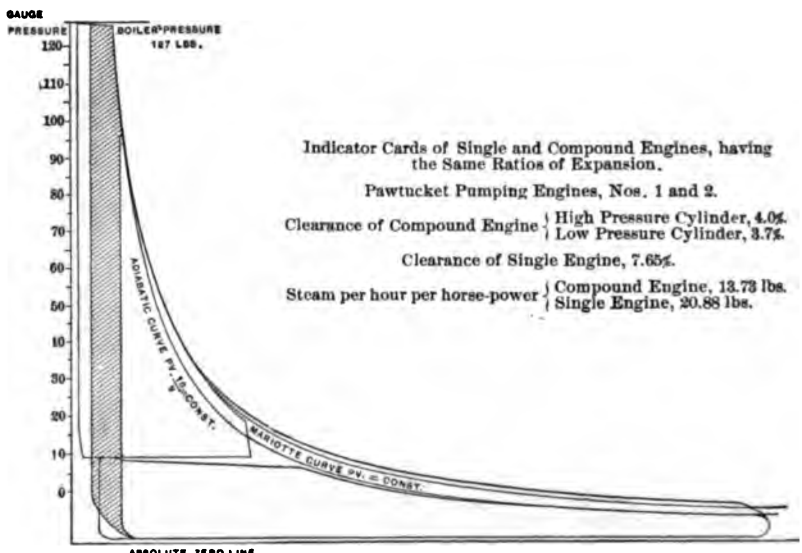


FIG. 334.

The loss due to clearance is seen to be much greater for the the single than for the compound.

If the clearance of the single engine was 3%, then we would gain the shaded area represented in the figure ; and, provided the effect of condensation remained the same, the steam per hour per horse-power would be reduced to 16.7 lbs. This illustrates one of the advantages of the compound engine over the single, for, although the clearance of the high-pressure cylinder of the compound is 4%, the corresponding loss of area, as represented on the diagram, is only about one-third that for a single cylinder engine having 3% clearance.

DISCUSSION.

Prof. R. H. Thurston.—This is the first comparison of engines of the two forms, doing substantially the same work, side by side, and under substantially the same external conditions, which I have met with, and I would like to thank Prof. Jacobus for what he has given, and express a desire for more. The engines both give exceptionally good results, and the comparison is a fair one, barring two points—the speed of the simple engine is 40% higher than that of the compound, and its steam pressure is 25 lbs. lower. Were the latter speeded up to equal velocity with the former it would presumably somewhat increase its efficiency —an efficiency already remarkably high. On the other hand, were the same steam pressure adopted, it would probably produce an opposite effect on their comparative efficiency. It is difficult to form an idea of the exact relative values of the two styles of engine, even with the results here given before us. If Prof. Jacobus can find an opportunity and time to trace the heat-flow in these engines from boiler to condenser, as is done by Prof. Peabody and Mr. Bird in their papers, and give us the measures of initial dryness of steam in the steam-chest, the initial condensation, the gain and loss of heat by the cylinder surfaces, and the effect of the jackets, he will add very greatly to our obligations. It is easily seen from the cards shown that there is, even now, in the simple engine, some considerable initial condensation, which reveals itself in the later re-evaporation and change of the expansion line, which rises well above the hyperbolic curve. The simple engine suffers in this comparison by its high percentage of clearance, and both would, I imagine, do still better were it practicable to give them large compression.

Prof. Cecil H. Peabody.—I have been much interested in this paper. For some time I have been studying engine tests, and especially comparative engine tests, and it has been my experience that it requires the greatest care to compare engine tests in such a way as to arrive at conclusions which are of any value. Now, while both these engines have the same amount of expansion, taking the ordinary way of comparing expansion in compound engines, yet it appears to me that these experiments ought not to be compared, and that a conclusion from them ought not to be drawn at all. It is my opinion (and I think it can scarcely be questioned) that, if you want to compare two engines to find out which is the more economical, you ought to put each engine in the best possible condition for itself; and I think it is perfectly clear that the single engine is not in the proper condition at all; that it has about 15 expansions, when it really ought to have 8 or, at the most, 10. Now, a certain test made upon an engine of about the same size, I think in 1881, showed a consumption of just about the same amount of steam without any jacket. There is a great deal of difference of opinion as to how much effect a jacket will have, but if it should be granted that a 10% gain of economy may be obtained from a jacketed engine, then this engine ought to be able to do as well as 18 lbs. of steam per horse-power per hour. As for the addition made by the author in drawing the diagram, that I am, in one sense, rather glad to see. I am glad to see some one who draws a diagram of that sort, and who will tell us why he draws it, and what conclusion can be drawn from it, because I have very much questioned what is the value of a diagram after it is drawn. And it appears to me also a question whether it is advisable to make any sort of a reduction for the consumption of steam engines and then compare the engines. It appears to me, in the first place, that the comparison of the engine as it stands is unfair to the simple engine; in the second place, the comparison which is made by aid of the diagram, bringing down the consumption to 16.7 lbs., is better than I have ever seen from a recognized test, and I should want to see it verified by something different from the method which has been used here.

Mr. F. Meriam Wheeler.—From a commercial point of view I consider that the economy shown by this engine is quite remarkable. I doubt very much whether any one present could give many cases where simple engines have done as well. There are, of course, a large number of compound engines which have made a good showing. I only mention this because the gentleman who preceded me rather gave the impression that it was a common thing to find simple engines doing as well as this particular engine. I know in New England, where so many Corliss (condensing) engines are used, that the average economy is much above 20 lbs.; the range is generally from 22 to 24 lbs., while it is a well-known fact that many compound engines built in the last few years have done as well as 15 lbs. and better, running six days in the week.

*Prof. D. S. Jacobus.**—The main object of this paper is to compare a single with a compound engine in the special case in which each expands 16 times, and to show the advantage of the compound engine.

This is the problem involved, as stated by the heading of the paper.

The remarks made by Prof. Peabody, in regard to comparing the economy of the two engines, therefore, do not apply to the case. Of course it may be that the single engine could be run at a more advantageous cut-off, and this is illustrated in the two tests given in the paper, for the test in which the expansion is the greatest does not give the lowest rate of steam consumption, and in all probability the rate may be slightly lowered for a still longer cut-off. It is not the object of this paper to compare the best economy of a single with that of a compound engine. It is simply to show the advantage of a compound over a single engine, in which each has a high rate of expansion. This is the only instance, so far as we have been able to ascertain, where we could obtain this abnormal condition of running, as it may be called, in the single engine, and where the same indicated work is produced with the same number of expansions as a compound engine.

In regard to adding the shaded area indicated on the diagram: This method for allowing for differences in the clearance is, of course, an approximate one. Perhaps this fact is not brought out as well as it should be in the paper, but a close examination will show that it includes a discussion bearing on this point, where an area is added to the indicator card to correct for differences in the steam pressure. Reducing the clearance may alter the effect of the cylinder condensation, but how much this will amount to we do not know. The method of adding the area is employed simply to give a possible explanation pointing out where we may look to find the cause of the great difference in the rate of steam consumption in the compound and the single engine.

16.7 lbs. is not put down as a conclusive figure, and should not be used in forming a practical comparison; it is derived simply to show that the effect of the clearance spaces produce a large portion of the gain of the compound over the single engine.

In regard to tracing the heat through the engine, it appears to

---

* Author's Closure.

Passing by such striking properties of manganese-steel as its freedom from blow-holes; the great difficulty with which it welds; the increase in its toughness caused by quenching from a yellow heat; its electric resistance—enormous, and very constant with changing temperature; its low thermal conductivity: passing these by, one group of qualities stands out as that which must create and yet limit its value in the arts. I refer to its remarkable combination of great hardness, which cannot be materially lessened by annealing, and great tensile strength, with astonishing toughness and ductility; this group, I say, must at once create and limit its usefulness. The very fact that manganese-steel cannot be softened, that it ever remains so hard that it can be machined only with great difficulty, at once sets up a great barrier to its usefulness; yet a barrier which, as we shall soon see, is not so insurmountable as it might at first appear.

On the other hand, if, as I believe, this cheap metal has a much higher combination of the important properties of hardness and ductility than exists in any other known substance, metallic or non-metallic, we can hardly hesitate to prophesy for it an important and useful future, for we can hardly doubt that for some important purposes this combination is extremely desirable. This combination is illustrated by strips which have been bent double cold, and yet can barely be filed.

Resistance to Abrasion.—The results given in Table I. were obtained by Mr. T. T. Morrell, chief chemist of the Cambria Iron Works, by pressing weighed pieces of manganese-steel and of other steel, with a constant pressure, against a rapidly revolving hardened steel shaft, and ascertaining the loss of weight which each piece underwent per thousand revolutions of this shaft. Here manganese-steel indeed wears faster than tempered tool steel; but this is not a material which enters into competition with manganese-steel, as it is not ductile. The difference between the wear of manganese-steel and of that of the other steels tested is very great.

TABLE I.

ABRASION BY A REVOLVING HARDENED STEEL SHAFT.

Loss of weight of manganese-steel	1.0
" blue-tempered hard tool steel	0.4
" annealed hard tool steel	7.5
" hardened Otis boiler-plate steel	7.0
" annealed " " "	14.0

CCCCLIX.*

MANGANESE-STEEL.

BY H. M. HOWE, BOSTON, MASS.
(Member of the Society.)

MANGANESE-STEEL is an alloy of iron and manganese, incidentally, and probably unavoidably, containing a considerable proportion of carbon.

The effect of small proportions of manganese on the hardness, strength, and ductility of iron is probably slight. The point at which manganese begins to have a predominant effect is not known: it may be somewhere about 2.5%. As the proportion of manganese rises above 2.5% the strength and ductility diminish, while the hardness increases. This effect reaches a maximum with somewhere about 6% of manganese. When the proportion of this element rises beyond 6% the strength and ductility both increase, while the hardness diminishes slightly, the maximum of both strength and ductility being reached with about 14% of manganese. With this proportion the metal is still so hard that it is very difficult to cut it with steel tools. As the proportion of manganese rises above 15% the ductility falls off abruptly, the strength remaining nearly constant till the manganese passes 18%, when it in turn diminishes suddenly.

Steel containing from 4% to 6.5% of manganese, even if it have but 0.37% of carbon, is reported to be so extremely brittle that it can be powdered under a hand-hammer when cold; yet it is ductile when hot.

We have here another of the endless cases in which the properties of the alloy differ greatly from the mean of those of its components. This usually astonishes the novice, and even experienced metallurgists are often betrayed by it into expressions of surprise. It is, in fact, about as surprising as the parallel fact that the properties of water are not the mean of those of oxygen and hydrogen, and that those of peroxide of hydrogen are not directly deducible from those of water and oxygen.

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Vol. XII. of the *Transactions*

ultimate tensile strength, since each is measured by the force needed to break continuity by tearing particle from particle. This force, at the plane where any two particles are torn apart, is in either case a pull; *i.e.*, tensile.

The resistance to abrasion, moreover, should be related to the coefficient of friction of the metal; or, regarding the surface of the metal as like a brush or the surface of a file, the abrasion and the coefficient of friction should depend on the depth to which the teeth of the abrading file fit between those which it is to abrade, and the mode of distribution of the teeth of the two files which are abrading each other.

Be this as it may, the hardness of manganese-steel seems to be of an anomalous kind. The alloy is hard, but under some conditions not rigid. It is very hard in its resistance to abrasion; it is not always hard in its resistance to impact. I do not recall a case in which it has not shown itself very hard when exposed merely to abrasion, yet there are cases in which it has not shown itself hard under repeated impact.

I do not refer to the mere fact that it can be indented by a sharp blow: this power of enduring distortion is almost a necessary consequence of its great ductility. I refer to its behavior under conditions like those of a hammer-head or stamp-shoe. As a material for stamp-shoes, for horseshoes, for the knuckles of an automatic car-coupler, manganese-steel has not met our expectations. When we remember how many of us have been studying carbon-steel these long years, and how ignorant we must yet confess ourselves concerning even some of its prominent characteristics, there is little wonder if, in speaking of so little-studied a material as manganese-steel, I must admit that its rigidity in certain cases, its pliancy in others, is puzzling.

Armor-plates.—The armor-plates which I will soon describe give an instance of apparently great rigidity under shock; but it is probable that the reason why the manganese-steel armor-plates did not crack while the steel one did, is that the manganese-steel can undergo such great distortion without failure of continuity. For, though the plates as a whole are but little distorted, the metal immediately about the holes must have been distorted greatly, yet but trifling cracks appear.

Car-axles.—Another instance of rigidity under shock is that of a manganese-steel car-axle tested by Hadfield, in comparison with a special carbon-steel axle. Each was struck repeatedly

The advantage of manganese-steel is much less striking, as explained beyond, when it is exposed to abrasion by an intensely hard substance, like emery. This is shown in Table II.

TABLE II.

ABRASION BY AN EMERY-WHEEL.

Loss of weight of hard manganese-steel wheels	1.00
softer " "	1.19
hardest carbon-steel wheels	1.28
soft " "	2.85

Hardness and Rigidity.—The hardness of manganese-steel seems at first to be of a peculiar nature. We class together under the single term "hardness" what seem to be several tolerably distinct properties. These, though they habitually coexist, do not necessarily. Resistance to abrasion, to indentation by a sharp object, and to compression, habitually go hand in hand; yet, on reflection, we see that there is no *a priori* reason why they must coexist. I think that the tensile elastic limit should be closely related to the cohesion of the substance along the lines of least cohesion, or to the minimum cohesion. The resistance to abrasion, should the lines of low resistance be far apart, may bear little relation to this minimum cohesion, but be dependent on the cohesion between the particles immediately exposed to abrasion and those next them. The tensile elastic limit and the ultimate tensile strength of an emery-wheel are very low; its resistance to abrasion, enormous. That is because its minimum cohesion, which occurs between the grains of emery and their cementing matter, is very low, while the cohesion between the molecules of any one individual grain of emery is enormous. Among absolutely homogeneous substances, free from initial stress, the resistance to compression, to abrasion, and to tensile distortion should be proportional. It is, I take it, because many substances are heterogeneous, with lines of low resistance, like the joints in brickwork, the cleavage in crystals, the grain in wood, that these properties often bear so little relation to each other.

Again, while the resistance to compression in a material which yields by bulging should be related to the tensile elastic limit —since such substances simply take set under compression, the particles changing their relative positions without solution of continuity—the resistance to abrasion should be related to the ultimate tensile strength, since each is measured by the force needed to break continuity by tearing particle from particle. This force, at the plane where any two particles are torn apart, is in either case a pull; *i.e.*, tensile.

The resistance to abrasion, moreover, should be related to the coefficient of friction of the metal; or, regarding the surface of the metal as like a brush or the surface of a file, the abrasion and the coefficient of friction should depend on the depth to which the teeth of the abrading file fit between those which it is to abrade, and the mode of distribution of the teeth of the two files which are abrading each other.

Be this as it may, the hardness of manganese-steel seems to be of an anomalous kind. The alloy is hard, but under some conditions not rigid. It is very hard in its resistance to abrasion; it is not always hard in its resistance to impact. I do not recall a case in which it has not shown itself very hard when exposed merely to abrasion, yet there are cases in which it has not shown itself hard under repeated impact.

I do not refer to the mere fact that it can be indented by a sharp blow: this power of enduring distortion is almost a necessary consequence of its great ductility. I refer to its behavior under conditions like those of a hammer-head or stamp-shoe. As a material for stamp-shoes, for horseshoes, for the knuckles of an automatic car-coupler, manganese-steel has not met our expectations. When we remember how many of us have been studying carbon-steel these long years, and how ignorant we must yet confess ourselves concerning even some of its prominent characteristics, there is little wonder if, in speaking of so little-studied a material as manganese-steel, I must admit that its rigidity in certain cases, its pliancy in others, is puzzling.

Armor-plates.—The armor-plates which I will soon describe give an instance of apparently great rigidity under shock; but it is probable that the reason why the manganese-steel armor-plates did not crack while the steel one did, is that the manganese-steel can undergo such great distortion without failure of continuity. For, though the plates as a whole are but little distorted, the metal immediately about the holes must have been distorted greatly, yet but trifling cracks appear.

Car-axles.—Another instance of rigidity under shock is that of a manganese-steel car-axle tested by Hadfield, in comparison with a special carbon-steel axle. Each was struck repeatedly by a 20.75-cwt. ram, while resting on supports three feet apart, and each was reversed after every blow, in the usual way. The manganese-steel axle received, up to the time of breaking, blows representing 43% more energy than those received by the carbon-steel axle; yet the total deflection of the manganese-steel axle was very much less than that of its competitor.

TABLE III.

TESTS OF MANGANESE- AND OF CARBON-STEEL AXLES.*—HADFIELD.

Effect produced by the Number of Blows as under.	Energy developed in Foot-tons.	Sum of Permanent Deflections or Bends in Inches produced on—	
		No. 1. Special Steel Axle.	No. 2. Manganese-steel Axle.
At the fifth	79.883	24.953	8.501
" tenth	208.581	66.188	19.403
" fifteenth	348.591	105.248 And broken.	30.212
" twentieth	497.988	....	39.491 And broken.

Car-wheels.—A case of good behavior under shock is furnished by a 33-inch manganese-steel car-wheel, weighing about 612 lbs. It was dropped edgewise on a one-ton steel block, bedded in the ground, repeatedly, from gradually increasing heights, with the results given in Table IV. For comparison, an American chilled cast-iron wheel, made by one of our best makers, and also some excellent carbon-steel wheels, were tested in the same way.

The total energy represented by the work done on the manganese-steel wheel was 100 foot-tons, or 18 times as much as in the case of the cast-iron, and twice as much as in the case of the carbon-steel; and this, although the average fall of the manganese-steel wheels was four times that of the chilled cast-iron and nearly double that of the carbon-steel.

* Excerpt *Proc. Inst. Civ. Engineers*, XCIII., Part III., 1888, p. 29.

TABLE IV.

DROP TESTS ON MANGANESE-STEEL AND OTHER CAR-WHEELS.

	Manganese-steel Wheel.		Soft Carbon-steel Wheel.		Chilled Cast-iron Wheel.
No. of Drops. N	Height of Fall. H	No. of Drops. N	Height of Fall. H	No. of Drops. N	Height of Fall. H
	Feet.		*Feet.*		*Feet.*
2	16	1	10	1	4
9	28	1	11	1	6
	Rim slightly cracked.	1	12	1	10
		1	13		Broke into six pieces.
3	28	1	14		
	Tread broken through.		Cracked slightly.		
		7	15		
		1	18		
			Cracked about ¼ across the face of the tread.		
Total height dropped....	368	....	183	....	20
Average height of drop ..	26.3	....	14.1	....	6.7
ΣNH^2 absolute.......	9,920	....	2,629	....	152
ΣNH^2 relative.......	3.8	....	1.	....	
ΣNH^2 relative........	65	....	17.3	....	1

Axes.—Again, manganese-steel axes have chopped cold iron bars through. Here we have great rigidity under shock.

Yet in other cases manganese-steel has not shown the rigidity which those I have just described would lead us to expect. However, I know of none in which manganese-steel has proved brittle under shock, except in early experimental work, in which the steel was unwisely given too much carbon, and further excepting untoughened manganese-steel castings.

A partial explanation of the cases I have referred to, and of others, and an inference from them, is that manganese-steel is more rigid than the soft steels with which it competes in ductility, but less rigid than the hard steels which it excels in hardness.

To sum this up, in resistance to abrasion alone, manganese-steel excels the hard carbon-steels (when unhardened) and, *a fortiori*, the soft steels. Where both abrasion and repeated shocks are to be resisted, manganese-steel is certainly less liable to break than the hard carbon-steels; but whether, under new conditions, combining shock and abrasion, it will prove as rigid as the carbon-steels with which it will then have to compete, direct experiment alone can tell.

Hardness Evaded and Surmounted.—Having called your attention to the great obstacle to the use of manganese-steel, its persistent hardness, let me now point out how this obstacle may be evaded and even surmounted.

Special Forging Appliances. — Fortunately, manganese-steel forges readily at a yellow heat, though at a bright white heat it crumbles under the hammer. But it offers greater resistance to deformation—*i.e.*, it is harder when hot—than carbon-steel. This increases the cost of forging to a certain extent, and, if very great reduction of cross-section is to be effected by forging, the increased cost may be very serious; hence it is desirable that the ingot or other casting which is to be forged should have, when cast, as nearly as possible the dimensions of the finished piece, so that but little forging may be needed.

Iron, carbon-steel, and the other metals and alloys in general use, are readily tooled when cold. Hence we have, in general, tolerated forging appliances which leave the piece in a shape in which a considerable amount of tooling is needed. But, even for these metals, recent years have developed ingenious methods of forging even complicated pieces so closely to the needed shape as to avoid subsequent tooling, or greatly to reduce it. We have not only the great advances in drop-forging, but several different kinds of rolling machines, of the general class to which the Simonds rolling machine belongs.

Bridge-pins.—For example, it appeared that manganese-steel should be a valuable material for bridge-pins, since these need not only great shearing strength but great hardness. If they wear away, the play which arises is extremely injurious to the bridge. We were at once confronted with the difficulty of providing threads on the ends of the pins to hold the lock-nuts. The difficulty had hardly arisen when it was overcome by the discovery that Messrs. Wyman & Gordon, of Worcester, had a machine which forges threads on round bars of any size, and apparently accurately enough for this purpose.

Cold-shaping.—Turning now from this way of evading the hardness of manganese-steel by special appliances for working it while hot, we come to appliances for shaping it while cold, despite its hardness. These are, first, emery-wheels of various kinds for cutting it; second, hardened carbon-steel rollers, dies, etc., which work it cold.

Emery is so much harder than manganese-steel that it cuts it with great ease. Indeed, as an observer from the summit of a lofty mountain hardly notes the little hills at its base, their elevation being so trifling compared with his, so the hardness of emery exceeds that of manganese-steel so greatly that, when cut by emery, the resistance of this metal to abrasion excels that of carbon-steel much less than when the abrasion is effected by a revolving steel shaft. This is illustrated by Table II.

I may almost say that what the common hardened steel tool is to most metals the emery-wheel will be to manganese-steel. Just as we turn other metals in the lathe by holding a hardened steel tool against them while they revolve, so we may turn objects of manganese-steel by mounting them in like manner, and simply substituting for the hardened steel tool an emery-wheel revolving swiftly. So, too, may the stationary tool of the planer, and in some cases the reciprocating tool of the slotting machine, be replaced by the revolving emery-wheel, the piece which is to be planed moving past the emery-wheel, that which is to be slotted remaining stationary while the emery-wheel, driven by belt or gearing, plays up and down.

Hardened Steel.—Turning now to means of working rather than of abrading it when cold, a thin wire may be made to show at once that it is possible to work the metal cold to a very considerable extent. Wire has been made by drawing it through dies when cold in the common way. It is, however, hardly likely that manganese-steel wire will be widely used, the material offers so much resistance in drawing, is so hard, and requires such frequent annealing. After every two draughts it must be highly heated and quenched in cold water.

The great amount of cold-working which wire-drawing requires may be so costly as to be prohibitory, yet a smaller amount may cost but little. If manganese-steel could not be worked cold at all, it might be a serious thing. Even the most accurate methods of hot-forging bring the metal only approximately to the size desired, for the following reason—the temperature at which the metal is forged while hot must vary considerably. As that temperature varies, so does the number of degrees through which the metal must pass in cooling thence to the common temperature; and as the contraction of the metal in cooling is proportional to the range of temperature through which it cools, so must the amount of contraction which the metal undergoes in cooling vary. So that, no matter how accurately we bring the metal to a given desired shape and size when hot, as we cannot regulate closely the temperature at which we give it that size and shape, so will the size and shape which the metal has after cooling vary. For many purposes this variation may not be serious. But for many others it is extremely serious, not to say prohibitory. Yet so slight is it that but a very little cold-working is needed to compensate for it.

Now it so happens that Mr. Samuel Johnston, of 140 Nassau Street, New York City, has devised and constructed machines capable of rolling manganese-steel and other metallic substances while cold, and with great ease. I have here a strip of manganese-steel which was originally throughout its length of the cross-section which its larger end now has. It has been worked down to its present shape while cold, and without annealing. I am informed that this machine can produce pieces of exceedingly irregular shape. With the slight familiarity which I have with it, I find it difficult to set a limit to the shapes which it can produce. As we can turn wooden gun-stocks and axe-handles in a lathe, so can this machine turn out even most irregular pieces.

It is still too early to assign to these two methods of shaping manganese-steel when cold their exact fields. In many cases it is probable that either can be used, as in turning bridge-pins and car-wheels, in trimming armor-plates, and in shaping plough-points. In other cases, such as that of car-axles, where an irregular cross-section must be given with extreme accuracy, the Johnston machine seems more applicable than the emery-wheel.

Established Uses.—When we come to the actually well-established uses of manganese-steel, I must act the apologist. They are few. In addition to the inertia of custom, which properly leads each to cling to the material which does well enough rather than brave the ills he knows not of, we have the really serious obstacles which the very nature—indeed, the very advantages—of this alloy oppose to its use, and certain personal considerations on which I may touch but lightly.

The death of the senior Hadfield shortly after the invention of manganese-steel threw on his son the cares of a very large and growing business, in addition to those of the development of manganese-steel and of many other extremely promising alloys. There has been lack of laborers rather than lack of promise of harvest.

The most important single use for manganese-steel is for the pins which hold the buckets of elevated dredgers. Here abrasion chiefly is to be resisted; here manganese-steel has

FIG. 277. FIG. 278.

given remarkable results ; and here its use has, I believe, ceased to be tentative. Figs. 277 and 278 show how much better manganese-steel resists abrasion than carbon-steel. The manganese-steel pin, Fig. 278, has endured more than three times as much use as the forged carbon-steel pin, Fig. 277, and yet has been worn away very much less than its competitor. Hard as this 6-inch pin is, it can be bent double cold without cracking.

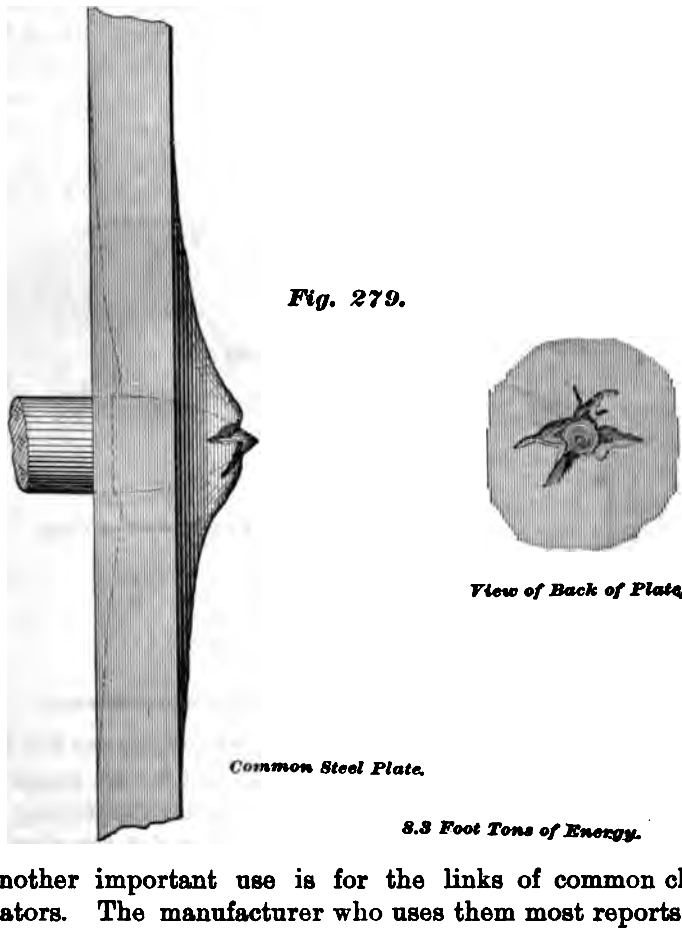


Fig. 279.

Another important use is for the links of common chain-elevators. The manufacturer who uses them most reports that they last more than twice as long as carbon-steel links.

The cyclone pulverizer consists of two propeller-blades, set end to end, in a closed chamber, and revolving rapidly in opposite directions. The ore or other substance to be powdered is dropped between these blades. They are thus subjected at once to severe abrasion and to heavy blows; hence they must be tough and hard. While chilled cast-iron is found to abrade slightly less under these conditions, yet it often breaks, causing serious accidents. Manganese-steel has been found perfectly safe, and has been regularly adopted.

Besides many smaller uses, a great many points and crossings for both steam and horse railroads have been made; but it is not yet known how the wear of manganese compares with that of carbon-steel under these conditions.

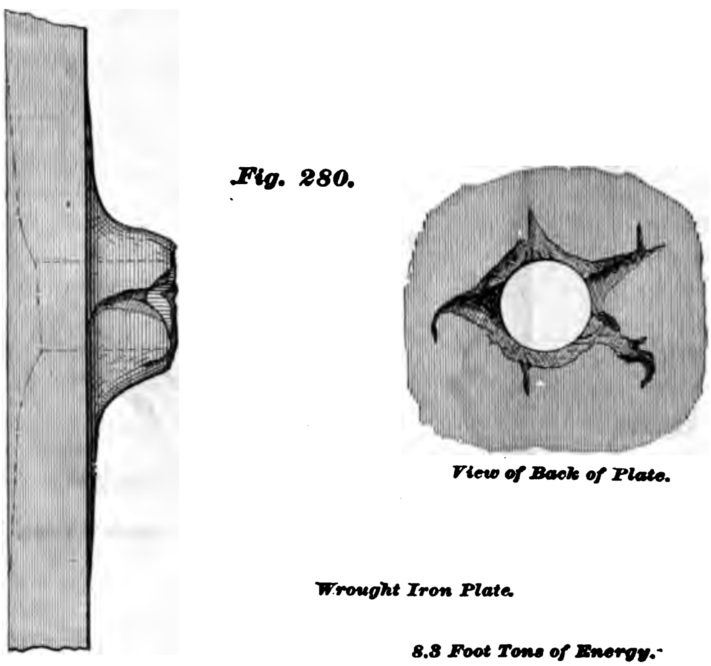


Fig. 280.

View of Back of Plate.

Wrought Iron Plate.

8.3 Foot Tons of Energy.

Car-wheels probably offer the most promising use for manganese-steel. A car-wheel must first of all be so tough and trustworthy that it will not break even under the trying conditions of use; and beyond this, it must resist abrasion; in short, it must have the combination of toughness and hardness which manganese-steel offers. Several serious obstacles have retarded the introduction of manganese-steel for this purpose, but we see our way pretty clearly to overcoming these completely. And I hope strongly to see manganese-steel become *the* material for passenger-car wheels within a few years. I am less hopeful as regards freight-car wheels, because the results are less serious when a cast-iron wheel breaks under a freight train than when it derails a passenger train.

The chief difficulty in the way of introducing manganese-steel car-wheels has been the extreme hardness of the metal, which made it difficult to turn the thread and to bore the hub true. This difficulty is now well in hand. The tread may be turned by means of an emery-wheel, the car-wheel itself being mounted in a lathe and revolving slowly so as to bring each point in its tread in succession against the emery-wheel.

The hub might be bored with an emery-cone such as is used successfully for dressing out the conical steel dies through which steel bars are drawn cold. But an objection to this is that the purchaser of a car-wheel may wish to bore out his wheel to fit a particular axle after it has left the maker's hands. Hence it will be better to weld or cast into the manganese-steel wheel a centre of iron or soft steel, which can be bored readily.

Manganese-steel wheels under a one-horse tram-car in Chester, England, were returned after running 80,000 miles. The service under horse-cars is very trying, as the brakes are so often applied and as so much sand and other gritty matter lie on the rails. The mileage on one of the most important street railroads in this country follows :

Maker.	Number of Wheels put in.	Number worn out.	Average Mileage.
X....................	515	293	7,748
Y....................	78	12	16,873
Z....................	154	14	4,587

Of these three classes of wheels two were of cast-iron and one of carbon-steel.

I am informed that some manganese-steel wheels have run over 300,000 miles each without turning, on a New England railroad.

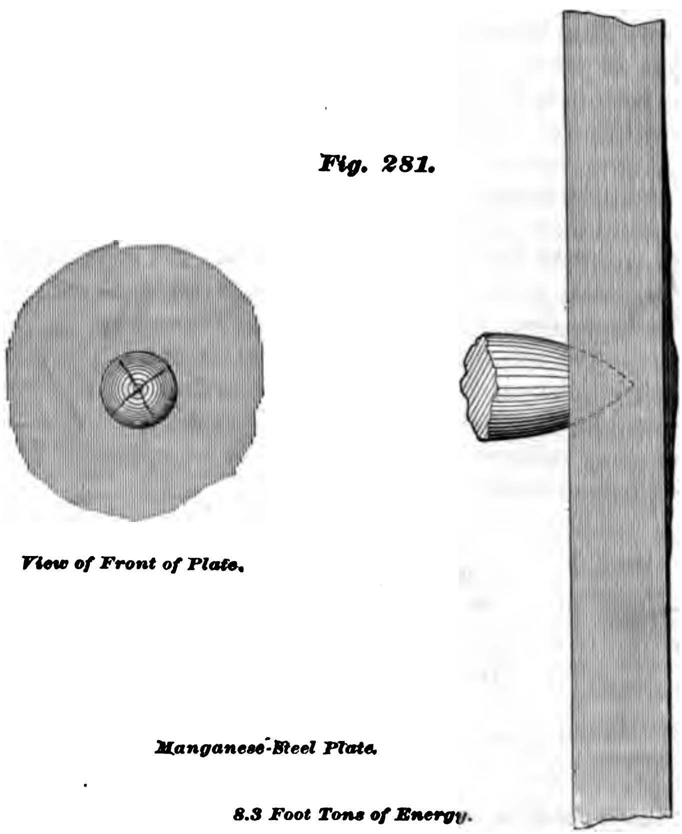
For gears exposed to excessive abrasion—*e.g.*, from grit—manganese-steel commends itself.

For safes, also, it should be well fitted, as it can neither be broken by sledging, nor cut, nor drilled.

Armor-plate.—It is to be expected that manganese-steel should make excellent armor-plates, for here hardness and toughness should be combined. The plate must be so hard that the shot does not penetrate it, yet so tough that it does not crack and let the water in; and such experimental data as we have indicate that manganese-steel will fulfil this expectation.

Figs. 278, 279, and 280 illustrate the effects of a shot on

Fig. 281.



Manganese-Steel Plate.

8.3 Foot Tons of Energy.

manganese-steel; on carbon-steel containing 0.25% of carbon; and on wrought-iron. The wrought-iron plate was completely perforated; the carbon-steel plate was cracked through; three manganese-steel plates received in all four such shots, which in two cases penetrated ½ inch, and in the other two ⅝ inch. The manganese-steel plates became somewhat dished by the blows, as is roughly shown in Fig. 281, and in two cases a slight surface crack resulted around the inside of the hole.

A sliding ram had hard steel armor-piercing shot attached to its lower face, and was dropped from a height of 8 feet upon these several plates, which were supported firmly on timber. The manganese-steel plates were 9 inches square, and 0.75 inch thick. The conditions of this test do not, indeed, reproduce those of actual warfare; in the one case we have a great mass moving slowly, in the other a small mass moving swiftly.

NOTE.—I must caution the reader that, though I have tried to treat the subject judicially, allowance should be made for bias on my part, owing to direct pecuniary interest.

DISCUSSION.

Mr. Gustavus C. Henning.—I should like to call attention to a few points in regard to manganese-steel. I tested some for Mr. Hadfield, of Sheffield, and found a very peculiar property about it—that its resistance to crushing is very slight, although it has great tensile strength. When put into the holders of a machine, notably Emery's testing machine, the end which is in the jaws will be completely crushed before you have reached the maximum tenacity. Materials can be arranged in three classes according to their relative properties in that respect. In the first of these can be grouped very hard metals of low tenacity, such as cast-iron, etc. In the second belong those in which hardness and tenacity are about equal. Then the third class is this of manganese-steel, and also some of the softer bronzes and copper, which are crushed before the maximum tenacity is reached. Hence the test piece will not break fairly, but just at the jaws of the holder; and for such cases special provision would have to be made to obtain good results. It is a peculiar quality of manganese-steel that it seems to be so soft and is still so tough and strong.

In regard to the Tables I. and II., obtained from Mr. Morrell, it is to be regretted that he does not give the amount of wear on the hardened steel shaft and on the emery-wheel, to record the effect produced on them by the other metals in contact, to give us a better idea of the actual results obtained. They are very interesting, and seem to show that a coarsely granular will have a different effect from a smooth, close-grained material, and I think that the wear of the manganese-steel should have been determined on other materials as well, not alone on the hardest and closest of the steels. It would have been very interesting in connection with this to have given another table showing how a soft, ductile material will behave with manganese-steel.

In the case of Table III. on axles, are given two columns of deflections. I think, without intending to be so, that table is very misleading. An axle is only eight or nine feet long, and if we read there, "the sum of permanent deflections or bends in inches produced," it would show that an axle less than 7 feet long had shown deflections of 105 inches. I think this apparent anomaly can only be explained by stating that the first blow would deflect the axle downward. The next blow is then given after reversing the axle on its bearings, and on a point opposite to that first struck, the axle being thus reversed after each blow. The total deflection as stated is the sum of deflections produced by each blow. The height of drop should be measured as the distance from the lower surface of the ball to the upper surface of the piece tested, and not to the bearings on which it rests.

It is a great pity that reports are given with such scant explanations as to how tests and observations were made, for without these, reports are frequently unintelligible. Of course, this is not Mr. Howe's work. A standard method of reporting is almost as necessary as to make a test. I know, as a matter of fact, in testing axles, that axles are always reversed on the bearings. Hence this apparent anomaly, which may be correct. I think it would be very desirable to have a little explanation of how these tests were made. They show that manganese-steel is very much better than the other steel; in fact, very much better than Table III. would make it appear. In Table IV. Mr. Howe has given a corrected total height of drop. In the table given, the total height of drop is merely the number of feet of fall of the ball.

The total height of the drop as there given is not a true criterion of the quality of that material. In the first place, we expect, of course, that the wheel is in the same condition at the end of the test in each case.

But I should like to object to the way the tests were made. The wheels were dropped upon the flanges. In the case of cast-iron that is a very unfair treatment of the material. The way it should be done, as recommended by the Standards Committee of our Society, is this: That a block should be fitted on the flange and tread, so as to give a reasonable bearing on the whole section of the wheel, and then the height of ball measured from this upper surface of the block to the bottom of the weight. In that manner we will obtain comparative results, and in no other. I think the cast-iron chilled wheel would have given better results in comparison with the other than those stated in the table. The results given of the cast-iron wheel are not as good as they should be, but the tests certainly show that in every case manganese-steel is very much better for most purposes, except where the resistance to crushing is the principal quality which is to be provided for.

Prof. W. A. Rogers.—I have made a few experiments upon a specimen of manganese-steel with reference to its adoption in the construction of standards of length. It was found that the particular specimen tested was capable of receiving a high polish, but under a microscopic examination the surface appeared to be an aggregation of exceedingly minute pits.

An interesting experiment was performed upon this bar of metal. After a surface of considerable extent had been ruled with rather heavy lines one-thousandth of an inch apart, the bar was placed upon an anvil and was struck a heavy blow with a large sledge-hammer without producing the slightest disfiguration of the ruled surface.

Mr. Kent.—I would like to ask Prof. Howe how manganese-steel acts in castings?

Mr. Robert W. Hunt.—I think the subject of manganese-steel as applied to steel car-wheels is certainly a very interesting and important one. You are all aware of the great and constant effort which is being made, particularly in this country, to get a solid steel wheel to replace the steel-tired ones. You probably recall the Fowler wheel, and those who are at all familiar with Mr. Fowler's labors are aware of the years of persevering investigations and experiments which he has made, and, of course, the large amount of money spent by himself and his company in that direction. At the risk of repeating what you know, allow me to recall to you that he first casts his wheel, and then places it in a rolling mill and reduces the diameter of the wheel by compression. His difficulty has been to get a metal which would stand this treatment: in the first place, metal free from physical defects; and then, one which would roll satisfactorily; and, lastly, one which would bear up under the fatigue of actual service. Latterly, within the last year or, perhaps, year and a half, he has been depending upon a soft steel made by the Robert or, as it is called in this country, Buchwalter process; and certainly has been getting wonderful results so far as solid castings are concerned. A number of analyses of his steel which I had occasion to have made ranged from 0.22 to 0.30 of carbon. That, upon its face, would seem to be entirely too soft to resist abrasion in service as a tire. But Mr. Fowler depended upon obtaining enough physical hardness from compression in rolling. I think, from some of the facts which I have been able to gather, he has been disappointed in this. They are still making wheels, and, I assume, are making them from higher carbon-steel. In New England they have another cast-steel wheel company which seem to obtain very good results from the service of their wheels. So, now, if manganese-steel will offer a cheap and constant material, giving both toughness and hardness against abrasion, it seems to me it is the thing which promises the best commercial results. Assuming that wear and safety are obtained, the next question is how it will be for cost. That is the great difficulty in regard to all solid cast-steel wheels. As long as our makers are getting such wonderful results from their best chilled wheels, it gives a knotty problem for cast-steel wheelmakers to meet. I trust, if Mr. Howe can, he will in closing the debate give us some data about this.

Mr. Gantt.—I should like to ask how much Mr. Fowler compressed those wheels.

The President.—He reduces a wheel nearly two inches—varying from an inch and a half to an inch and three-quarters.

Mr. Howe.—That is, parallel to the axle.

The President.—Yes, sir.

*Mr. Howe.**—In reply to what Mr. Henning stated, I would say that there is no doubt that manganese-steel does show a rather low compressive strength, considering its hardness in resisting abrasion; in most cases it does not stand well. There are cases in which it has stood very well, and the difference is probably due chiefly to difference in composition of manganese-steel, because some kinds have stood very well in compression. The emery-wheel tests were not Mr. Morrell's. As to the axle test, it was simply carried on under the usual conditions. The axle was always reversed after every blow, and the height was measured to the original line and not to the top of the axle. As to castings, I should say that the difference between the value of manganese-

* Author's Closure.

steel castings and manganese-steel forgings was about the same as that between carbon-steel castings and carbon-steel forgings. I doubt if a very considerable degree of distortion of the particles is required to change a casting to a forging, and to give the material practically fully the properties of a forging. It is probably a question of the temperature at which that distortion takes place. A good many things that have been used of manganese-steel—those elevator links, for instance—are castings. The castings are very hard; I do not think that the hardness is materially increased by the distortion which takes place in the forging. So that what you would get by forging is an increase in tensile strength and ductility, rather than in hardness. Of course, in the case of car-wheels—*e.g.*, that car-wheel that was dropped was a casting pure and simple, without any kind of work having been put upon it; and the other car-wheels, those that Captain Hunt refers to, were manganese-steel wheels, made in New England, and were simple castings. In many cases they gave very good results They were entirely unforged. The extra cost of manganese-steel above that of carbon-steel is chiefly due to the difference in the first cost of the material. Ferro-manganese is more expensive than cast or scrap iron, and I think one-quarter of a cent a pound ought to represent the difference between manganese-steel and carbon-steel castings

The President.—Must you provide for as much manganese-steel scrap as in carbon-steel?

Mr. Howe.—I do not think there is a very great difference. The manganese-steel shrinks and pipes a good deal. I am not able to answer that question accurately, however. I think it certainly will be as large in the case of manganese-steel as in the case of carbon-steel, and probably somewhat larger. Then there is a further difficulty in the cutting off of the sinking head and runners of this very hard steel. However, if it is done hot that does not amount to a great deal.

ADDED AFTER ADJOURNMENT.

While Mr. Henning is perfectly right in demanding full statements as to the conditions of tests and other experiments, yet this applies strictly only to the original published description. In later ones, such as my own, where the reference is simply to establish a comparison, I think that it is better to state the comparative result as briefly as possible, and to refer those who would look further to the original papers.

It is to be regretted that most engineers do not record the conditions of their tests as fully as could be desired. In their tables, as at tables of another kind, we must take what is set before us, and be thankful.

I am not sure that Mr. Henning's criticism on the drop test applied to these wheels is just. I hold that, while the test should reproduce as faithfully as possible the conditions under which the material is likely to fail in service, yet it should aim to prove the material faulty rather than to prove it good : they should assail its weakest, not its strongest points. Now I take it that the flange of a chilled cast-iron wheel is its weakest point; for instance, in striking hard against the outer rail of a curve or against a frog, is it not the flange that is liable to break, leaving the wheel free to derail itself at the first favorable opportunity? If so, then I say attack that wheel at its flange when you test it. But I should leave this question to those more competent to discuss it.

CCCCLX.*

PERFORMANCE OF A WORTHINGTON HIGH DUTY PUMPING ENGINE OF ONE AND ONE-HALF MILLION GALLONS CAPACITY PER TWENTY-FOUR HOURS, AGAINST A HEAD EQUIVALENT TO TWO THOUSAND FEET OF WATER.

BY J. E. DENTON, HOBOKEN, N. J.
(Member of the Society.)

INTRODUCTORY.

THE pumping engine under notice is in use for pumping crude petroleum over one of the eleven 30-mile sections of the Standard Oil Pipe Line connecting New York City with Olean, Pa.

It is located in a valley of the Alleghany Mountains at Swartout, about four miles from Port Jervis, New York. A bird's-eye view of the pumping station is given in Fig. 282. The general features of such a pumping station are a boiler and engine house, a telegraph office, and two receiving tanks nearly 100 feet in diameter and 30 feet in height. Oil is supplied to the tanks from another station 30 miles westward, by 6-inch pipes running across the country a few feet beneath the surface of the ground. The light streak running up the mountain at the background of Fig. 282 shows the route of the pipe. It represents the path worn by the passage of the "line walkers," who constantly patrol the line. A telegraph operator, belonging to each station, measures the depth of oil in the tanks every hour of the day and night, and reports the result throughout the line, so that the utmost system prevails, and every station is under perfect control from the headquarters. Fig. 283 shows an operator in the act of measuring the depth of oil in a tank by means of a steel tape dropped to the bottom of the tank from a reel held in his hand.

At night, sufficient light is thrown upon the tape to enable it

* Presented at the Providence meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XII. of the *Transactions*.

to be accurately read, by means of a reflector mounted upon a tower situated midway between the tanks, and at such a distance that all danger from fire is avoided. Figs. 284 and 285 show general views of the pumping engine. In the foreground are the oscillating or compensating cylinders, constituting the high duty attachment which has enabled the expansive principle of using steam to be developed in the Worthington direct-act-

FIG. 282.

Bird's-eye view of pumping station, showing boiler and engine houses, two oil receiving tanks, 95 feet in diameter and 30 feet high, with light tower between, to enable readings of depth of oil in tanks at night, as per Fig. 283. The route of the pipe line is shown by the light streak running over the mountain in the background, which represents a path worn by the passage of the "line walkers," who constantly patrol the country through which the pipe runs. Eleven such stations, about 30 miles apart, enable one and one-quarter million gallons of oil to be transported daily from Oil City, Pa., to New York City.

ing pump, so as to afford a saving of coal amounting to about 50% of the consumption involved without this attachment.

The pump tested represents the first application of the high duty attachment to the pipe-line service. This service is very severe, inasmuch as the pressure which the pump must exert often amounts to upward of 900 lbs. per square inch, and the work is continuous throughout the day and night. By aid of the devices shown in Fig. 286 and Fig. 287, the conditions of service are maintained comparatively uniform, and consequently afford excellent opportunity for the accurate measurement of the duty of a pump under the actual conditions of practice; and the object of this paper is to present the result of a series of duty trials recently made by the writer, in the joint interest of the pipe-line management and the makers of the pump. I am indebted to the officers of the pipe line, and especially to Messrs.

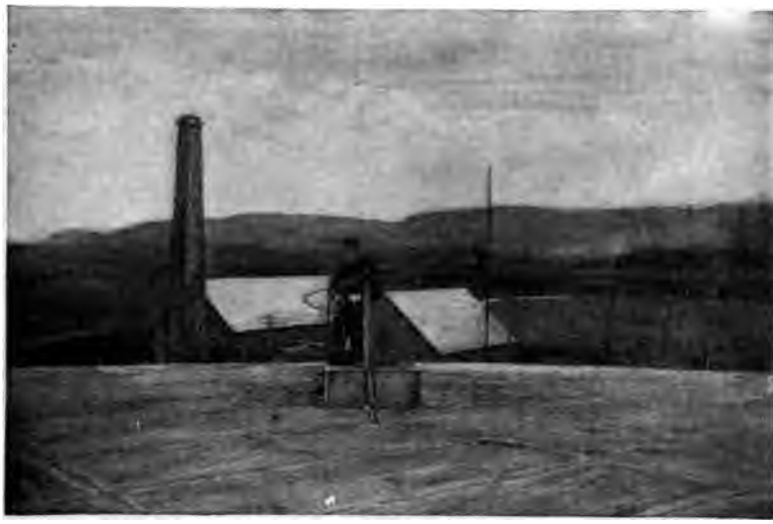


FIG. 283.

Showing station telegraph operator measuring depth of oil, in one of the receiving tanks, by means of a steel tape, dropped to the bottom, off a reel held in his hand. Measurements are made every hour, day and night, and the reading telegraphed to the adjoining stations, 30 miles distant, east and west. At night sufficient light to read the tape is thrown upon the tape by a large reflector located midway between the two tanks (Fig. 282), and at such a distance as to prevent any danger of ignition to the contents of the tanks.

Pilkington and Cobb, for cordial coöperation and assistance in carrying out the experimental work.

SPECIAL PREPARATIONS FOR TESTS.

The station is provided with eight boilers, six of which are in use at one time to supply steam to operate the pumping engine, an independent steam feed-pump, a small engine operating a dynamo and for heating the telegraph office and engine house. A sectional view of the engine is shown in Fig.

291. The air pumps are operated by direct connections from the cross heads. All of the cylinders are steam-jacketed, and the steam condensed in the jackets is returned to the boilers by an automatic Pratt & Cady steam-trap. The remaining portion of feed-water is taken from the hot well by the independent steam feed-pump, and forced through a pipe surrounded, over a portion of its length, by a larger pipe, into which the steam

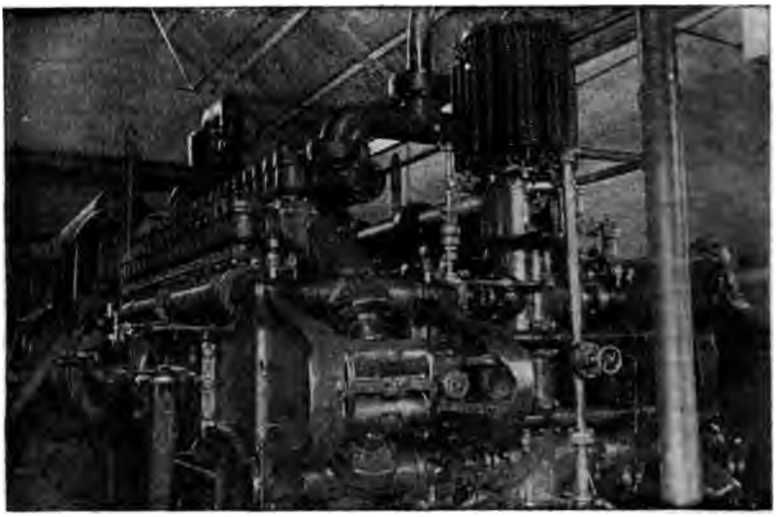


FIG. 284.

General view of pumping engine, showing 6-inch vertical delivery pipe, compensating or oscillating cylinders, and the group of springs acting as an equivalent of an air chamber. The springs resist the upward movement of a vertical plunger, 6 inches diameter, which works in and out of a cylinder connected with the delivery chamber of the pump by the 2-inch vertical pipe in front of the pump. The oscillating cylinders are discharged or filled by oil flowing through their hollow trunnions, to or from the cylinder containing this plunger, and the springs control the fluctuations of pressure which occur.

feed-pump and the dynamo engine and heating coils exhaust, thereby slightly heating the feed-water, taken from the hot well, above the temperature in the latter.

The condensed steam from the jackets was cut off from the automatic trap and taken through a Nason dumping trap into a surface condenser, which cooled it sufficiently to allow it to be received in an open barrel, and thence pumped through a three-quarter Worthington meter, by means of a small steam pump provided for the purpose. The feed-water then all entered the boilers through the independent feed-pump, and was measured by a 2-inch Worthington meter located between this pump and the boilers. One of the six boilers was devoted to supplying steam for all purposes, except running the pumping engines, a separate steam main connecting it with the dynamo engine, the heating coils, the feed-pump, and the small pump used for forcing the jacket steam through the water. A second three-

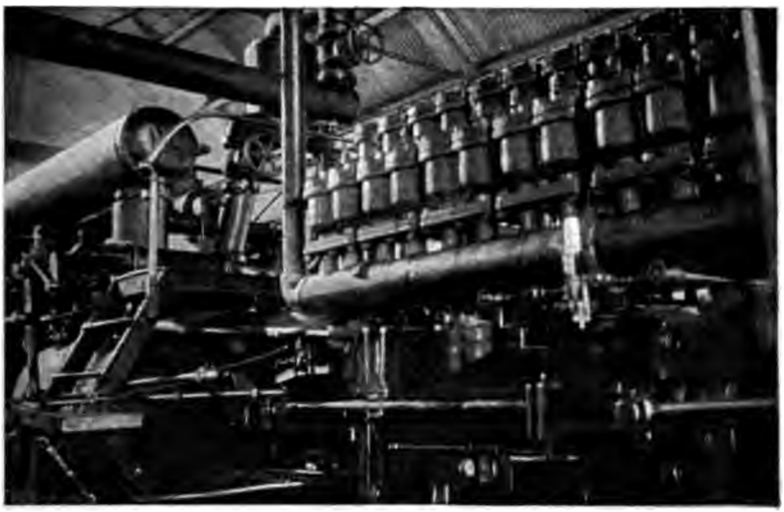


FIG. 285.

Showing lazy tongs, indicator motion, main steam-pipe, main throttle valve, branch throttle valves to each side of engine and half-inch vertical pipe, to which the universal calorimeter was attached to measure moisture in steam supplied to engine; also 1¼-inch pipe, out of bottom of steam-pipe, directly below main throttle valve, which supplies steam to all jackets, and to reheater in receivers.

quarter meter measured the feed-water supplied to this boiler. Hence, the feed-water registered by the 2-inch meter, less that registered by the three-quarter meter attached to the single boiler, was the amount of water evaporated into steam to operate the pumping engine under test, and included both the steam condensed in the jackets and that entering the cylinders. Coal was weighed in portions of about a ton each on the scales with which the station is provided, the latter being calibrated with an accurate steel-yard provided by the writer, and by means of which, dead weight was accumulated equal to the greatest weight for which the scales were used. Indicators were fitted to all the steam and pump cylinders, and were operated by a lazy-tongs motion. A "stroke register" was provided to integrate automatically the lengths of the strokes of the pump during any interval of time.

A Barrus universal calorimeter was attached to the steam

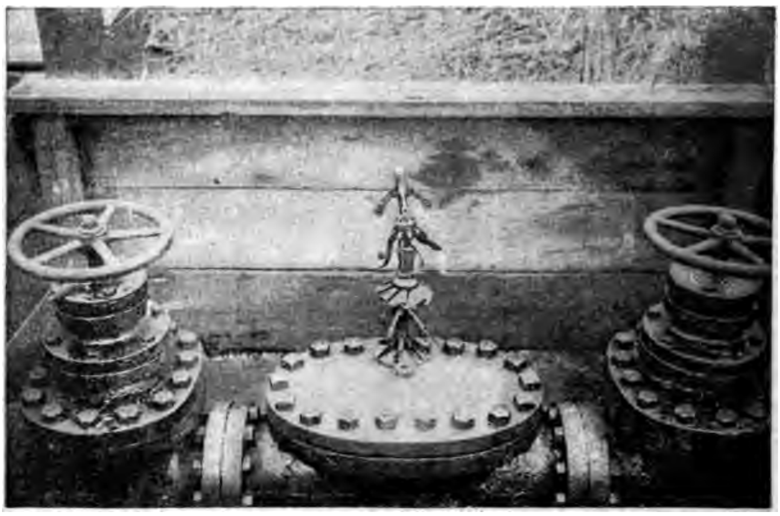


FIG. 286.

Showing the "go devil," or instrument which is allowed to travel through the oil pipe, to cleanse the interior of the latter of oil accumulations tending to obstruct the flow. The stop valves at either side are closed, and then the oblong man-hole plate removed. The instrument is then laid in the pipe, the cover replaced, and the valves opened. The current of oil, which travels at the rate of about three miles per hour, pushes the "go devil" along, and also revolves it with sufficient force to cause the bent steel knives, near its upper end, to cut all residue off from the interior of the pipe, and thereby prevent undue expense of power in pumping the oil. The instrument creates an audible buzzing sound, which, together with the fact that its arrival is anticipated by telegraph notice, enables it to be caught at the next station, or as soon as it arrives at the proper depot.

supply pipe immediately before the latter entered the main throttle valve. (See description, Fig. 285.)

Steam-gauges, carefully calibrated, were located at the boiler, at the steam-pipe, at its entrance to the engine, just beyond the main throttle, and below the branch throttles leading to either side of the engine. A steam-gauge and thermometer were located on the pipe, through which all steam condensed in the jackets flowed to the Nason trap, at a point about 3 feet from the centre line of the engine.

DESCRIPTION AND CALIBRATION OF STROKE REGISTER.

One of these instruments is shown in Fig. 288. It consists of a ratchet-wheel attached to the primary arbor of the train of gear-wheels, and indices, constituting the recording mechanism

FIG. 287.

Showing yard containing boiler, engine, and telegraph houses and machine shop; also, in the foreground, the jacket or sleeve clamped on the outside of the line pipe, over a break or rupture in the latter.

A rubber packing ring is squeezed together upon the pipe at each end of the casting, and a gland or stuffing-box attached by means of the stud bolts shown. This device enables a leak to be controlled without removing the ruptured length of pipe from the line.

of water meters. Mounted on the same arbor with the ratchet-wheel is a pinion gearing into a rack attached to the sliding piece *A*, and the latter carries a Stubbs steel rod upon which are mounted the stops *BB*, which are adjustable, so that the projecting points *CC* may be set any desired distance apart. The teeth of the ratchet are so finely cut that by the aid of two pawls, set so as to subdivide a tooth-space, a movement of the rack as small as one-thirty-second of an inch will move the primary index of the recording mechanism. If, therefore, the apparatus is mounted so that an arm clamped to the cross-head of the pumping engine can strike the projecting points *CC*, and the distance of the latter apart be any amount, not more than 2 inches, less than the maximum stroke of the engine, the recording mechanism will register numbers representing the total travel of the engine, in excess of that corresponding to a length of stroke equal to the distance apart of the projecting points *CC*, plus the thickness of the arm attached to the cross-head. Thus, in the present case, the stroke of the engine was about 37$\frac{3}{4}$ inches. The stops, or projecting points *CC*, were therefore set so that their distance apart was 36 inches, plus the thickness of the arm fastened to the cross-head of the engine. The proportions of the ratchet and rack motion were such that a movement of the

FIG. 288.

rack equal to five-ninths of an inch moved the primary hand of the recording mechanisms one unit of its dial, or one-tenth of a revolution. Hence the average length of stroke made by the engine was given as follows :

Let
N_1 = Initial reading of revolution counter of engine.
N_2 = Final " " " " " "
R_1 = Initial reading of stroke register.
R_2 = Final " " " "
Then average length of stroke in inches is

$$L = 36 + \frac{5\,(R_2 - R_1)}{9\,(N_2 - N_1)}.$$

The instruments were calibrated by moving the rack a known number of times between stops a measured distance apart, before attachment to the engine. This calibration was also checked by several series of observations of the movement of the rack A, while attached to the engine. The rack being at rest while the piston of the engine is travelling 36 inches, a scale can be placed beneath it, and the excess of piston travel, over 36 inches, be easily read off each stroke. Examples of such a series of observations, one for each side of the engine, are given below.

LENGTH OF STROKE IN EXCESS OF 36 INCHES FOR 100 DIVISIONS OF PRIMARY INDEX.

North Side.				
No. of Double Strokes.	Rack Movement. Inches.	No. of Double Strokes.	Rack Movement. Inches.	
1	$1\frac{8}{16}$	19	$1\frac{9}{16}$	Total rack movement, 54.53 inches; or roundly, $\frac{5}{9}$ inch per division of primary index.
2	$1\frac{9}{16}$	20	$1\frac{8}{16}$	
3	$1\frac{9}{16}$	21	$1\frac{9}{16}$	
4	$1\frac{9}{16}$	22	$1\frac{9}{16}$	
5	$1\frac{9}{16}$	23	$1\frac{9}{16}$	
6	$1\frac{9}{16}$	24	$1\frac{9}{16}$	
7	$1\frac{9}{16}$	25	$1\frac{9}{16}$	
8	$1\frac{9}{16}$	26	$1\frac{9}{16}$	
9	1 [illegible]	27	$1\frac{8}{16}$	
10	$1\frac{9}{16}$	28	$1\frac{7}{16}$	
11	$1\frac{8}{16}$	29	$1\frac{8}{16}$	
12	$1\frac{9}{16}$	30	$1\frac{8}{16}$	
13	$1\frac{9}{16}$	31	$1\frac{7}{16}$	
14	$1\frac{9}{16}$	32	$1\frac{8}{16}$	
15	$1\frac{9}{16}$	33	1 [illegible]	
16	$1\frac{9}{16}$	34	$1\frac{8}{16}$	
17	1 [illegible]	35	$1\frac{8}{16}$	
18	$1\frac{9}{16}$	36		

LENGTH OF STROKE IN EXCESS OF 36 INCHES FOR 150 DIVISIONS OF PRIMARY INDEX.

South Side.			
No. of Double Strokes.	Rack Movement. Inches.	No. of Double Strokes.	Rack Movement. Inches.
1	$1\frac{5}{16}$	32	$1\frac{5}{16}$
2	$1\frac{4}{16}$	33	$1\frac{5}{16}$
3	$1\frac{3}{16}$	34	$1\frac{4}{16}$
4	$1\frac{4}{16}$	35	$1\frac{4}{16}$
5	$1\frac{5}{16}$	36	$1\frac{6}{16}$
6	$1\frac{4}{16}$	37	$1\frac{4}{16}$
7	$1\frac{5}{16}$	38	$1\frac{5}{16}$
8	$1\frac{5}{16}$	39	$1\frac{6}{16}$
9	$1\frac{5}{16}$	40	$1\frac{6}{16}$
10	$1\frac{5}{16}$	41	$1\frac{4}{16}$
11	$1\frac{4}{16}$	42	$1\frac{5}{16}$
12	$1\frac{5}{16}$	43	$1\frac{5}{16}$
13	$1\frac{4}{16}$	44	$1\frac{5}{16}$
14	$1\frac{5}{16}$	45	$1\frac{5}{16}$
15	$1\frac{5}{16}$	46	$1\frac{6}{16}$
16	$1\frac{6}{16}$	47	$1\frac{5}{16}$
17	$1\frac{5}{16}$	48	$1\frac{4}{16}$
18	$1\frac{6}{16}$	49	$1\frac{4}{16}$
19	$1\frac{4}{16}$	50	$1\frac{4}{16}$
20	$1\frac{5}{16}$	51	$1\frac{5}{16}$
21	$1\frac{4}{16}$	52	$1\frac{6}{16}$
22	$1\frac{4}{16}$	53	$1\frac{5}{16}$
23	$1\frac{4}{16}$	54	$1\frac{5}{16}$
24	$1\frac{4}{16}$	55	$1\frac{9}{16}$
25	$1\frac{4}{16}$	56	$1\frac{6}{16}$
26	$1\frac{3}{16}$	57	$1\frac{6}{16}$
27	$1\frac{4}{16}$	58	$1\frac{9}{16}$
28	$1\frac{4}{16}$	59	$1\frac{7}{16}$
29	$1\frac{5}{16}$	60	$1\frac{6}{16}$
30	$1\frac{7}{16}$	61	$1\frac{4}{16}$
31	$1\frac{6}{16}$	62	$1\frac{6}{16}$

Total rack movement, 81.3 inches; or roundly, $\frac{5}{9}$ inch per division of primary index.

DESCRIPTION AND CALIBRATION OF INDICATOR APPLIED TO PUMP CYLINDERS.

This consisted of a steel plunger one-quarter of an inch in diameter, which received the pressure of the pump cylinders against its lower end through the medium of a mass of cylinder oil, and communicated this pressure to the piston of a Crosby indicator, against which the upper end of the plunger abutted. The apparatus, seen as adjusted for calibration and on the engine, respectively, is shown in Figs. 289 and 284. The brass siphon *A* contained the cylinder oil, and the quarter-inch plunger works oil-tight through the hexagonal nut *B*. A three-way cock at *C* is set so that there is communication through it, with the atmosphere, while communication toward the pump cylinders is closed. The cock *D* being then opened, a charge

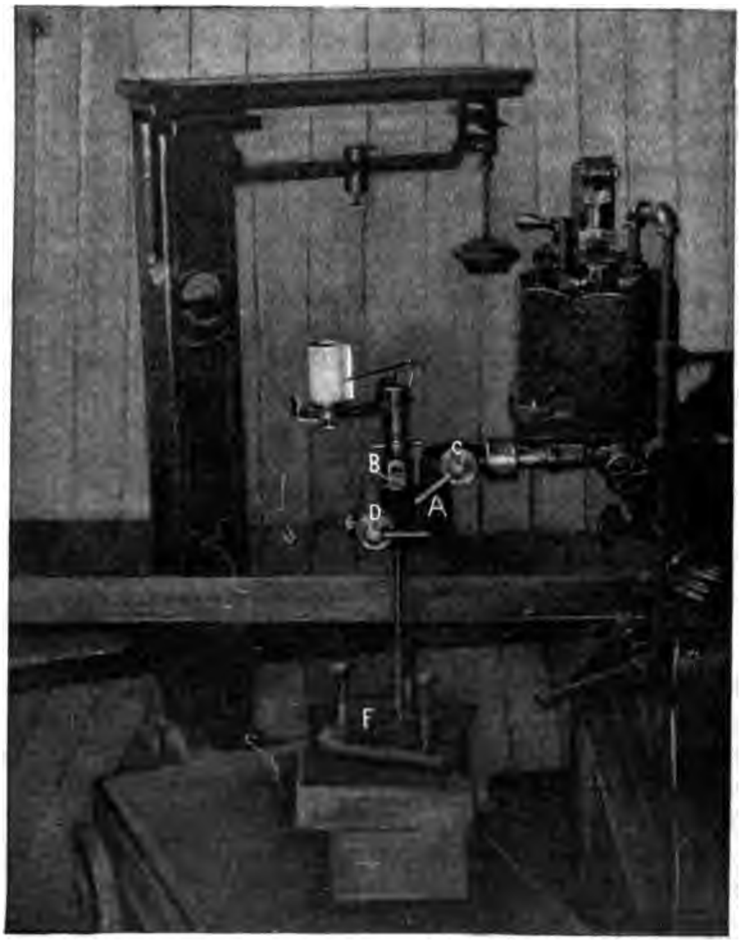


FIG. 289.
INDICATOR FOR PUMPS.

of cylinder oil was introduced with a syringe, until this oil was seen escaping to the atmosphere at *C*.

The cock *D* is then closed, and *C* turned so as to communicate with the cylinders, and allow the pump pressure to act upon the plunger.

There was practically no leakage with the plunger fitted a thousandth of an inch less in diameter than the hole in the nut

B. The instrument was filled, or recharged, only when it was thought that the crude oil from the pump cylinders may have taken the place of the cylinder oil, through the escape of the latter by the frequent setting of the cock *C* to connect with the atmosphere, in taking the atmospheric lines of cards. To calibrate the instrument, it was emptied of oil and a plug directly under the plunger removed. The brass rod *E* was then adjusted so that, by working the levelling screws in the platform *F*, the indicator pencil was moved over its range of travel. The average readings of the platform scale, corresponding to each sixth of an inch of pencil travel, gave the scale of the instrument as 610 lbs. per inch.

As a check upon this method of calibration, the spring in the Crosby instrument and a small spring attached to the plunger, to enable it to measure pressures below the atmosphere, were calibrated separately, and gave the same value for the scale of the instrument.

It is probable that this value is correct to within about 1½%. As no mercury column exists which affords a means of calibrating pressure gauges to pressures as high as 900 lbs. per square inch, this indicator was the sole basis of oil pressure measurement available, and the several Bourdon gauges used for recording the oil pressure were all referred to this instrument. Both the stroke register and the hydraulic indicator were designed, at the request of the writer, by Prof. J. B. Webb. The steam indicators were calibrated from 27 inches of vacuum to 20 lbs. above the atmosphere, with mercury columns, and for higher pressures, by means of the "dead weight test-plate" of the Utica Gauge Co. In the course of the experiments seven Ashcroft indicators and one Thompson indicator were used. Four of the Ashcroft were new, being kindly loaned the writer by the Ashcroft Co. The scales of the latter were found correct to within 1%. The water meters were all calibrated in place, under the exact conditions of pressure and rate at which they were used. Calibrations were made before and after the experiments, and the variations amounted to only four-tenths of 1%.

PRINCIPAL DIMENSIONS OF ENGINE AND BOILERS.

The boilers were of the return tubular type, the gases passing once under shell, once through the tubes, and thence to the chimney.

Diameter of shell	5 ft.
Number of tubes, each boiler	82 "
Diameter of tubes	3 in.
Length of boiler and tubes	14 ft.
Width of grate	4 ft. 8 in.
Length of grate	6 ft.
Grate surface per boiler	28 sq. ft.
Heating surface in tubes per boiler	895 "
Heating surface in shell "	111 "
Total heating surface "	906 "
Superheating surface	none
Ratio of heating to grate surface	32

The engine was of the tandem compound type, with a receiver between each pair of high and low cylinders, the steam being heated while passing through a receiver by contact with a nest of piping filled with steam at boiler pressure. (See Fig. 291.) Both the barrels and heads of all cylinders were jacketed with steam taken directly from the boiler. Steam for the jackets and the reheater in receiver was taken from the main steam-pipe by an inch and a quarter pipe leaving the bottom of the latter immediately under the main throttle valve. (See Fig. 285.) From this pipe, branch pipes, one and one quarter inches diameter, led steam to the reheater and jackets. The drainage from the reheater passed through the cylinder jackets, and the drains from the latter all united in a single pipe underneath the cylinders, leading to a Nason trap in the case of the test, but to an automatic steam return trap on top of the boilers, in the ordinary running of the station. The pump plungers were of the outside-packed style, working through stuffing boxes packed with rubber and hemp. To distinguish between the two sides of the engine, we designate as the No. 1 engine the left-hand side looking from the steam toward the pump end; and as No. 2 engine, the right-hand side from the same stand-point.

DIMENSIONS OF ENGINES.

Diameter of both high-pressure cylinders	33 in.
" " low-pressure cylinders	66 "
" " four pump cylinders	$9\frac{1}{4}$ "
" " low-pressure piston rod	$5\frac{1}{4}$ "
" " high-pressure piston rod	$5\frac{1}{2}$ "
" " piston of both air pumps, double acting	$1\frac{7}{16}$ "
Average stroke of air pumps	$37\frac{1}{2}$ in.
Total possible travel of No. 1 engine pistons	$37\frac{84}{100}$ in.
Distance between prick punch marks, No. 1 engine	$37\frac{5}{8}$ "
Average travel determined by stroke register	$37\frac{34}{100}$ "

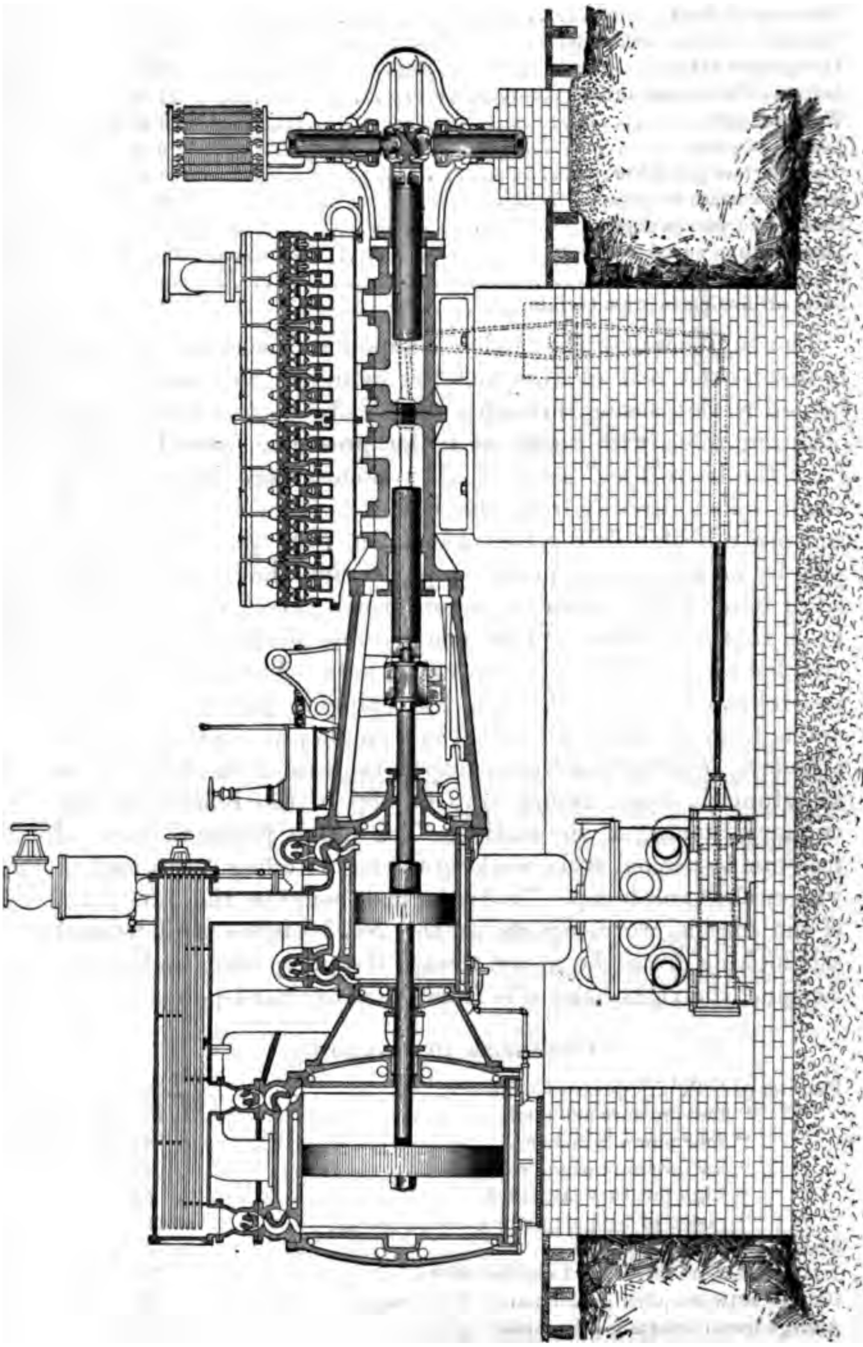


FIG. 201.

Total possible travel, No. 2 engine pistons	37$\frac{88}{100}$ in.
Distance between prick punch marks, No. 2 engine	37$\frac{79}{100}$ "
Average travel determined by stroke register	37$\frac{71}{100}$ "
Ratio of high piston area to cut-off port area	.021
Ratio of high piston area to exhaust port area	.0468
Ratio of low piston area to cut-off port area	.0145
Ratio of low piston area to exhaust port area	.0211
Net area, each low piston	3,410 sq. in.
Net area, each high piston	832 "
Area of each pump plunger	67$\frac{3}{10}$ "
Ratio of low piston area to high piston area	4.097
Ratio of low piston area to pump plunger area	50$\frac{7}{10}$

Clearance of high piston in per cent. of average piston displacement, both engines:

Cut-off ports	0.52%
Outside of exhaust valve	0.79%
Exhaust ports	0.75%
At ends of cylinders—average, both engines	0.45%
Total average clearance, high cylinders	2.51%

Clearance of low piston in per cent. of average piston displacement, both engines:

Cut-off ports	0.31%
Outside of exhaust valve	$\frac{5}{8}$%
Exhaust ports	0.44%
At ends of cylinders—average, both engines	0.45%
Total average clearance, low cylinders	1$\frac{7}{8}$%

Live steam-heating surface in receiver, each engine	228 sq. ft.
Ratio of cubic contents of receiver space, " " to contents of high cylinder	0.7
Number of inlet pump valves, " "	8
Total area inlet pump valves, " "	40 sq. in.
Area of inlet valves in per cent. of plunger area	59%
Number of outlet valves, each engine	8
Total area of outlet valves, " "	40 sq. in.
Diameter of compensating piston	6 in.
Stroke of compensating piston	9 "
Diameter of trunnions of compensating cylinders	6 "
" " thrust bearing of compensating piston	7 "
" " main steam supply pipe	6 "
Length of main steam supply pipe	100 ft.
Cubic contents of foundation, both engines	980 cu. ft.

RESULTS OF EXPERIMENTS.

PERFORMANCE OF BOILERS.

Four boiler-tests were made, as follows:

First.—A twenty-four hour test, April 1st, using pea coal, and six boilers all connected, no separation being made of the steam used by the various motors.

Second.—A twenty-four hour test, April 3d, using pea coal, five boilers being devoted to the pumping engine and a sixth to the other motors.

Third.—A twenty-four hour test, April 6th, using anthracite stove coal, five boilers being used for the engine, and one for the other motors.

Fourth.—A twelve-hour test, April 8th, using anthracite stove coal, four boilers being used for the engine, and one for the other motors.

The fires were not drawn in starting the boiler test, but the time of starting was made to coincide with the hour when the fires were all cleaned, 1 o'clock—the regular habits of the firemen being regarded as insuring as much accuracy, by this plan, as would ensue from attempting to draw the fires of so many boilers, in order to start with wood.

BOILER ECONOMY.

	Pea Coal.		Anthracite Coal.	
	First Test.	Second Test.	First Test.	Second Test.
Date of test	Apr. 1.	Apr. 3.	Apr. 6.	Apr. 8.
Duration of test	24 hrs.	24 hrs.	24 hrs.	24 hrs.
Boilers in use under equal conditions	6	5	5	4
Pounds consumed per hour	940	865	839	800
Pounds water evaporated per hour from actual temperature of feed and at actual boiler pressure	8225	7574	7489	7523
Average temperature of feed, degrees Fahr.	118	119	113	116
Average boiler pressure	89	90	90	89
Pounds actual evaporation per pound of coal.	8.78	8.74	8.91	9.46
Ashes, per cent			10.0	11.0
Temperature of feed-water if jacket water returned to boiler as in usual working of station, degrees Fahr.	147	151	146	147
Pounds evaporation per pound of coal for 147° temperature of feed and actual steam pressure	9.09	9.04	9.22	9.79
Pounds evaporation per pound of coal from and at 212° Fahr.	9.94	9.89	10.14	10.71
Pounds evaporation per pound of combustible from and at 212°			11.26	11.91
Chimney temperature, degrees Fahr.	340	360	390	410
Condition of steam	dry	dry	dry	dry
Pounds coal consumed per square foot of grate per hour	5.6	6.2	6.0	7.1
Pounds water evaporated per hour per square foot of heating surface from and at 212° Fahr.	1.77	1.89	1.88	2.25

The single boiler had to be operated with the fire doors open It evaporated about 410 lbs. of water per hour, with an economy of 6¼ lbs. of water per pound of coal at actual steam pressure and from actual feed-temperature. The steam consumed by the feed-pump was probably about 1% of that used by the main engine.

This corresponds to a duty of 10,000,000 foot lbs., and an allowance of 5% for friction. If the exhaust of the feed-pump was absorbed in the feed-water, the temperature of the latter would be increased about 10°. The expense of operating the feed-pump would then increase the steam-consumption of the main engine one-eighth of 1%.

The boiler-flues were cleaned on March 28th and on April 4th. The interior surfaces of the boilers were clean, and quite free from scale.

There was no leakage at the blow-off valves of the boilers, but some of the safety-valves leaked slightly.

PERFORMANCE OF ENGINE.

No attempt was made to adjust the engine especially for testing. It was run under exactly the conditions of its operation for the previous six months, without any extra attendance or care. No test was made to determine the leakage of the valves or pistons.

The engine differed from the standard construction of Worthington Water Works Engine in the following details instituted by the pipe-line management.

First.—The admission-valves of both the high and low cylinders were fitted with strips lying at an angle with the edge of the port so as to cause the latter to open gradually.

Second.—The steam-cylinders were lagged simply with a 2-inch layer of asbestos covered with tarred canvas.

ENGINE ECONOMY.

THROTTLE VALVE AND VALVE BETWEEN RECEIVERS WIDE OPEN.

1	Date of test	April 3.	April 6.
2	Duration of test	24 hours.	24 hours.
3	Average steam pressure at boiler, in pounds, per square inch above atmosphere	89	89
4	Average steam pressure at engine, in pounds, per square inch above atmosphere	88	89
5	Average steam pressure under main throttle, in pounds, per square inch above atmosphere	84	85
6	Average steam pressure under branch throttle, in pounds, per square inch above atmosphere	82	83
7	Average steam pressure at bottom of steam jackets, in pounds, per square inch above atmosphere	82	83

8	Average steam pressure during admission to high cylinder, in pounds, per sq. in. above atmosphere..	74	74
9	Average oil pressure at delivery, in pounds, per square inch above atmosphere	872	858
10	Average oil pressure at suction, in pounds, per square inch above atmosphere........................	0	0
11	Average vacuum, inches of mercury.................	27.25	27.25
12	" barometric pressure, inches of mercury... ..	29.75	29.75
13	" temperature outlet to jackets, degrees Fahr..	325	326
14	" " feed-water....................	119	113
15	Temperature of feed-water, if drainage from jackets had returned to the boilers, degrees Fahr.........	151	146
16	Thermal units per pound of steam at boiler pressure..	1214	1214
17	" " to evaporate 1 pound of steam, if jacket water was returned to boiler.....................	1064	1068
18	Total pounds steam supplied engine per hour........	7574	7489
19	Per cent. of latter condensed by jackets and reheater..	15.5	16.1
20	Average length of stroke in feet...................	3.127	3.128
21	" revolutions per minute.....................	20.16	20.13
22	" feet of piston travel per minute.............	126	125.9
23	" horse-power at pump end	446.4	439.9
24	Pounds of steam condensed per hour per pump horse-power...	16.96	17,02
25	Pounds of coal per hour per pump horse-power, assuming 10 pounds of steam to be evaporated per pound of coal..	1.696	1.702
26	Foot pounds duty per 100 pounds of coal at 10 pounds evaporation..................................	116730000	116330000
27	Foot pounds duty per 1,000,000 thermal units of steam consumption, neglecting steam consumed by feed-pump..	110000000	109000000
28	Horse-power to operate feed-pump..................	0.9	0.9
29	Foot pounds duty per million heat units of steam consumption, including steam consumed by feed-pump, assuming latter to give a duty of 10,000,000 foot pounds under actual conditions, and to exhaust into the feed-water	109850000	108860000
		Pea coal.	Anthracite coal.
30	Foot pounds duty per 100 pounds of coal at average rate of evaporation, under working conditions.....	105395000	110943000
31	Thermal units of steam consumption per pump horse-power per minute..............................	300	302
32	Average mean effective pressure, high cylinders, pounds per square inch..		34.08
33	Average mean effective pressure, low cylinders, pounds per square inch..		9.02
34	Average mean effective pressure, all cylinders, per square inch of low piston		17.53
35	Total horse-power, steam end....................................		456.7
36	Pounds water consumption per steam horse-power by measurement..		16.40
37	Friction of mechanism per cent. of steam horse-power..........		3.6
38	Thermal units of steam consumption per steam horse-power per minute		291
39	Average apparent cut-off high cylinder.........................		0.33
40	Total real expansions by volume, both cylinders................		13.76
41	Per cent. of total feed-water not accounted for at cut-off high cylinder..		34.0
42	Ditto at release of high cylinder..............................		30.0
43	Ditto at cut-off of low cylinder		12.0
44	Ditto at release of low cylinder...............................		12.0
45	Moisture in steam entering engine.............................		1.1%
46	Oil consumed in cylinders, gallons per 24 hours		1.0
47	Oil consumed on outside bearings, gallons per 24 hours....... .		1.5

NOTES REGARDING FIGURES IN ABOVE TABLE.

First.—The oil pressure given in item 9 represents both the mean effective pressure in the pump cylinders, and the pressure shown by a gauge attached to the delivery chamber of the pump; because there was no practical pressure required to operate the delivery valves. This fact is shown by the sample pump cards (Fig. 292), in which the full lines were taken with the indicator attached to the pump cylinders, or inside the delivery valves, and the dotted line, with the indicator attached to the same pipe as the station pressure gauge, which shows the pressure in the delivery chamber of the pump, or outside the delivery valves.

CARDS TAKEN AT OIL END OF ENGINE.

Scale, 610 lbs. = 1 inch.

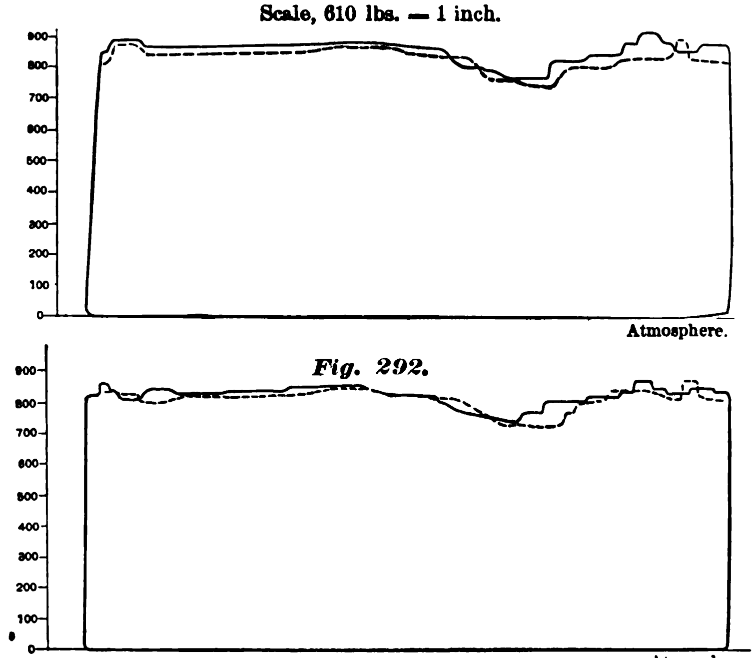


Fig. 292.

Full line—Indicator connected to pump cylinder inside of outlet valves.

Dotted line—Indicator connected to delivery chamber outside of outlet valves.

Second.—The duty, item 25, is calculated as follows: Test of April 6th.

Oil press.	Area plungers.	No. plungers.	Revs. per min.	Stroke.	Pump H.P.
858 ×	67.2 ×	4 ×	20.13 ×	3.127	

$$\frac{858 \times 67.2 \times 4 \times 20.13 \times 3.127}{33000} = 439.9$$

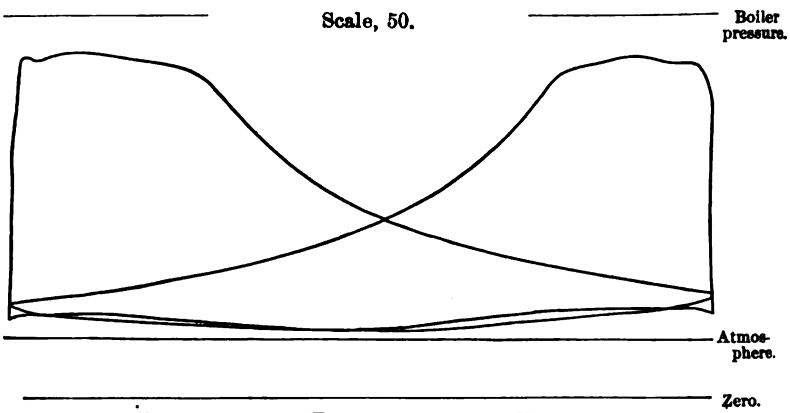
Water per hour.		Pump H.P.		Water per hr. per pump H.P.
7.489	÷	439.9	=	17.02 lbs.
Water per pump H.P.		Assumed evaporation per lb. of coal		Coal per hr. per pump H.P.
17.02	÷	10	=	1.702 lbs.
Ft. lbs. per hr. for one H.P.		Coal per hr. per pump H.P.		Duty per one lb. of coal,
1980000	÷	1.702	=	1163300
				Duty per 100 lbs. of coal.
1163300	×	100	=	116330000
		Ratio, item 17, to 1000000		Duty per 1000000 thermal units.
116330000	÷	1.068	=	109000000

Third.—The steam indicator cards were taken from both ends of each cylinder with one indicator, located at a tee midway between the ends.

Such an arrangement, with high speeds of rotation, is inadmis-

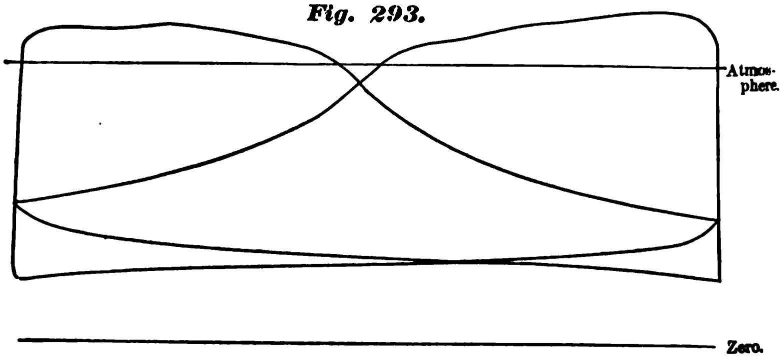
TEST OF APRIL 6TH.

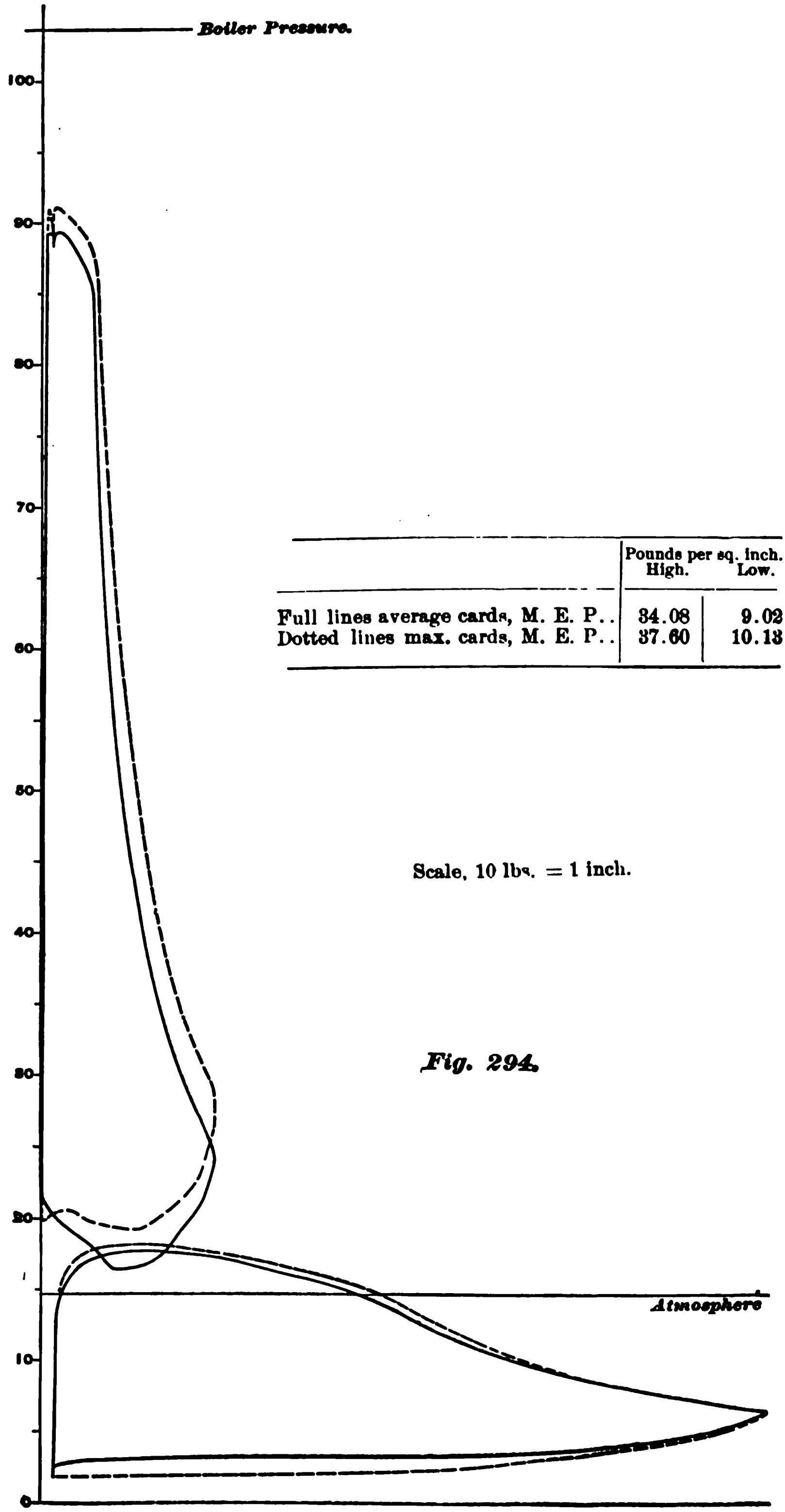
Scale, 50.



Cards of average M. E. P. combined in full lines, Fig. 294.
Scale, 10.

Fig. 293.





	Pounds per sq. inch. High.	Low.
Full lines average cards, M. E. P..	34.08	9.02
Dotted lines max. cards, M. E. P..	37.60	10.13

Scale, 10 lbs. = 1 inch.

Fig. 294.

TEST OF APRIL 6TH.

Scale, 50.

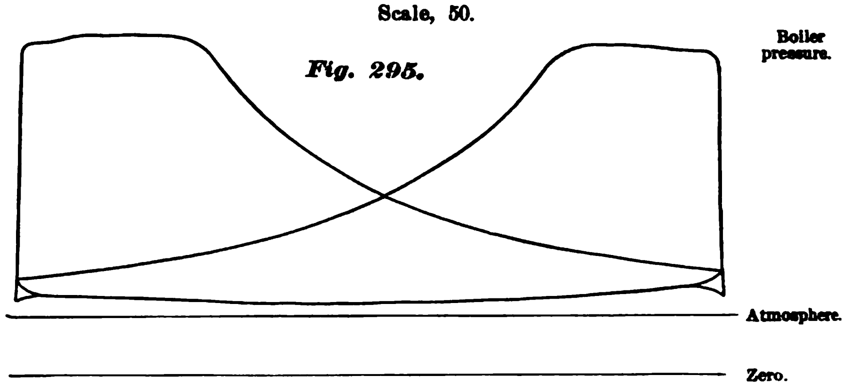
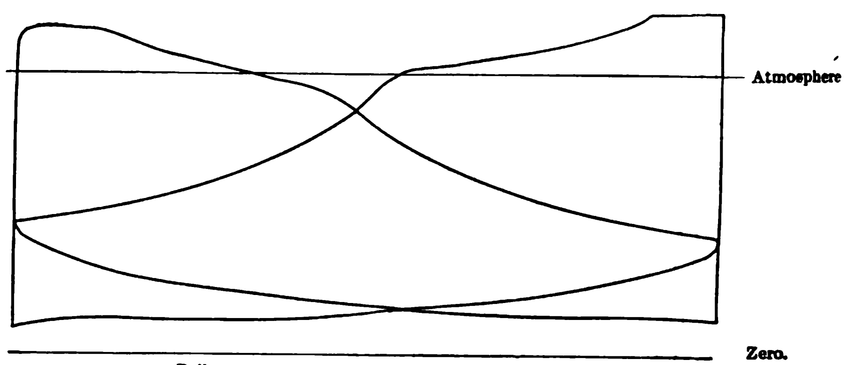


Fig. 295.

Cards of best form combined in Fig. 296.

Scale, 10.



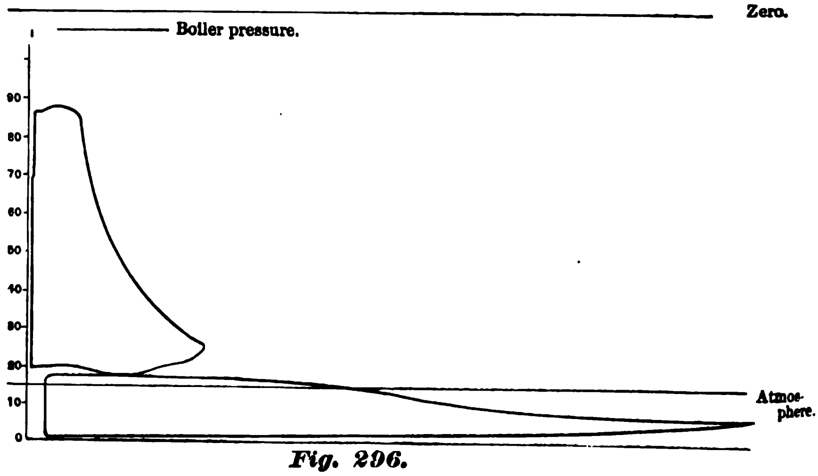


Fig. 296.

Scale, 50 lbs. = 1 inch.

sible, but it was determined in the present instance that the arrangement gave exactly the same results as the use of separate indicators at each end of the steam-cylinders, with only a straight half-inch nipple between the indicator cock and the interior of the cylinder.

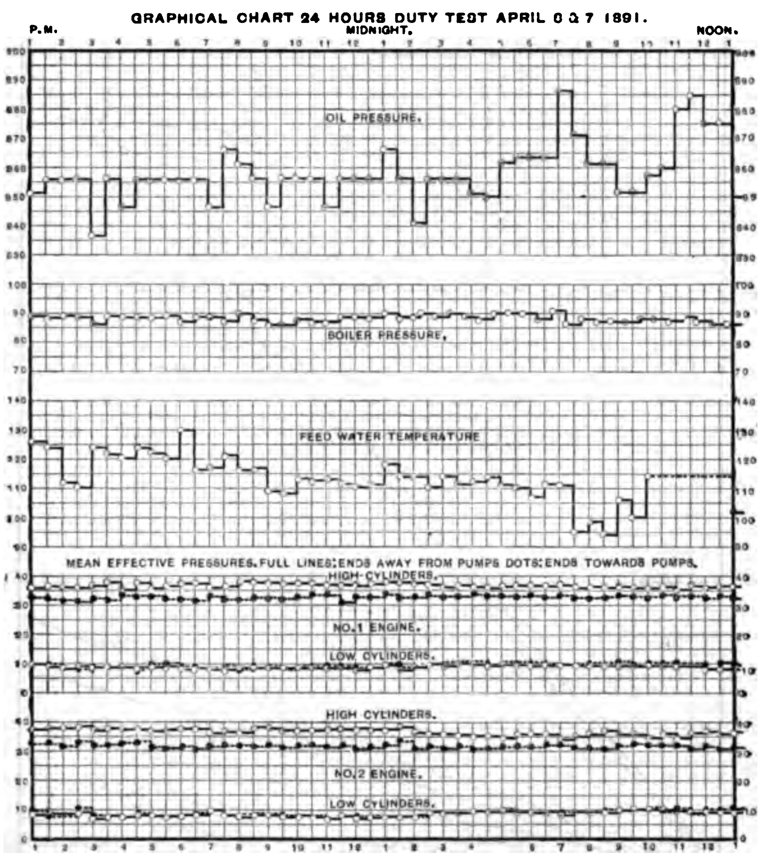


Fig. 297.

Sample indicator cards are shown in Figs. 293, 294, 295, 296; and their range of variation in mean effective pressure is shown by the graphical log, Figs. 297 and 298, and the tables in Appendix. It may be noticed that the variation in mean effective pressure is comparatively large, amounting to about 10% for the high cylinder and 15% for the low cylinder. Such variations are believed to accurately represent the accidental discrepancies to which the best work with indicators is subject. The effect of the irregularities is not great, however, with so many observations. The method of least squares shows that the probable error of the arithmetical mean of the mean effective pressures, reduced to the

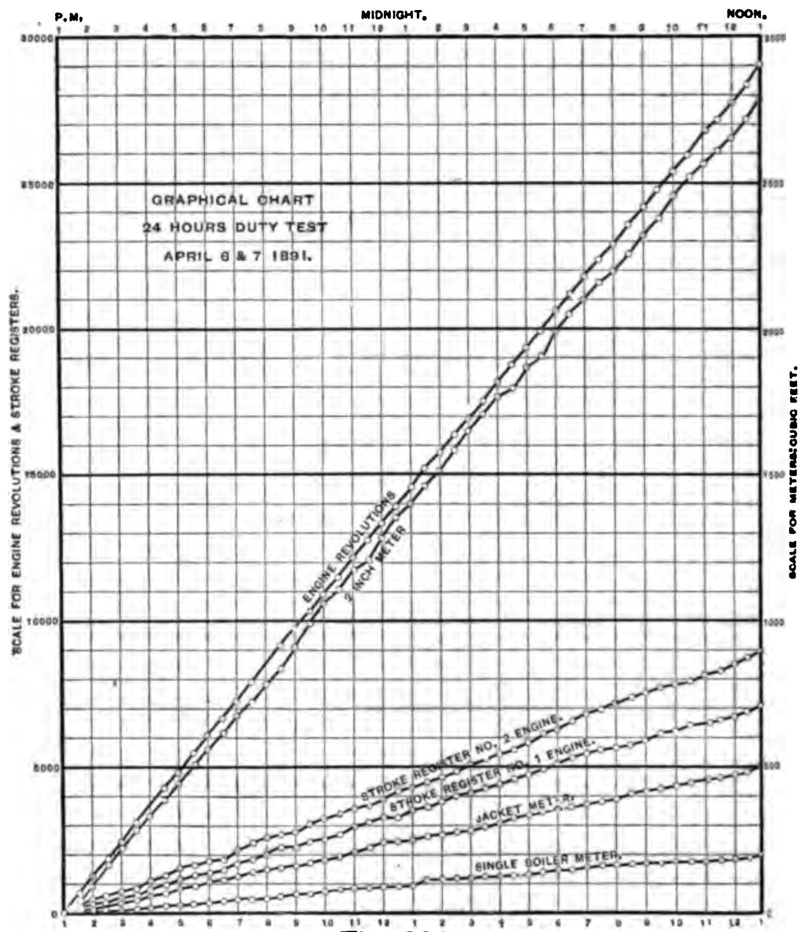


Fig. 298.

low piston, only aggregates 1%, and none of the values are rejectable as abnormally large or small, according to *Pierce's Criterion.* The mean effective oil pressure is subject to the same amount of error, due to variations of value, and the scale of the oil indicator is believed liable to an error of 1½%. Such errors may be either plus or minus, and may partly neutralize or magnify each other's effects in the main result. In the present instance, as the average steam-cards give a very low value of friction, it has been judged proper to consider the true steam horse-power as that due to the arithmetical mean of the steam mean effective pressures, increased by 1%. Also, since the value used for the scale of the hydraulic indicator, 610 lbs. per inch, is probably on the low side of the truth, the true pump-power is taken as that due to exactly the arithmetical mean of the mean effective pressures of the pump cylinders.

Fourth.—The friction of the mechanism, item 33, is considerably less than has been found in measurements of inside plunger Worthington pumps, and, hence, the pumps, in this instance having two outside packed stuffing-boxes on each side, might naturally be expected to waste more power in friction than an outside plunger working freely in the midst of water. But the plunger and oscillators in the case of pumping oil lubricate themselves so completely that, when the packing is so nicely and permanently adjusted as in the present case, the friction of the several plungers may be very little. A computation of the power required to operate the air pumps, assuming their mean effective pressure to be $6\frac{1}{2}$ pounds, as usually shown by air-pump cards, gives three-fourths of 1% of the steam horse-power.

A computation of the friction of the several moving parts of the engine, based upon allowances which have been found to give correct results in similar estimates,* indicates that their aggregate friction might easily be only 3% of the steam-power. Finally, by equalizing the pressure on both sides of the pumps, by means of a by-pass, and taking the mean pressure which, exerted on the high piston only, will just move the pump, such pressure corresponds to less than 5% of the steam-power. The friction of the pump plungers under these circumstances is greater than while running, as the lubrication is practically destroyed. It is, therefore, concluded† that the 3.6 per cent. of friction found is an entirely reasonable amount.

Fifth.—The moisture in the steam, item 41, is the average of several determinations made at different times when the calorim-

* See "Computation of Friction," of Pawtucket pumping engine and other motors, *Stevens Indicator*, Vol. VII., Nos. 1 and 2.

† A study of the possibility of the variations of pressure of the oil, during a stroke, causing the oscillating cylinders to act as a slight back pressure on the pumps, results in the conclusion that such action cannot exert such an effect of more than three-fourths of 1%, if any whatever.

eter was observed for an interval of about an hour and a half. The variation of the different tests was only a fraction of 1%.

The moisture given, 1.1%, is fairly attributable to the radiation, etc., of the steam supply pipes, which, while well covered, aggregated 100 feet of 6-inch, and 125 feet of 4-inch pipe.

DETERMINATION OF SLIP OF PUMPS.

On April 23d the pump drew oil steadily for six and one-half hours from one of the receiving tanks, which, isolated for several days, showed no leakage or gain of contents.

The contents of the tank for each foot of depth is known within one-twentieth of 1%, and, combined with a record of the stroke register and revolutions of the engine, afforded an unusual opportunity to determine the "slip" of the pump. The results obtained are as follows:

Revolutions per minute for 6½ hours	22.05
Average length of stroke, No. 1 engine	37.42 in.
" " " " " 2 "	37.91 "
" " " both strokes	3.14 ft.
Aggregate area of four plungers	1.867 sq. ft.
Cubic feet displaced by plungers, per hour	7758.07
Depth of tank "pump out" in 6½ hours	7.896 ft.
Cubic contents of each foot of tank in barrels of 42 gallons each, 231 cu. in. per gallon	1121.9
Cubic feet "pumped out" per hour	7652.44
Slip in cubic feet per hour, 7758.07—7652.44 =	105.63
" " per cent. of plunger displacement	1.36

As the leakage from the plungers was comparatively infinitesimal, this slip is due wholly to valve action.

BEHAVIOR OF ENGINE WITHOUT STEAM IN JACKETS AND REHEATER.

On April 4th steam was shut off from the jackets and reheater, and the latter emptied into the hot well. The engine was then run one hour and a half. The stroke was very irregular. If adjusted to run 2 inches short of striking the heads, a piston would often strike for a few strokes and then the stroke again shorten.

The revolutions dropped from 20.13 to 16.6 per minute, and the oil pressure from 850 to 750 lbs. The indicator cards were very irregular from water in the cylinders, but occasionally a

Cards taken fifteen minutes after the steam was shut off from jackets.

Full lines are cards taken five minutes before shutting off steam from jackets. Dotted lines are cards taken fifteen minutes after shutting off steam from jackets.

Pump under excellent control and making full stroke.

High-pressure cylinder, No. 2 engine.
Scale of spring: 49.5
Mean effective pressures:
Right...... 29.10
Left....... 34.02

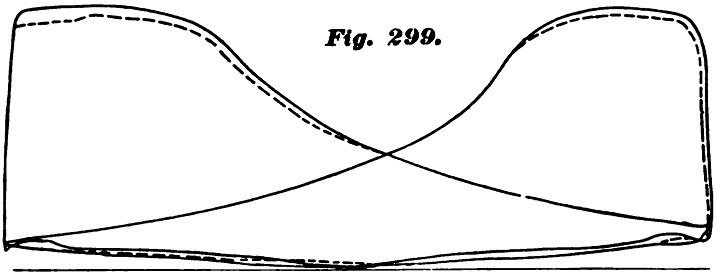


Fig. 299.

Low-pressure cylinder, No. 2 engine.
Scale of spring: 10.1
Mean effective pressures:
Right....... 8.38
Left........ 8.18

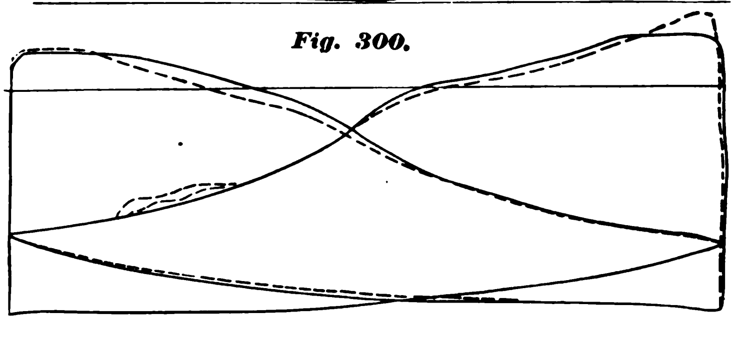


Fig. 300.

High-pressure cylinder, No. 1 engine.
Scale of spring: 49.5
Mean effective pressures:
Right....... 28.6
Left......... 33.9

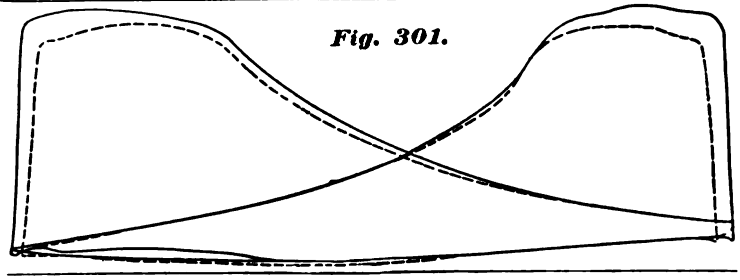


Fig. 301.

Low-pressure cylinder, No. 1 engine.
Scale of spring: 10.1
Mean effective pressures:
Right....... 8.28
Left........ 7.07

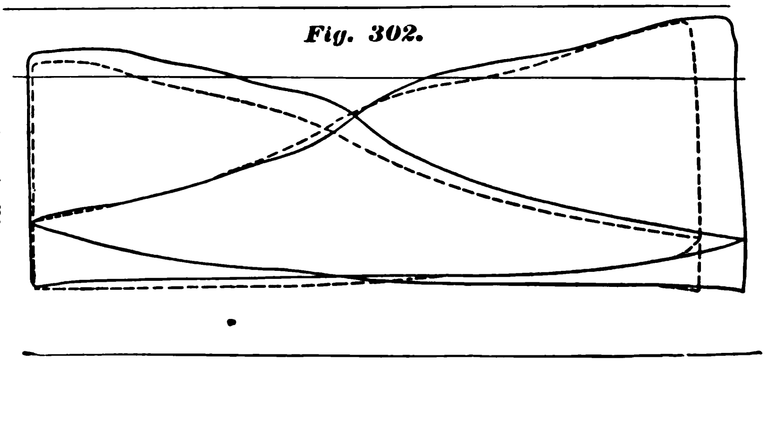


Fig. 302.

Cards taken twenty-five minutes after shutting off steam from jackets.

Dotted lines represent variations in the card while the pencil was in contact with the paper one minute.

Pump under excellent control and making full stroke.

High-pressure cylinder, No. 2 engine.
Scale of spring : 49.5
Mean effective pressures :
Right....... 27.6
Left........ 33.9

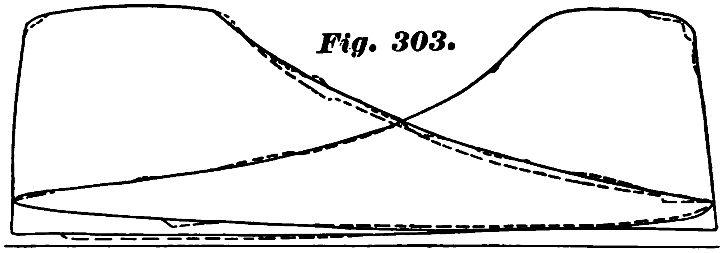


Fig. 303.

Low-pressure cylinder, No. 2 engine.
Scale of spring : 10.1
Mean effective pressures :
Right....... 7.27
Left........ 6.87

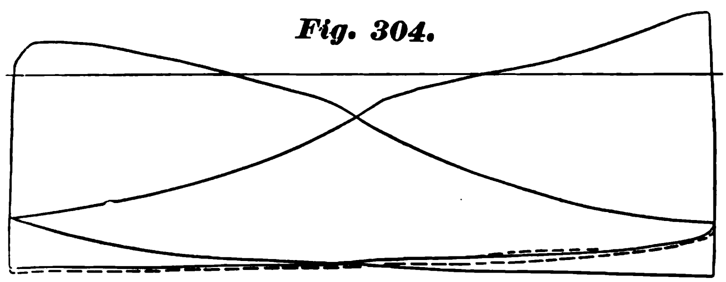


Fig. 304.

High-pressure cylinder, No. 1 engine.
Scale of spring : 49.5
Mean effective pressures :
Right....... 29.1
Left........ 35.0

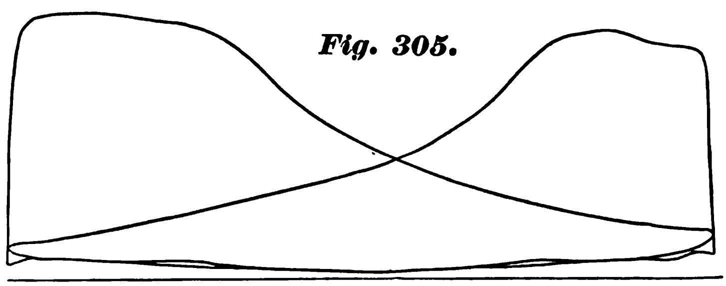


Fig. 305.

Low-pressure cylinder, No. 1 engine.
Scale of spring : 10.1
Mean effective pressures :
Right....... 7.57
Left........ 6.77

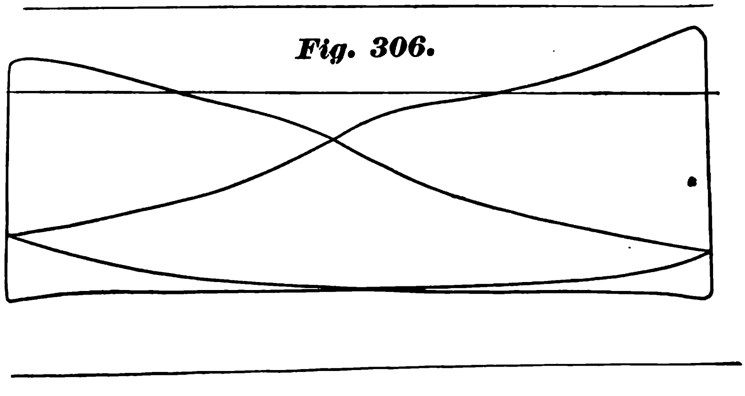


Fig. 306.

Cards taken forty-five minutes after shutting off steam from jackets.

Dotted lines represent the variations in the card while the pencil was in contact with the paper one minute.

Pump under excellent control and making full stroke.

Fig. 307.

High-pressure cylinder, No. 2 engine.
Scale of spring : 49.5
Mean effective pressures :
Right....... 29.6
Left........ 36.5

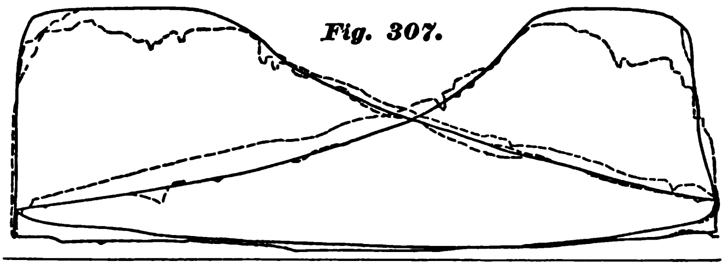


Fig. 308.

Low-pressure cylinder, No. 2 engine.
Scale of spring : 10.1
Mean effective pressures :
Right...... 6.21
Left........ 7.17

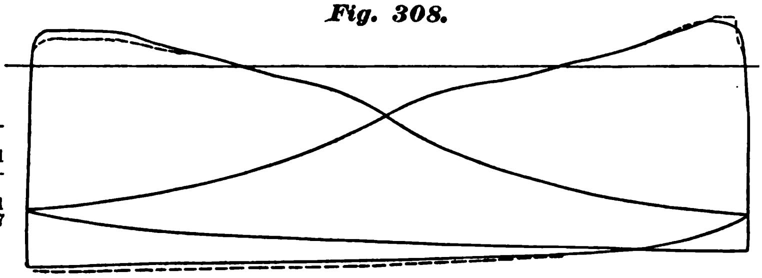


Fig. 309.

High-pressure cylinder, No. 1 engine.
Scale of spring : 49.5
Mean effective pressures :
Right....... 31.8
Left........ 37.5

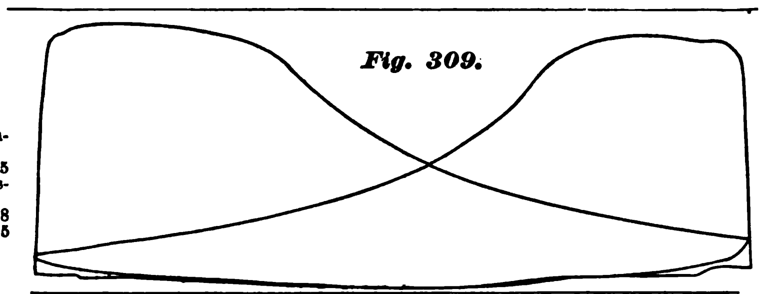
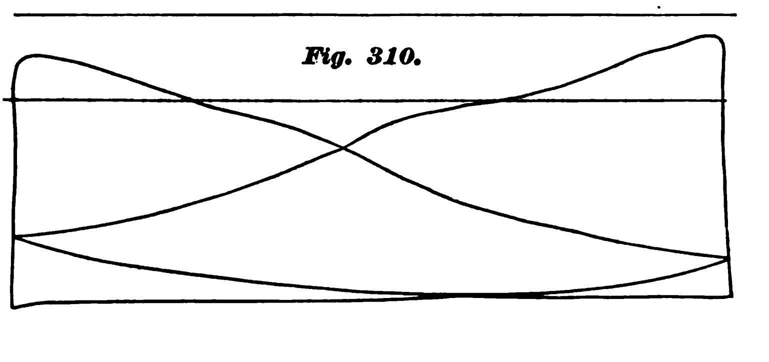


Fig. 310.

Low-pressure cylinder, No. 1 engine.
Scale of spring : 10.1
Mean effective pressures :
Right....... 7.37
Left........ 7.07



well-formed card could be obtained. The water consumption per horse-power was increased 20%, but, as less work was being done, the coal consumption did not noticeably increase.

On April 23d steam was again shut off the jackets and reheater, and, from the experience of the previous experiment, the engine was more quickly and nicely adjusted to the new conditions.

The stroke was easily maintained fairly regular for an hour, and the indicator cards, Figs. 299 to 310, obtained. The speed fell from 22.05 to 20.2 per minute, and the oil pressure a corresponding amount. No means of measuring the water consumption was then available.

The coal consumption did not noticeably change, but it is too coarse a means of measurement to be of any value in the case. After seven minutes of running, the water in one low cylinder could be heard to splash as the piston reached the end of its stroke. The vacuum fell from 27¼ to 23 inches. After running an hour, it was evident that the water in the cylinders was still increasing, although the pump was still under good control. Five minutes after restoring the steam to the jackets, the full vacuum had returned, the sound of the water in the cylinders disappeared, and the engine regained her speed. The indicator cards show about 20% less mean effective pressure in the low cylinders, and about equal mean effective pressure in the high cylinders, compared with the cards taken with steam in the jackets and reheater. It was evident that, letting alone the question of economy of steam, steam-jackets are essential to the successful working of this style of engine.

CONCLUSIONS.

First.—The boiler economy of the station is equivalent to 9.1 and 9.8 lbs. evaporation per pound of pea and stove anthracite respectively, from the actual temperature of feed, 147° Fahr., and average steam pressure of 90 lbs. above the atmosphere. It is probable that an evaporation greater than 10 lbs. per pound of coal, under working conditions, could easily be obtained with the best bituminous coal.

Second.—The average economy of the engine is represented by any of the following statements:

[*a*] The steam consumed per hour per horse-power pump end is.............................. 17.0 lbs.

[b] The steam consumed per hour per horse-power steam end is	16.4 lbs.
[c] The duty per 100 lbs. of coal, if the boiler supplies to the engine 10 lbs. of steam per pound of coal burned, is	116.5 mill. ft. lbs.
[d] The duty, per each million thermal units of heat imparted to the feed-water by the boilers neglecting steam used by feed-pumps, is........	109.5 " " "
[e] Ditto, including steam used by pump, if latter exhausts into feed-water	109.3 " " "

Third.—The engine works smoothly while pumping against upward of 900 lbs. pressure per square inch. Under fluctuations of steam pressure amounting to 5%, and of oil pressure amounting to 10%, the engine maintains an average length of stroke within three-thirty-seconds of an inch of a maximum possible stroke of 37.85 inches, or within 0.33%. The greatest variation of individual strokes was nine-thirty-seconds of an inch, or 1.0%.

Fourth.—The steam condensed in the jackets and reheater, in per cent. of the total steam supplied to engine, is 15.8%.

Fifth.—The power wasted in friction of the mechanism, and operating the two air pumps, in per cent. of the horse-power of the steam end, is 3.6%.

Sixth.—The actual delivery of oil was within 1.36% of the cubical displacement of the pump plungers, or, the "slip" was 1.36%.

Seventh.—Without the use of steam in the jackets and reheater, water accumulates in the steam cylinders, in an hour's time, to such an extent as to be heard to splash against the pistons, while the loss of mean effective power, principally in the low cylinders, causes at least 10% loss of speed, which is probably attended with a considerable loss of economy, but to what extent was not determined. The use of steam-jackets is undoubtedly necessary to the successful operation of a steam-engine of this particular type.

APPENDIX.

MEAN EFFECTIVE PRESSURE OF STEAM CARDS. TEST OF APRIL 6, 1891.

No.	Low Cylinder. No. 1 Engine, Steam End.			Low Cylinder. No. 1 Engine, Pump End.			High Cylinder. No. 1 Engine, Steam End.			High Cylinder. No. 1 Engine, Pump End.		
	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.
1	9.39	0.50	0.2500	9.30	0.12	0.0144	35.72	0.57	0.3249	33.00	0.73	0.5329
2	9.02	0.13	0.0169	9.36	0.18	0.0324	35.90	0.39	0.1521	32.90	0.63	0.3969
3	8.14	0.75	0.5625	8.48	0.70	0.4900	35.60	0.69	0.4761	32.65	0.38	0.1444
4	9.09	0.20	0.0400	8.30	0.88	0.7744	35.40	0.89	0.7921	31.05	1.22	1.4884
5	8.43	0.46	0.2116	8.41	0.77	0.5929	35.95	0.34	0.1156	32.20	0.07	0.0049
6	8.68	0.21	0.0441	8.36	0.82	0.6724	37.35	1.06	1.1236	31.30	0.97	0.9409
7	8.41	0.48	0.2304	8.30	0.88	0.7744	35.00	1.29	1.6641	33.35	1.08	1.1664
8	8.19	0.70	0.4900	8.81	0.37	0.1369	37.45	1.16	1.3456	33.85	1.58	2.4964
9	8.65	0.24	0.0576	8.62	0.56	0.3136	35.40	0.89	0.7921	32.65	0.38	0.1444
10	7.93	0.96	0.9216	8.52	0.66	0.4356	36.04	0.25	0.0625	31.90	0.37	0.1369
11	8.12	0.77	0.5929	8.26	0.92	0.8464	37.40	1.11	1.2321	31.90	0.37	0.1369
12	7.87	1.02	1.0404	8.53	0.65	0.4225	37.30	1.01	1.0201	31.50	0.77	0.5929
13	9.76	0.87	0.7569	9.69	0.51	0.2601	35.60	0.69	0.4761	32.70	0.43	0.1849
14	8.48	0.41	0.1681	8.93	0.25	0.0625	36.30	0.01	0.0001	31.40	0.87	0.7569
15	8.25	0.64	0.4096	8.72	0.46	0.2116	38.50	2.21	4.8841	31.15	1.12	1.2544
16	8.33	0.56	0.3136	8.35	0.83	0.6889	37.45	1.16	1.3456	31.25	1.02	1.0404
17	8.45	0.44	0.1936	9.00	0.18	0.0324	37.75	1.46	2.1316	31.65	0.62	0.3844
18	8.41	0.48	0.2304	8.05	1.13	1.2769	37.10	0.81	0.6561	31.10	1.17	1.3689
19	8.11	0.78	0.6084	8.42	0.76	0.5776	37.40	1.11	1.2321	31.90	0.37	0.1369
20	8.01	0.88	0.7744	8.33	0.85	0.7225	36.60	0.31	0.0961	32.80	0.53	0.2809
21	8.05	0.84	0.7056	8.08	1.10	1.2100	36.35	0.06	0.0036	32.30	0.03	0.0009
22	8.11	0.78	0.6084	8.28	0.90	0.8100	36.65	0.36	0.1296	30.20	2.07	4.2849

23	7.71	1.18	1.3924	8.34	0.84	0.7056	36.10	0.19	0.0361	32.40	0.13	0.0169
24	8.21	0.68	0.4624	8.51	0.67	0.4489	37.00	0.71	0.5041	32.45	0.18	0.0324
25	9.67	0.78	0.6084	10.09	0.91	0.8281	37.60	1.31	1.7161	33.00	0.73	0.5329
26	7.56	1.33	1.7689	8.60	0.58	0.3364	37.50	1.21	1.4641	32.30	0.03	0.0009
27	8.40	0.49	0.2401	8.75	0.43	0.1849	37.35	1.06	1.1236	32.10	0.17	0.0289
28	9.83	0.94	0.8836	9.78	0.60	0.3600	36.70	0.41	0.1681	32.65	0.38	0.1444
29	9.64	0.75	0.5625	10.13	0.95	0.9025	35.80	0.49	0.2401	32.80	0.53	0.2809
30	10.09	1.20	1.4400	10.40	0.22	0.0484	36.30	0.01	0.0001	33.10	0.83	0.6889
31	9.56	0.67	0.4489	10.13	0.95	0.9025	35.90	0.29	0.0841	32.50	0.23	0.0529
32	9.38	0.49	0.2401	9.50	0.32	0.1024	35.70	0.59	0.3481	32.70	0.43	0.1849
33	9.58	0.69	0.4761	9.99	0.81	0.6561	36.30	0.01	0.0001	32.65	0.38	0.1444
34	9.63	0.74	0.5476	10.01	0.83	0.6889	36.60	0.31	0.0961	32.65	0.38	0.1444
35	9.95	1.06	1.1236	10.03	0.85	0.7225	36.65	0.36	0.1296	32.80	0.53	0.2809
36	9.45	0.56	0.3136	9.82	0.64	0.4096	36.00	0.29	0.0841	32.20	0.07	0.0049
37	10.05	1.16	1.3456	10.02	0.84	0.7056	36.85	0.56	0.3136	32.85	0.58	0.3364
38	9.63	0.74	0.5476	9.92	0.74	0.5476	35.25	1.04	1.0816	32.25	0.02	0.0004
39	9.23	0.34	0.1156	9.75	0.57	0.3249	35.75	0.54	0.2916	32.30	0.03	0.0009
40	9.67	0.78	0.6084	9.86	0.68	0.4624	35.65	0.64	0.4096	32.40	0.13	0.0169
41	9.72	0.83	0.6889	10.27	1.09	1.1881	35.85	0.44	0.1936	32.85	0.58	0.3364
42	9.35	0.46	0.2116	9.14	0.04	0.0016	35.35	0.94	0.8836	32.40	0.13	0.0169
43	9.83	0.94	0.8836	10.10	0.92	0.8464	35.40	0.89	0.7921	32.40	0.13	0.0169
44	9.71	0.82	0.6724	10.00	0.82	0.6724	35.75	0.54	0.2916	32.90	0.63	0.3969
45	9.65	0.76	0.5776	10.05	0.87	0.7569	34.90	1.39	1.9321	32.45	0.18	0.0324
46	8.73	0.16	0.0256	9.25	0.07	0.0049	35.65	0.64	0.4096	31.65	0.62	0.3844
47	9.72	0.83	0.6889	10.17	0.99	0.9801	35.50	0.79	0.6241	31.85	0.92	0.8464
48	8.67	0.22	0.0484	9.31	0.13	0.0169	35.75	0.54	0.2916	31.85	0.42	0.1764
49	8.44	0.45	0.2025	9.29	0.11	0.0121	35.75	0.54	0.2916	32.20	0.07	0.0049
50	8.15	0.74	0.5476	9.11	0.07	0.0049	35.60	0.69	0.4761	32.40	0.13	0.0169
51	8.18	0.71	0.5041	8.60	0.58	0.3364						
Aver....	8.89	Sum of sq. 27.4035		Av. 9.19	Sum of sq. 25.5165		Av. 36.29	Sum of sq. 34.3327		Av. 32.27	Sum of sq. 22.9975	
0.079 probable error of average.				0.067 probable error of average.			0.075 probable error of average.			0.068 probable error of average.		

MEAN EFFECTIVE PRESSURE OF STEAM CARDS. TEST OF APRIL 6, 1891.

No.	Low Cylinder. No. 2 Engine, Steam End.			Low Cylinder. No. 2 Engine, Pump End.			High Cylinder. No. 2 Engine, Steam End.			High Cylinder. No. 2 Engine, Pump End.		
	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.	M. E. P. lbs. per sq. in.	Residuals.	Squares of residuals.
1	9.00	0.11	0.0121	9.69	0.55	0.3025	37.60	1.28	1.6384	32.40	0.96	0.9216
2	9.12	0.23	0.0529	8.76	0.33	0.1089	38.25	1.93	3.7249	32.45	1.01	1.0201
3	9.75	0.86	0.7396	10.38	1.29	1.6641	38.30	1.98	3.9204	31.90	0.46	0.2116
4	7.70	1.19	1.4161	8.33	0.76	0.5776	39.10	2.78	7.7284	33.60	2.16	4.6656
5	8.21	0.68	0.4624	8.35	0.74	0.5476	37.55	1.23	1.5129	31.50	0.06	0.0036
6	8.17	0.72	0.5184	8.27	0.82	0.6724	38.40	2.08	4.3264	32.10	0.66	0.4356
7	8.27	0.62	0.3844	8.60	0.49	0.2401	37.60	1.28	1.6384	33.20	1.76	3.0976
8	8.03	0.86	0.7396	8.10	0.99	0.9801	37.50	1.18	1.3924	33.80	2.36	5.5696
9	7.97	0.92	0.8464	7.90	1.19	1.4161	36.70	0.38	0.1444	31.25	0.19	0.0361
10	8.29	0.60	0.3600	8.21	0.88	0.7744	36.90	0.58	0.3364	30.90	0.54	0.2916
11	8.28	0.61	0.3721	8.41	0.68	0.4624	37.25	0.93	0.8649	31.80	0.36	0.1296
12	9.73	0.84	0.7056	9.87	0.76	0.5776	37.10	0.78	0.6084	30.10	1.34	1.7956
13	8.76	0.13	0.0169	8.70	0.39	0.1521	35.50	0.82	0.6724	31.00	0.44	0.1936
14	7.94	0.95	0.9025	8.48	0.61	0.3721	35.80	0.52	0.2704	31.80	0.36	0.1296
15	8.09	0.80	0.6400	8.64	0.45	0.2025	35.60	0.72	0.5184	32.00	0.56	0.3136
16	8.19	0.70	0.4900	8.62	0.47	0.2209	37.40	1.08	1.1664	31.15	0.29	0.0841
17	8.08	0.81	0.6561	8.27	0.82	0.6724	36.40	0.08	0.0064	31.45	0.01	0.0001
18	8.39	0.50	0.2500	8.67	0.42	0.1764	36.10	0.22	0.0484	30.60	0.84	0.7056
19	8.03	0.86	0.7396	8.45	0.64	0.4096	36.30	0.02	0.0004	31.20	0.24	0.0576
20	7.74	1.15	1.3225	7.93	1.16	1.3456	36.30	0.02	0.0004	31.20	0.24	0.0576
21	8.36	0.53	0.2809	8.24	0.85	0.7225	36.65	0.33	0.1089	31.05	0.39	0.1521
22	7.89	1.00	1.0000	8.51	0.58	0.3364	36.70	0.38	0.1444	31.75	0.31	0.0961
23	8.20	0.69	0.4761	8.41	0.68	0.4624	36.90	0.58	0.3364	30.75	0.69	0.4761
24	8.33	0.56	0.3136	8.49	0.60	0.3600	36.45	0.13	0.0169	31.00	0.44	0.1936
25	8.85	0.04	0.0016	9.66	0.57	0 3249	36.80	0.48	0.2304	31.60	0.16	0.0256

26	9.64	0.75	0.5625	9.83	0.74	0.5476	38.15	1.83	3.3489	33.60	2.16	4.6656
27	9.49	0.60	0.3600	9.56	0.47	0.2209	36.00	0.32	0.1024	30.50	0.94	0.8836
28	9.54	0.65	0.4225	9.62	0.53	0.2809	35.50	0.82	0.6724	31.30	0.14	0.0196
29	9.73	0.84	0.7056	9.78	0.69	0.4761	35.40	0.92	0.8464	30.60	0.84	0.7056
30	9.73	0.84	0.7056	9.70	0.61	0.3721	35.30	1.02	1.0404	31.10	0.34	0.1156
31	9.82	0.93	0.8649	9.91	0.82	0.6724	35.10	1.22	1.4884	30.80	0.64	0.4096
32	9.56	0.67	0.4489	9.62	0.53	0.2809	35.05	1.27	1.6129	31.05	0.39	0.1521
33	9.26	0.37	0.1369	9.37	0.28	0.0784	34.75	1.57	2.4649	31.35	0.09	0.0081
34	9.61	0.72	0.5184	9.65	0.56	0.3136	35.45	0.87	0.7569	31.35	0.09	0.0081
35	8.31	0.58	0.3364	8.81	0.28	0.0784	35.50	0.82	0.6724	31.35	0.09	0.0081
36	9.55	0.66	0.4356	9.56	0.47	0.2209	35.45	0.87	0.7569	31.30	0.14	0.0196
37	9.53	0.64	0.4096	9.47	0.38	0.1444	34.30	2.08	4.3264	31.40	0.04	0.0016
38	9.10	0.21	0.0441	9.50	0.41	0.1681	34.75	1.57	2.4649	30.25	1.19	1.4161
39	9.72	0.83	0.6889	9.71	0.62	0.3844	35.40	0.92	0.8464	30.85	0.59	0.3481
40	9.90	1.01	1.0201	10.10	1.01	1.2001	37.00	0.68	0.4624	30.65	0.79	0.6241
41	10.00	1.11	1.2321	9.98	0.89	0.7921	35.60	0.78	0.6084	31.55	0.11	0.0121
42	10.01	1.12	1.2544	9.90	0.81	0.6561	35.65	0.67	0.4489	32.35	0.91	0.8281
43	9.80	0.91	0.8281	10.13	1.04	1.0816	34.70	1.62	2.6244	31.40	0.04	0.0016
44	9.26	0.37	0.1369	9.17	0.08	0.0064	35.90	0.42	0.1764	31.60	0.16	0.0256
45	9.11	0.22	0.0484	9.63	0.54	0.2916	34.75	1.57	2.4649	30.85	0.59	0.3481
46	9.50	0.61	0.3721	9.68	0.59	0.3481	35.90	0.42	0.1764	30.65	0.79	0.6241
47	....	....	....	8.80	0.29	0.0841	36.15	0.17	0.0289	31.10	0.34	0.1156
48	....	....	....	....	....	....	35.15	1.17	1.3689	30.75	0.69	0.4761
49	....	....	....	....	....	....	36.45	0.13	0.0169	31.65	0.21	0.0441
50	....	....	....	....	....	....	36.70	0.38	0.1444	31.15	0.29	0.0841
51	....	....	....	....	....	....	36.40	0.08	0.0064	30.75	0.69	0.4761
52	....	....	....	....	....	....	36.55	0.23	0.0529	31.10	0.84	0.1156
Aver....8.89	Sum of sq., 25.2214			Av. 9.09	Sum of sq., 22.6007		Av. 36.32	Sum of sq., 61.3263		Av. 31.44	Sum of sq., 38.2012	
0.074 probable error of average.				0.068 probable error of average.			0.102 probable error of average.			0.075 probable error of average.		

$$\left.\begin{matrix}\text{Aggregate error re-}\\ \text{duced to area of low}\\ \text{cylinder.}\end{matrix}\right\} = \sqrt{\left(\overline{0.079}^2 + \overline{0.067}^2 + \overline{0.074}^2 + \overline{0.068}^2 + \frac{\overline{0.075}^2 + \overline{0.068}^2 + \overline{0.102}^2 + \overline{0.075}^2}{[4.097]^2}\right)} = \left\{\begin{matrix}0.17 \text{ lbs. per sq. in., or}\\ 1\% \text{ of steam H. P.}\end{matrix}\right.$$

DISCUSSION.

Prof. R. H. Thurston.—The builders of the Worthington engine are to be most heartily congratulated on this very admirable and remarkable result of their recent improvements. It is an especially interesting case, as exhibiting the peculiar conditions of oil-pumping as well as of this form of engine. In the earlier days of the Worthington engine we considered a duty of 70,000,000 both high and satisfactory, and the makers were satisfied to base their claims in the market upon the low money-cost of the machine—a very good basis, too. But the simple and beautiful device of Mr. Charles C. Worthington has put the machine among the "high duty" engines, and it is a great pleasure to his friends, and the friends of the senior Worthington—the inventor of the original type—to see the claims for the new form so handsomely sustained. Were our old friend to-day living he would, I am sure, tell us that nothing outside the walls of his own home could give him greater pleasure than this success of his son. A duty of over 116,000,000 from a direct-acting engine is without precedent.

In seeking the source of this economy we, of course, at once note the exceptional head which, by reducing the proportion of loss at the valves and in the passages of the pump, gives, other things equal, some advantage in duty, and the low friction of pumps consequent upon their friction-producing parts working in an oil-bath. The low figure for slip must be attributed to excellence of design and construction. But setting aside the fact of a friction of pump falling inside 4%, a slip less than 1½%, and a head equivalent to one-third of a mile, it is still evident that the steam-end and its recent improvements are to be credited with the main portion of the gain in economy lately made and here illustrated. The equalizing mechanism permits the practical employment of a ratio of expansion equal to that adopted in the best forms of modern high duty engines of other types. Its perfection is perhaps best exhibited by the uniformity of stroke-length shown so prettily by Prof. Webb's ingenious and simple register; while the delicacy of the balance sustained throughout the stroke is shown by the behavior of the engine with and without jacket. I observe that the temperature of the hot-well is kept well down, presumably to secure a good vacuum; but it would be interesting to ascertain whether, on the whole, a warmer condenser and hotter feed would not give still better economy. With the usual construction, involving rubber valves in the air pump, it is often impracticable to carry the condenser very warm; but I am strongly of the opinion that the temperature of best effect, especially with high pressures of steam, would be found above that commonly adopted, in many cases at least.

I am glad to get some facts relative to the behavior of this engine with and without its jacket. It has been a perfectly well-known fact for a long time that the jacket of the Worthington engine had a special office aside from economic, in giving a better sustained pressure toward the end of the stroke; but the makers themselves have not been sure that it was otherwise profitable. I think that question is settled by direct experiment, though probably not very doubtful before. Sustained pressure means less waste by internal condensation, and economy would seem to be inevitably the consequence. We here find that 15% of the steam does its work through the jacket, producing a gain of 20%, at the same time keeping up the terminal pressure, giving smoother and better engine speed, and reducing the annoyances and dangers of water in the cylinders. Hirn's phenomenon, the gradual modification of the action of the internal walls of the cylinder, as the jackets are thrown on or off, shows in turn, as it seems to me, the truth of Rankine's description of the process of initiation of this action in part at least. It is a striking way of stating the effect of jacketing here to say that the steam used by way of the jacket saves its own value and a third more besides, while its duty is 75% higher than that used directly by way of the steam-ports. Evidently, the larger the proportion that the engine can be thus persuaded to accept in the ordinary case the better.

This is, to my mind, one of the most interesting and instructive cases which has ever been reported to the Society, and all concerned should be most heartily and cordially congratulated.

Mr. Fk. Meriam Wheeler.—This paper may be considered as probably one of the most attractive of its kind that has ever been presented, and is of special import to many of the pump builders present, and those interested in moving large bodies of liquids under heavy pressures. Prof. Denton, with his usual care and thoroughness, has certainly given the Society a very complete paper. There are one or two points, however, about which I wish to speak. First, the remarkable fact that he got so steady and so great a stroke At the Richmond meeting, in discussing the subject of the testing of pumping engines, this matter of stroke came up, and I made a few points at the time and quoted several pumping engines which I had seen in operation of the direct-acting type where the strokes were quite variable. The point I made was that *each* and *every* stroke of the pumping engine should be recorded during the test by some simple instrument designed especially for the purpose. Taking a rule measure and laying out the stroke, and every ten, fifteen, or twenty minutes taking a reading, is not the proper way to get the correct average length of stroke of a direct-acting pumping engine. I am therefore very glad to see a proper "length of stroke recorder," as the one brought out so promptly and used on this test, and would like to ask Prof. Denton if it was designed by himself or a device produced by his clients. Of course, we all know that ordinary hydraulic-pressure pumps work up to pressures greater than what this oil-line engine pumped against; still, it will strike the average layman with surprise to learn of a pump of this kind working day and night against a pressure equal to a head of about 2,000 feet of water. I am surprised at the small amount of "slippage" shown, as under heavy pressures, with a material like petroleum, which carries with it a great deal of sediment and sand, it is very hard on the valves and plungers. The great trouble is to keep the efficiency of the pumping engine up in that respect, as the material pumped is so hard on the valves, etc. Another remarkable thing about this engine is its high steam economy, considering the very low rate of piston travel. I cannot find readily in the paper where the speed is stated, and would like to ask Prof. Denton what it was.

Mr. Geo. H. Barrus.—I wish to add a word to the discussion, complimenting Prof. Denton on the character of the report of this test, which is most admirable in every way. Prof. Denton gives an object lesson at the same time that he describes his test, by the full illustrations of the locality where the engine was placed, the various views of the engine itself, and much of the apparatus used in conducting the tests. He has done the same thing in much of the other work which he has presented to the Society, and he should be thanked for adding to his reports so much, in the way of elucidation, which pictures furnish. The report gives full description and dimensions of the engine, together with various tables and diagrams relating to the work of the test, all of which give a complete idea of the performance. Among other points of interest which can be specially commended, may be mentioned the combined diagrams, which are carefully worked out; the graphical charts, which show the full log of the records taken during the tests; and, finally, a summary, which closes the paper, giving the gist of the results. Prof. Denton should also be thanked for the work which he did in integrating the lengths of the strokes by means of the instrument designed by Prof. Webb, which is of special interest at this time on account of the work just completed by the Duty Trial Committee. In looking over the table of results, it is gratifying to see that the duty is expressed on the basis of the recommendation of the committee referred to; that is, on the basis of 1,000,000 heat units consumed. Altogether, the test and its report is a model one, and I wish to congratulate the author on his work.

Mr. E. F. C. Davis.—Would Prof. Denton give us some idea how that accuracy of stroke was arranged for?

Prof. Denton.—There was no special arrangement about the pump to secure the remarkable constancy of stroke. It was attended by one man, who was not interfered with at all or instructed, and, so far as I could judge, it was the natural tendency of the pump to maintain its length of stroke as uniform as recorded. We worked about the station for upward of three weeks, so that I am certain it was under ordinary conditions.

In answer to Mr. Wheeler's inquiries:

The piston speed was 126 feet, and the integrating apparatus was designed by my colleague, Prof. Webb. Regarding the perfect action of the valves, resulting in such small slippage, I would say that the valve seats were of rubber, and the systematic record of the delivery enabled their leakage by wear to be maintained at a minimum by the renewal of the seats. I am under obligations for the courteous manner in which the paper has been discussed.

CCCCLXI.*

HEAT TRANSMISSION THROUGH CAST-IRON PLATES PICKLED IN ACIDS.

BY DANIEL ROYSE, ITHACA, N. Y.
(Junior Member of the Society.)

THE writer has recently made some experiments upon plates pickled in dilute nitric, hydrochloric, and sulphuric acids. Inasmuch as these experiments were a continuation of those made by

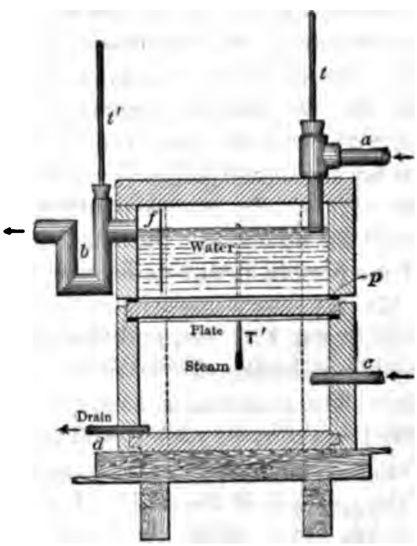


FIG. 225.

Prof. R. C. Carpenter,† the title given to this paper is the one under which he communicated his results to the Society.

The calorimeter used in making the comparisons is shown in section in the accompanying illustration (Fig. 225). The lower part is a pine box, 4.5 × 7.5 inches and 4.5 inches deep inside meas-

---

* Presented at the Providence meeting of the American Society of Mechanical Engineers (1891), and forming part of Volume XII. of the *Transactions.*

† Presented at XXIId Meeting, Richmond, November, 1890, page 174, Vol. XII.

urements, the plate through which the heat is to be transmitted forming the top. The upper portion consists of a frame or bottomless box, 4.5 × 7.5 inches inside and 3¾ inches deep, which is placed above the plate, and covered with a lid as shown.

The lower compartment is supplied with steam through a ¼-inch pipe, *c*, and is drained through the pipe *d*. In one side a thermometer, T', is inserted—making an angle of 45° with the vertical—so that the bulb is near the centre of the box. Above the plate, water enters through a pipe, *a*, placed in the lid, its temperature being taken by the thermometer *t*, and flows over the plate and out at the pipe *b*, the thermometer t' registering the temperature at which it leaves; *f* is a tin baffle-plate.

The method pursued was to turn on the steam and water and, after the temperatures as indicated by the thermometers became constant, note the time and weigh the water which flowed through in half an hour, reading the three thermometers at intervals of a minute. Owing to an imperfect mixing of the water the temperature of the outflow was not absolutely constant, the maximum variation being about 2° Fahr.

To compute the heat transmitted:

Let M represent the weight of water flowing through the calorimeter in pounds;

t represent its mean temperature at entrance;

t' " " " " " exit;

T' " mean temperature in steam chamber,

and a " area of plate exposed to steam below and water above, in square inches.

Then $Q = M(t' - t)$ is the heat transmitted in B.T.U.;

$T = \frac{t + t'}{2}$ is mean temperature in water chamber;

$q' = Q \div (T' - T)$ is the heat transmitted per degree difference of temperature,

and $q = K \times q'$ is the heat transmitted per square foot per hour per degree difference of temperature between the two fluids bounding the plate,

where $K = \frac{2}{a \div 144} = \frac{288}{a}$; since each run lasts ½ hour.

The value of a varied somewhat in different experiments, because the rubber packing was expanded laterally, thus reducing the effective area of the plate to a slight extent.

The temperature of the steam chamber was close to 212° Fahr.

during all the experiments, though the rate of flow of water and, hence, M, t', $(t' - t)$, $(T' - T)$, varied considerably with different runs; being, however, constant for any particular one.

$(T' - T)$ varied from 145° to 165° Fahr. in different experiments.

Experiments were made upon nineteen plates, six pickled by Prof. Carpenter, which were immersed in different acid solutions of various strengths for different lengths of time. Of these, ten were rough and nine planed on both sides. Two others planed on one side only were also used.

The depth to which any plate was attacked by the acid may be approximately computed by knowing its loss of weight and the extent of surface exposed to the acid. Each plate was 5.5 inches by 8.5 inches and the thickness was from one-quarter inch to one-half inch, according as whether planed or rough, and the superficial area of each was about 106 square inches. Since a cubic inch of cast iron weighs .26 lb., a loss of weight of .01 lb. means that it has been attacked to a depth of $\left(.01 \times \frac{1}{.26}\right) \div 106 = .00036''$.

All plates pickled by the writer were immersed in a vertical position, there being 11 lbs. of the solution to 318 square inches of surface of iron. From the atomic weights of the elements involved it was computed that in the 5% and 10% solutions only about one-third of the acid was consumed in attacking the iron. In all cases the effect of the treatment was to form a coating or scale, black in color, upon the surface of the plate. This, presumably carbon or, in the case of rough plates, a mixture of carbon and silicates, was quite adhesive, but easily removed with a knife. Plates pickled in nitric acid, when removed, were black in color, while those which had been treated with hydrochloric or sulphuric acid were quite rusty when removed.

The loss of weight of the plates when pickled in the latter acids was about twice as much as when a nitric solution of the same strength was used.

In Table I. are shown the results of all the experiments made by the writer, the qualities Q, q', q, loss of weight and depth attacked, and the character of the surface and method of treatment being given.

TABLE I.

No. of plate.	No. of experiment.	Heat transmitted, Q.	Heat transmitted per degree difference of temperature, q'.	Heat transmitted per sq. ft. per hour per degree difference of temperature, q.	Loss of weight in lbs.	Depth attacked in inches.	DESCRIPTION.
4	1	4218.2	29.0	263.1	......	....	Planed on one side only. Rough side exposed to steam.
	2	4194.9	29.1	263.5	......	....	
	3	4235.2	29.3	265.6	......	....	
	4	4289.5	28.3	256.7	......	....	
	5	4378.7	28.9	262.1	......	....	
	6	3924.6	28.8	261.1	......	....	
	7	3971.0	28.5	258.6	......	....	
	8	4566.8	28.9	261.9	......	....	
	9	4643.0	29.4	266.8	......	....	
	Mean	....	....	262.1	......	....	
	1	4632.2	31.3	283.8	......	....	Planed on one side only. Planed side exposed to steam.
	2	3461.0	26.8	242.9	......	....	
	3	4785.4	32.3	293.0	......	....	
	4	4299.7	30.8	279.5	......	....	
	5	4291.5	30.7	278.3	......	....	
	6	3784.5	29.2	265.2	......	....	
	7	4763.3	30.7	278.2	......	....	
	8	3931.6	28.0	253.8	......	....	
	Mean	....	....	270.7	......	....	
5	1	4485.7	28.6	259.5	......	....	Planed on one side. Rough side exposed to steam.
	1	4677.3	29.9	271.9	......	....	Planed on one side. Planed side exposed to steam.
X1	1	4041.0	26.1	236.6	......	....	Both sides planed.
	1	4222.8	26.6	226.6	.10	.0036	After being immersed in 10% nitric acid for 10 days.
	2	4004.4	25.5	217.9			
	3	4037.5	26.7	227.9			
	Mean	....	....	224.1			
X2	1	4114.0	26.8	242.8	......	....	Both sides planed.
	1	3901.0	24.5	209.3	.10	.0036	After being immersed in 10% nitric acid 20 days.
	2	4002.3	25.2	215.3			
	3	3954.4	25.3	215.7			
	Mean	....	....	213.4			
X3	1	4109.1	26.2	237.8	......	....	Both sides planed.
	1	4027.2	25.6	218.2	.13	.0047	After being immersed in 10% nitric acid 30 days.
	2	4083.7	25.9	221.3	......	....	
	Mean	....	....	219.8	......	....	
1	1	4986.5	31.2	283.0	......	....	Both sides planed.
	2	4551.5	30.0	273.3	......	....	
	3	4775.1	30.0	272.2	......	....	
	4	4649.2	30.0	271.9	......	....	
	5	5058.8	32.1	290.9	......	....	
	Mean	....	....	278.0	......	....	

TABLE I.—*Continued.*

No. of plate.	No. of experiment.	Heat transmitted, Q.	Heat transmitted per degree difference of temperature, q.	Heat transmitted per sq. ft. per hour per degree difference of temperature, q.	Loss of weight in lbs.	Depth attacked in inches.	DESCRIPTION.
	1	4472.7	28.5	243.5	.06	.0022	After being immersed in 5% nitric acid for 10 days.
	2	4421.7	28.0	240.6	......	....	
	3	4137.0	27.1	231.2	......	....	
	4	4278.2	28.2	240.3	......	....	
	Mean	....	....	238.9	......	....	
	1	4321.9	27.7	236.5	.13	.0047	After being immersed in 5% hydrochloric acid for 15 days.
	2	4330.9	27.8	237.0	......	....	
	3	4487.9	28.4	242.1	......	....	
	Mean	....	....	238.5	......	....	
2	1	4053.2	26.5	240.4	......	....	Both sides planed.
	1	3756.1	23.0	196.3	.06	.0022	After being immersed in 5% nitric acid 20 days.
	2	3889.1	23.8	203.6	......	....	
	3	3681.7	23.0	196.6	......	....	
	Mean	....	....	198.5	......	....	
	1	2500.3	15.3	130.5	......	....	One side varnished and dried 2 days. Varnished side exposed to steam.
	2	2413.2	14.8	126.2	......	....	
	Mean	....	....	128.4	......	....	
3	1	4339.4	28.3	256.2	......	....	Both sides planed.
	2	4513.7	29.5	267.3	......	....	
	Mean	....	....	261.8	......	....	
	1	3587.7	22.7	193.4	.08	.0029	After being immersed in 5% nitric acid for 30 days.
	2	3679.8	23.3	198.8	......	....	
	3	3694.8	24.0	204.9	......	....	
	Mean	....	....	199.0	......	....	
6	1	4315.5	27.2	247.2	......	....	Both sides planed.
	1	4344.6	28.2	240.3	.03	.0011	After being immersed in 1% nitric acid 10 days.
	2	4245.8	27.6	235.8	......	....	
	Mean	....	....	238.0	......	....	
	1	4561.9	28.4	242.5	.16	.0054	After being immersed in 5% sulphuric acid 15 days.
	2	4386.2	27.3	232.8	......	....	
	Mean	....	....	237.6	......	....	
7	1	3708.9	24.5	221.8	......	....	Both sides planed.
	2	3877.6	25.5	231.0	......	....	
	3	4107.8	26.8	243.5	......	....	
	Mean	....	....	232.1	......	....	
	1	4354.9	28.9	246.7	.03	.0011	After being immersed in 1% nitric acid 20 days.
	2	5106.8	31.6	268.6	......	....	
	3	5056.4	31.5	268.5	......	....	
	Mean	....	....	261.3	......	....	

TABLE I.—*Continued.*

No. of plate.	No. of experiment.	Heat transmitted, Q.	Heat transmitted per degree difference of temperature, q'.	Heat transmitted per sq. ft. per hour per degree difference of temperature, q.	Loss of weight in lbs.	Depth attacked in inches.	DESCRIPTION.
	1	1985.6	12.0	102.0	......	....	After being painted with coach varnish and dried 2 days.
	2	1791.9	11.0	94.2	......	....	
	3	1638.7	10.1	85.9	......	....	
	Mean	....	....	94.0	......	....	
8	1	3992.6	25.9	234.8	......	....	Planed on both sides.
	2	4116.8	26.6	241.3	......	....	
	3	4328.0	28.8	253.7	......	....	
	4	3977.9	25.5	231.2	......	....	
	Mean	....	....	240.3	......	....	
	1	5374.6	34.5	294.4	.03	.0011	After being immersed in 1% nitric acid 30 days.
	2	5175.1	33.2	283.5	......	....	
	Mean	....	....	289.0	......	....	
9	1	4160.0	26.7	242.4	......	....	Rough, as from foundry.
	2	4102.4	26.3	239.2	......	....	
	Mean	....	....	240.8	......	....	
	1	3968.6	24.5	209.0	.09	.0033	After being immersed in 10% nitric acid for 10 days.
	2	4011.5	24.6	210.0	......	....	
	Mean	....	....	209.5	......	....	
	1	1426.8	8.6	73.5	......	....	After being painted with coach varnish and dried 3 days.
	2	1476.8	8.9	76.3	......	....	
	3	1547.8	9.3	79.5	......	....	
	Mean	....	....	76.4	......	....	
	1	1479.9	8.9	75.7	......	....	After scraping off such of the varnish as was loosened by the exposure to steam at previous trial.
11	1	3547.2	22.4	203.3	......	....	Rough, as from foundry.
	2	3679.2	22.8	207.2	......	....	
	Mean	....	....	205.3	......	....	
	1	3284.8	20.1	182.6	.08	.0029	After being immersed in 10% nitric acid for 20 days.
	2	3170.7	19.4	176.3	......	....	
	3	2972.0	18.7	171.6	......	....	
	Mean	....	....	176.8	......	....	
	1	1625.3	9.9	84.2	......	....	After being painted with coach varnish and dried 3 days.
12	1	4098.4	25.5	231.2	......	....	Rough, as from foundry.
	2	4083.8	26.4	239.2	......	....	
	Mean	....	....	235.2	......	....	
	1	3408.8	21.7	185.0	.08	.0029	After being immersed in 10% nitric acid for 30 days.
	2	3414.5	21.7	185.5	......	....	
	3	3614.8	22.6	192.7	......	....	
	Mean	....	....	187.7	......	....	

TABLE I.—*Continued.*

No. of plate.	No. of experiment.	Heat transmitted, Q.	Heat transmitted per degree difference of temperature, q'.	Heat transmitted per sq. ft. per hour per degree difference of temperature, q.	Loss of weight in lbs.	Depth attacked in inches.	DESCRIPTION.
	1	1503.8	9.2	78.4	......	....	After being painted with coach varnish and dried 3 days.
E	1	4178.9	25.9	234.8	......	....	Rough, as from foundry. Pickled in 1% nitric acid 9 days.
	2	4183:5	26.6	240.9	......	....	
	3	4064.7	26.5	240.6	......	....	
	4	4389.8	27.1	246.2	......	....	
	Mean	....	....	240.6	......	....	
B	1	3315.5	21.6	223.0	......	....	Rough, as from foundry. Pickled in 1% nitric acid 18 days.
	2	3900.5	25 2	228.1	......	....	
	3	4257 5	26.5	240.2	......	....	
	4	4255.8	26.5	240.1	......	....	
	Mean	....	....	232.9	......	....	
F	1	3940.7	24.4	221.4	......	....	Rough, as from foundry. Pickled in 1% nitric acid 40 days.
	2	3753.3	23.8	215.8	......	....	
	3	3826.7	24·0	217.5	......	....	
	Mean	....	....	218.2	......	....	
N	1	4432.2	27.4	243.1	......	....	Rough, as from foundry. Pickled in 5% nitric acid 9 days.
	2	4604.5	28.5	252.9	......	....	
	3	4245.6	26.8	238.1	......	....	
	4	4270.7	27.6	244.9	......	....	
	Mean	....	....	244.6	......	....	
K	1	4042.0	25.1	223.1	......	....	Rough, as from foundry. Pickled in 5% nitric acid 18 days.
	2	3951.7	24.4	217.2	......	....	
	3	3751.7	24.3	215.9	......	....	
	4	3790.2	24.6	220.7	......	....	
	Mean	....	....	219.2	......	....	
P	1	3353.8	20.7	183.8	......	....	Rough, as from foundry. Pickled in 5% nitric acid 40 days.
	2	3617.9	22.5	199.1	......	....	
	3	3485.2	22.5	199.1	......	....	
	4	3621.2	23.3	206.4	......	....	
	Mean	....	....	197.1	......	....	
	1	3950.3	24.4	208.3	.11	.0040	After being immersed in 5% sulphuric acid 15 days.
	2	3979.6	24.6	209.8	......	....	
	Mean	....	....	209.1	......	....	
13	1	3884.9	24.0	218.1	......	....	Rough, as from foundry.
	2	3858.5	24.9	226.4	......	....	
	Mean	....	....	222.2	......	....	
	1	4058.8	24.8	211.3	......	....	After being immersed in 5% hydrochloric acid 15 days.
	2	4024.7	24.9	212.4	......	....	
	Mean	....	....	211.8	......	....	

It will be noted in the foregoing table that the heat transmitted through unpickled planed plates varies from 278.0 to 232.1 B.T.U. per square foot per hour per degree difference of temperature of the bounding fluids. The writer attributes this variation to some plates being rougher than others; to the rust which accumulated on the surface exposed to the water; and also to a different amount of the water of condensation clinging to the surface exposed to the steam during different experiments. These latter difficulties could not be obviated with the apparatus used. In computing the reduction which the treatment effected in the amount of heat transmitted, all planed plates will be compared with No. 1, which transmitted 278.0 B.T.U. Similarly, rough plates will be compared with No. 9, which transmitted 240.8 B.T.U.

It is important to mention that before plates *N*, *K*, *P*, *E*, *B*, and *F* (pickled by Prof. Carpenter in spring of 1890) were placed in the calorimeter the heavy coatings of rust with which they were covered were scraped off, and very probably in this operation a portion of the carbon left on the surface by the acid treatment was also removed. This will account for the marked difference between the results which were obtained from the same plates, as shown by curves V. and VII., and by VI. and VIII.

Table II. gives the heat transmitted by each plate in percentages of that transmitted by plates 1 and 9, as before explained. It is necessary to make the comparisons by percentages because the calorimeters used by Prof. Carpenter and the writer do not admit of direct comparisons of the quantity q, it being in one case 278.0, and in the other, under other conditions, 113.2 B.T.U.

From the data given in Table II. the curves shown were plotted with percentages of heat transmitted as ordinates and days of immersion as abscissæ, cases No. 1 and No. 9, the untreated plates, being taken as unity.

.TABLE II.

No. of plate.	% heat transmitted.	Days of immersion.	Solution in which immersed.	REMARKS.
1	100.0	0	Untreated.	
X1 X2 X3	81.0 76.7 79.0	10 20 30	10% nitric. " " " "	Curve I.
1 2 3	85.9 71.4 71.6	10 20 30	5% " " " " "	Curve II.
6 7 8	85.6 94.0 103.9	10 20 30	1% " " " " "	Curve III. Peculiar results, but believed to be accurate.
9	100.0	0	Untreated.	
9 11 12	87.0 73.4 77.9	10 20 30	10% nitric. " " " "	Curve IV.
N K P	76.8 70.0 68.5	9 18 40	5% " " " " "	Curve V. From Prof. Carpenter's results.
E B F	86.3 70.7 68.7	9 18 40	1% " " " " "	Curve VI. From Prof. Carpenter's results.
N K P	102.0 91.0 81.8	91 18 40	5% " " " " "	Curve VII.
				After removing rust as explained.
E B F	100.0 96.7 90.6	9 18 40	1% " " " " "	Curve VIII.
13 1	87.9 85.9	15 15	5% hydrochloric. " "	Rough. Both sides planed. — Marked by numbered points on sheet of curves.
P 6	86.8 85.5	15 15	5% sulphuric. " "	Rough. Both sides planed.
7 9 11 12	33.8 31.7 35 0 32.6			Both sides painted with coach varnish.
2	46.2			One side varnished.

In these curves (Fig. 256), the only striking discrepancy is No. III., which is given by the planed plates, in 1% nitric acid. From the study of others we conclude that a minimum of heat transmitted is reached, in these cases, after about 20 days' immer-

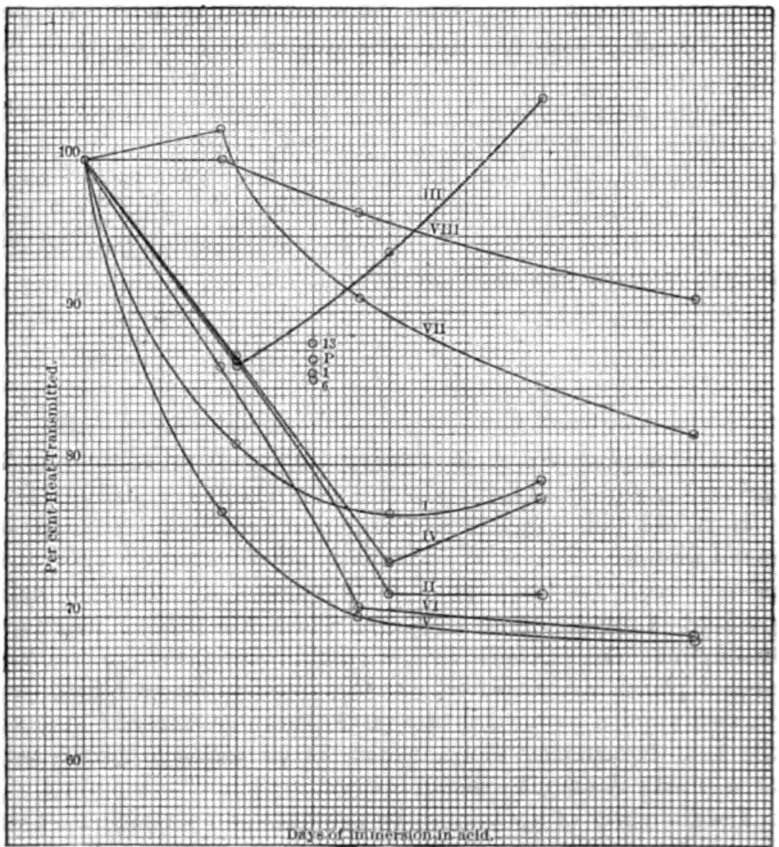


FIG. 256.

sion in a 5% or 10% solution, when nitric acid is the attacking agent.

With the other two acids we see that nearly the same result was achieved by them with a 5% solution and 15 days' immersion as with a 5% nitric acid solution and 10 days' immersion.

When both sides of the plates were varnished it is shown that, on an average, but 33% as much heat was transmitted as when the plate was untreated. The reduction was practically the same with planed as with rough plates, but it was found that the varnish adhered much better to the rough plates, and especially if they had been pickled before applying the varnish.

It must be borne in mind that the percentages given in Table II. are for plates having both sides attacked by the acid. The probable effect with one side only treated may be found as follows:

Assume each surface to present the same resistance to the transfer of heat through it and denote by x the fraction stopped by the first surface; then the second surface will stop the same fraction of what passed through the first, or $x(1-x)$ of the whole. Denote by y the fraction of the heat which passes through the plates; then we have the equation:

$$x + x(1-x) = 1 - y,$$
$$(1-x)^2 = y,$$
$$\text{or } x = 1 - \sqrt{y}.$$

Consider plate No. 2 after being pickled in 5% nitric acid 20 days: $-y$ equals .714, hence $x = 1 - \sqrt{.714} = 1 - .845 = .155$.

Next, the side exposed to steam was varnished. It must be noted that the effect of the varnish is not superposed upon that of the acid, but that when it is applied the result is independent of any previous treatment which the surface may have undergone. This is seen in the results obtained from the varnished plates. Also Mr. P. M. Chamberlain found (1890) that varnish applied to one side of a plate otherwise untreated reduced the heat transmitted 41.9%, which, as we will see, agrees very closely with the results shown below.*

Let the effect of the varnish be to intercept a fraction, z, of the heat; then for No. 2 we have

$$z + .155(1-z) = 1 - .4617;$$

hence, $z = .454$, or the single coating of varnish intercepts 45.4% of the heat.

Apply the equation $x = 1 - \sqrt{y}$ to the results obtained from the four plates varnished on both sides, and we have for the reduction of heat transferred, which would occur were one side only varnished and the other left untreated, the following values:

* *Trans. Am. Soc. Civ. Engrs.*, Vol. XXIII., July, 1890, No. 444, "A Practical Method of Reducing Internal Wastes in Steam Engines."—R. H. THURSTON.

For No. 7,	41.9%
9,	43.7
11,	40.9
12,	43.0
Mean,	42.4%

It is expected that when, as is now proposed, this treatment is applied to the surfaces bounding the clearance spaces of a steam-cylinder, the extent to which the iron, so treated, will store and restore heat, introduced into the engine by the steam, will be diminished about 40%, and thus some practically useful reduction of the internal wastes will be effected. In the engine the varnish is relied upon to retard this interchange of heat, and the alteration of surface produced by the action of the acid is expected to render the varnish more adhesive and permanent.

DISCUSSION.

Prof. R. H. Thurston.—The purpose of Mr. Royse's investigation is ultimately, of course, that of ascertaining the possibility of applying this method of retardation of heat flow to the reduction of the heat wastes of the steam-engine and other motors. The theory is simple and obvious: The waste which we are now most interested in reducing is that which results from the temporary storage of heat from the entering steam, and its later rejection from the metal of the cylinder walls, without application to the production of useful effect. This process is facilitated by high specific heat and high conductivity as well; it is checked by reduction of either specific heat or conductivity, or both. Make the conductivity zero, and the heat cannot enter the metal; make its specific heat zero, and the metal can store no heat; reduce both, and the waste varies as some function of their product, or the product of some function of both. This is an attempt to secure improvement by reducing both. Should it prove possible thus to modify the physical state of the interior of the engine to any extent, this now serious loss by "cylinder condensation" would be correspondingly lessened.

It is, of course, impossible to apply this process to the internal surfaces, so far as they are exposed to friction of the piston; but as the waste takes place mainly at the beginning of the stroke, and as the rubbing surfaces there constitute but a small fraction of the area wasting steam, it would seem possible, by thus treating the surfaces of the piston, the heads, and the port-spaces, the real heat-wasting areas, somewhat to reduce the loss. As we do not know just what part the moisture in the engine and adhering to its inner walls may play in this matter, it remains very uncertain to what extent even effective treatment of these metallic surfaces may prove useful, until the experiment is tried. These experiments, and those of Mr. Chamberlain and Prof. Carpenter, already described, show a possibility of reduction to the extent of 40% by even the imperfect process thus far applied. This would mean the reduction of the expenditures of heat and steam in the engine, could it be thoroughly applied, of from 10% in large unjacketed engines, to 20 and 30%, and even more, in small sizes. But we have not yet found a way of securing that condition of surface which is practically attainable, and the results reported must be taken as representative of the most superficial and incomplete treatment. I have seen iron altered as here intended to the depth of half an inch, and even an inch. In these cases we have only secured what we seek in the most superficial manner.

Since this paper was forwarded, Mr. Roysc has completed his first series of applications of the Hirn and Dwelshauvers analysis to a little 6 H.P. engine, and, after so adjusting the machine as to secure the best performance in its history—about 40 lbs. of steam per horse-power per hour—applied this process, and at the first attempt, superficially as was the metal affected, secured a gain of 10% in the steam consumption. Mr. Emery has long since done much better than this by other methods, and it seems to me that the outlook in this direction is by no means unpromising. Should the matter prove to interest members generally, we will endeavor to give the facts in some detail at another time. I am not at all sure that the particular method proposed by me will prove practically useful; but I think it probable that it may, and am very confident that, in some way and by some method, this now burdensome tax on the engine and its users will be removed, and before very long.

Prof. R. C. Carpenter.—This experiment being a continuation of one communicated to the Society by myself,* and to a certain extent under my auspices, has naturally attracted the most lively interest on my part. In the discussion of my paper, read at the

* American Society Mechanical Engineers, *Transactions*, Vol. XII., page 174.

Richmond meeting, several eminent engineers* predicted that, while the investigation might bear fruit by application in certain directions, it was a waste of time and effort to expect to reduce the loss of heat in the steam-engine cylinder, as claimed by Prof. Thurston, and made the subject of a patent, by any treatment or similar device. The objections, although considered of very great weight, seemed to the writer to be founded largely on a misunderstanding of the method of treatment proposed by Dr. Thurston, and of the action of the proposed covering of the metal, so I proposed to continue the investigation, but at my request Mr. Royse relieved me of further experimental work, promised in my previous paper, and continued the investigation as previously planned.

The plan of this investigation was, first, to find that surface treatment of a plate which, if not subjected to the wear from rubbing surfaces, would remain fairly permanent in an engine cylinder, and would be most efficient in reducing the flow of heat.

Second, knowing how to treat the surfaces of castings, this method was to be applied to an actual engine, and the effect determined by a comparison of careful tests made before and after the treatment.

What has been done in the first part of the investigation is now before you; it is imperfect, in that the field of experiment has been very limited, and it is probably true that other methods of treatment may be much more efficient than the ones investigated; but Mr. Royse has shown that the proposed treatment may reduce the flow of heat, as compared with an untreated plate, some 30% by treatment with acid alone, and nearly 70% by treatment with acid and a drying varnish, thus verifying the results obtained by previous experimenters.

Mr. Royse has gone further than this, although the later results are not given in his paper. He has carried the application of the method to an actual engine.

The engine used was a small plain slide-valve engine with throttling governor, known locally as the Payne engine, and used to a great extent in laboratory experiments. The diameter of cylinder is six inches, the stroke eight inches. The engine was in good condition, and had been previously tested within a few months, thirty or forty times, in class exercises, so that its usual

* American Society Mechanical Engineers, *Transactions*, Vol. XII., page 180.

economy was well known, and the steam consumption per I.H.P. had varied in these tests, under different conditions, from 45 to 60 lbs.

The first test was made by Mr. Royse, April 13, 1891, the method of testing being exactly as described in my paper, CCCCLI.,* "Application of Hirn's Analysis," as presented to this meeting, and a figure of the engine as arranged for the test is shown on page 809, Fig. 236 of that paper.

The power was absorbed by an Alden brake.†

A valve with an early cut-off was selected, and all the load that the engine would carry was put on the brake, without losing speed, and this load was kept constant throughout the test. The condensed steam and condensing water were carefully weighed on calibrated scales, and every precaution taken to prevent error. The calibration of indicator springs, gauges, and thermometers is shown in the complete record of the test. The result of the test was a steam consumption of 39.351 lbs. per I.H.P., and of 55.351 lbs. per brake horse-power, which was the best result ever yet obtained with that engine in any test, due, no doubt, to the load being sufficiently heavy to counteract any throttling effect of the governor. A copy of the card obtained is submitted with the test.

The working parts were then removed from the cylinder, and the sides protected by coating with paraffine, not acted on by dilute acid. The cylinder was then filled with a 10% solution of nitric acid, arranged to act on the clearance spaces, steam-ports, and heads of the cylinder. The piston, with rings removed, was similarly treated in a bucket containing the solution.

After the expiration of twelve days the acid solution was removed, and the cylinder was thoroughly cleaned; the heads and ports and piston were then given a heavy coat of varnish, and the engine put together in the same condition as before the treatment.

It was then tested in exactly the same manner as before the treatment, on date of May 1, 1891, giving as a result a steam consumption of 36.0413 lbs. per I.H.P., and 47.8022 lbs. per brake horse-power.

A sample of the resulting cards is given with the data of the test (Figs. 328–331). The result shows an increase in efficiency

* American Society Mechanical Engineers, *Transactions*, Vol. XII., page 790.

† American Society Mechanical Engineers, *Transactions*, Vol. XI., page 959.

respecting consumption of water, reckoned from the I.H.P., of 8.41%, reckoned from the brake horse-power, of 18.5%, this difference being due to the fact that the friction on the second trial was less than on the first, due probably to new and better lubricated packings. The I.H.P. was essentially the same in both trials.

To ascertain whether or not there was in reality any saving of

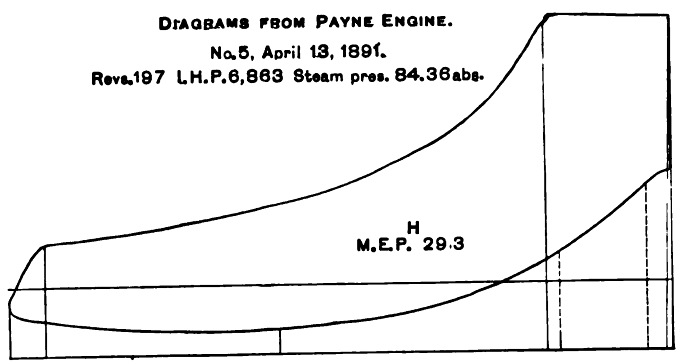


FIG. 328.

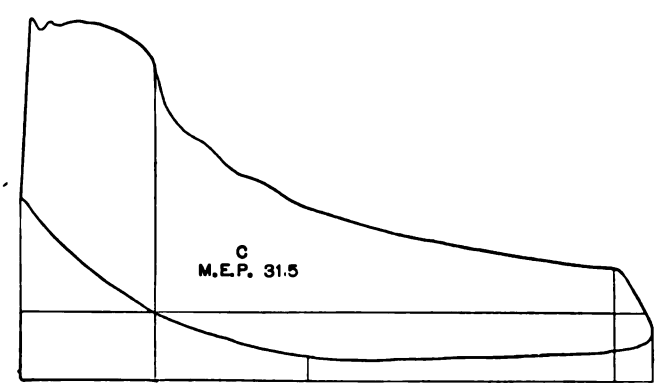


FIG. 329.

heat due to this treatment, Mr. Royse applied the methods of Hirn's analysis to both these tests, in which he took into consideration every diagram taken, and in which he checked up the numerical results to four decimal places. He used in the test the calorimeter described in Paper CCCCLI. for measuring the quality during compression, and in his calculation this result checked exactly with the measured quality. The methods of calculation are fully explained in that paper.

From his figures, which are submitted in full, it is seen that the initial cylinder condensation—which is denoted in the summary as "heat lost, admission"—was, in the first trial with the

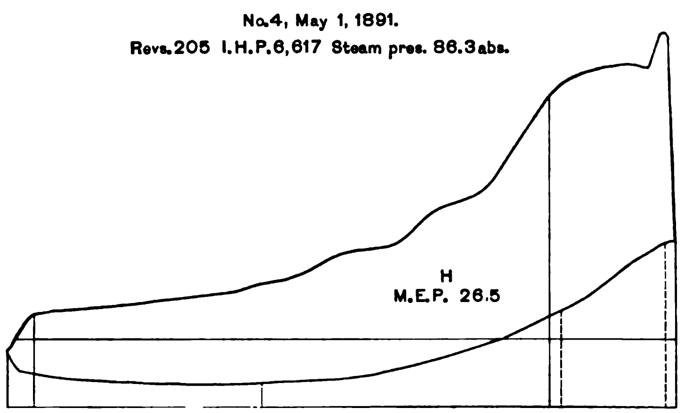


FIG. 330.

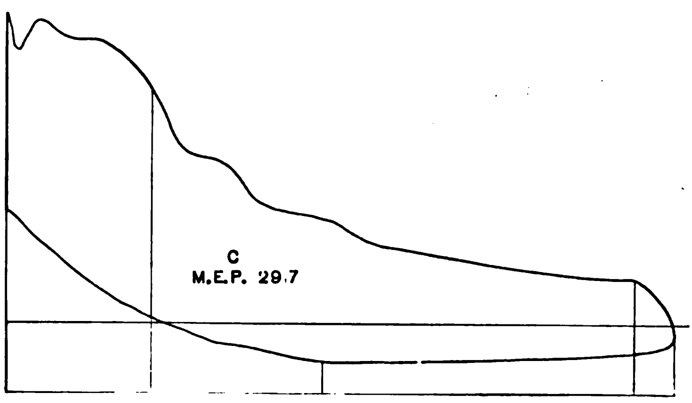


FIG. 331.

untreated engine, 42.893% of the total heat admitted, and in the second trial, with the treated engine, 36.403%, thus indicating a reduction of this loss of 15.1%. As shown in the summary, the heat restored and rejected is sensibly less. The thermodynamical efficiency was computed from data obtained from the indicator cards, and shows slightly in favor of the first trial. Had this been computed from the pressure and vacuum gauges, it would have been the same in both cases.

The actual efficiency, or the work done compared with the heat supplied in B.T.U., was 5.518% in the first trial, and 6.086% in the second trial, thus showing a gain due to the treatment of over 10% on that basis of computation.

Mr. Royse is entitled to much credit for the care with which these experiments have been performed, and for the exhaustive analysis made of the results. His investigation serves to show not so much the absolute value of this method of reducing cylinder condensation, as it does that there is a promising field for investigation to the American engineer in this direction.

CALIBRATION OF INSTRUMENTS.

STEAM-ENGINE INDICATORS.

	Head.		Crank.	
	Apr. 13.	May 1.	Apr. 13.	May 1.
Maker's Name	Ashcroft.	Bachelder.	Ashcroft.	Bachelder.
Maker's Number	Tabor, 782	111	Tabor, 781	110
Scale of Spring	40	40	40	40
Number of Spring	B.		A.	
When Tested	4:20	5:2	4:20	5:1
How Tested	Dead Weight.		Dead Weight.	
Per cent. Error	0	0	0	0

STEAM-GAUGES.

Maker.	Position.	Number.	Error, Lbs.	When Tested.	How Tested.
4:13 Ashcroft	Steam-chest.	6,888	See Curves.	4:15	Mercury Column.
5:1 Crosby	Steam-chest.	65,985		5:2	Mercury Column.

THERMOMETERS.

Position.	Registered Number.	Boiling Point.			Freezing Point.			Barometer.
		Reading.	Per Barom.	Error.	Reading.		Error.	
4:13 External Air & Engine Room	5,490	209.4	210.77	−1.37	31.7	32.0	.7	4:15, 29.200 5:1, 29.184
4:13 & 5:1 Condensed Steam.....	5,495	209.7	210.77	−1.07	31.8	32.0	.8	
4:13 & 5:1 Injection Water......	5,500	209.7	210.77	−1.07	32.0	32.0	.0	
4:13 & 5:1 Discharge Water.....	5,504	209.6	210.77	−1.17	31.9	32.0	.1	
4:13 { Exhaust Steam...........	5,262	210.8	210.77	− .03	32.0	32.0	.0	
4:13 { Calorimeter, compression..	5,258	211.3	210.77	+ .53	32.0	32.0	.0	
4:13 { Calorimeter, Steam-chest..	5,260	210.8	210.77	− .03	32.0	32.0	.0	
5:1 { Exhaust Steam...........	5,966	210.0	210.62	− .62	32.0	32.0	.0	
5:1 { Calorimeter, compression...	6,942	210.62	210.62	.0	32.0	32.0	.0	
5:1 { Calorimeter, Steam-chest ..	6,955	210.62	210.62	.0	32.0	32.0	.0	

APPLICATION OF HIRN'S ANALYSIS.

APRIL 13, 1891.

DATA AND RESULTS.

Test of steam-engine, made by Daniel Royse, at Sibley College, C. U.

Kind of engine..........................	Slide valve, throttling.
Diameter cylinder........................	6.06 inches.
Length stroke...........................	8 inches.
Diameter piston rod......................	1$\frac{3}{16}$ inches.
Volume cylinder, crank end.............	.12921.
Volume, head end.......................	.13354.
Volume clearance, cubic feet, head........	.01744.
Clearance in per cent. of stroke..........	13.06.
Volume clearance, cubic feet, crank.......	.01616.
Clearance in per cent. of stroke...........	12.51.
Pressure by gauge, steam-chest...........	64.80.
Pressure absolute, steam-chest............	79.155.
Revolutions per hour....................	11890.20.
Quality of steam in steam-pipe............	
Quality of steam in compression..........	1.0205.
Weight of condensed steam per hour......	259.92.
Barometer..............................	29.276 inches.
Boiling temperature, atmosphere pressure.	210.70.
Steam used during run, pounds...........	716.4240.
Quality of steam in steam-chest...........	.9941.
Quality of steam in exhaust..............	.9021.
Pounds of wet steam per stroke...........	
Head..................................	.0109707.
Crank.................................	.0109383.
Temperatures condensed steam...........	103.475° = Sg. + 32.
Temperatures condensing water, cold....	42.758° = Si. + 32.

Hot....................................	93.219° = Sk. + 32.
Pounds of condensing water per hour.....	5,044.878.
Per stroke............................	H .212016. C .212274.

SYMBOLS.

To denote different portions of the stroke, the following subscripts are used:

Admission, (*a*); expansion, (*b*); exhaust, (*c*); compression, (*d*).

To denote different events of the stroke, the following sub-numbers are used: Cut-off, (1); release, (2); compression, beginning of, (3); admission, beginning of, (0); in exhaust, (5).

Quality of steam denoted by X.

Cut-off, crank end, per cent. of stroke.....	20.544.
Release, crank end.......................	93.958.
Cut-off, head end, per cent. of stroke......	18.963.
Release, head end....	94.971.
Compression, crank end, per cent. of stroke	52.341.
Compression, head end, per cent. of stroke.	39.770.
Pounds of steam per I.H.P...............	39.351.
Pounds of steam per brake H.P	55.314.
I.H.P. H 3.3152, C 3.3054	6.6206.
Brake horse-power.......................	4.7100.

MECHANICAL LABORATORY, SIBLEY COLLEGE, CORNELL UNIVERSITY.

Test of Engine built by MESSRS. PAYNE, Elmira.

LOG OF TEST. ITHACA, N. Y., *April* 13, 1891.

		Revs.		Gauge Readings.						Temperatures.										Weights.			Head.		Crank.			Brake		
				Pounds.			Inches Hg.									Calorimeter.														
Number.	Time.	Continuous Counter.	Speed Indicator.	Boiler.	Steam-pipe.	Steam-chest.	Exhaust.	Condenser.	Barometer.	External Air.	Engine Room.	Condensed Steam.	Exhaust Steam.	Injection Water.	Discharge Water.	Steam-pipe.	Steam-chest.	Cylinder.	Exhaust.	Condensed Steam.	Feed Water.	Injection Water.	M.E.P.	I.H.P.	M.E.P.	I.H.P.	I.H.P. Total.	Load.	D.H.P.	Scale of Spring.
1	2:45		200			72.5	22	23^2			77.8	110.4	158	43.6	105.8			263.5					29.0	3.4	29.2	3.3				
2	3:00		201			72.0	22	23^3		66.2	79.4	111.6	158.5	43.7	108.5			272.5					27.3	3.2	29.7	3.4				
3	15		199			70.0	22	22^3			80	123.6	161	44.0	110.6			277.0					29.0	3.4	29.3	3.3				
4	30		198			72.0	22^1	24^2			79.4	113.6	157	42.8	83.3			280.0					30.4	3.5	32.5	3.6				
5	45		197			70.0	23	24^1			80.6	99.0	157	42.9	83.4			280					29.3	3.4	31.5	3.5				
6	4:00		203			71.0	22^2	24^2		66.4	79.6	110.8	157	43.4	98.6			280					29.7	3.5	30.2	3.4				
7	15		202			72.0	23	25^1	29.276		81.5	104.2	155.5	42.0	98.2			281					31.2	3.7	29.5	3.4				
8	30		196			72.0	23	24			81.4	104.0	158	42.9	91.4			281					29.7	3.4	31.2	3.4				
9	45		202			68.0	23	25^2			82.4	97.6	158	42.9	85.6			277					27.0	3.2	27.6	3.1				
10	5:00		200			62.0	23^2	25^1		67.0	83.0	92.2	158.5	43.0	89.1			276					28.3	3.3	29.0	3.3				
11	15		180			60.0	22^2	23^2			82	89.0	157	43.0	85.8			273					26.5	2.8	26.5	2.7				
12	30		200			65.0	24	26			81.2	97.8	151.5	43.0	83.2			271					27.0	3.1	28.5	3.2				
Mean			198.17			68.88	22.73	24.42		66.5	80.7	104.5	157¼	43.9	93.14			276												
Correct'd mean			198.17			64.80 79.155 abs.			14.355 lbs.	65.865°	79.923°	103.475°	157.250°	42.758°	92.219°			276.828°		714.750		13873.41		3.3152		3.3054	6.6206	57	4.7100	40

DATA FROM DIAGRAMS—APRIL 13, 1891.

No.	Total. M.E.P.	Total. I.H.P.	Total. Length, in.	Admission. M.E.P.	Admission. P_0.	Admission. Length, in.	Expansion. M.E.P.	Expansion. P_1.	Expansion. Length, in.	Exhaust. M.E.P.	Exhaust. P_2.	Compression. M.E.P.	Compression. P_3.	Compression. Length, in.	Calor. M.E.P.	Calor. Length, in.
Head.																
1....	29.0	3.379	3.48	13.5	38.0	.66	26.1	67.6	2.65	1.4	22.0	9.2	4.6	2.07	3.6	.46
2....	27.3	3.197	3.48	12.5	38.0	.66	25.7	63.2	2.65	1.3	22.0	9.6	4.6	2.06	3.6	.46
3....	29.0	3.362	3.48	13.2	38.8	.66	26.9	68.4	2.65	1.4	22.8	9.7	5.2	2.04	3.6	.46
4....	30.4	3.507	3.48	13.2	35.6	.66	27.1	68.0	2.64	1.2	22.8	8.7	4.4	2.11	3.4	.46
5....	29.3	3.363	3.48	12.7	36.8	.66	26.9	68.8	2.64	1.3	22.8	9.0	4.8	2.04	3.4	.46
6....	29.7	3.512	3.48	13.1	36.0	.66	26.5	66.6	2.64	1.2	22.0	8.7	4.4	2.16	3.4	.46
7....	31.2	3.672	3.48	14.3	40.0	.66	28.2	71.6	2.65	1.3	23.6	10.0	5.2	2.07	3.7	.46
8....	29.7	3.391	3.48	13.1	36.4	.66	26.6	66.8	2.64	1.1	22.4	8.9	4.0	2.06	3.6	.46
9....	27.0	3.177	3.48	12.0	34.0	.66	24.0	60.0	2.65	1.2	20.8	7.8	4.4	2.00	3.4	.46
10....	28.3	3.297	3.48	12.5	35.2	.66	25.2	63.2	2.65	1.2	20.8	8.2	4.0	2.04	3.5	.46
11....	26.5	2.810	3.48	12.1	34.4	.66	24.2	62.0	2.64	1.3	20.4	8.5	4.4	2.01	3.2	.46
12....	27.0	3.146	3.48	12.4	35.2	.66	24.3	60.4	2.64	1.3	20.8	8.4	4.4	2.06	3.4	.46
Mean..	28.7	3.315	3.48	12.88	36.533	.66	25.978	65.55	2.645	1.267	21.933	8.891	4.533	2.06	3.483	.46
Crank.																
1....	29.2	3.291	3.31	14.7	38.0	.68	24.0	61.2	2.42	1.7	22.8	7.8	5.2	1.64	.....	...
2 ..	29.7	3.367	3.31	14.7	39.2	.68	24.5	62.8	2.42	2.0	23.6	7.5	5.6	1.53	.....	...
3....	29.3	3.288	3.31	14.5	38.8	.68	25.0	62.0	2.44	2.4	23.6	7.8	5.6	1.61	.....	...
4....	32.5	3.629	3.31	15.3	38.4	.68	26.2	67.2	2.44	2.0	24.0	7.0	5.2	1.57	.....	...
5 ...	31.5	3.500	3.31	14.7	38.4	.68	25.5	66.0	2.43	1.7	24.0	7.0	4.8	1.50	.....	...
6....	30.2	3.458	3.31	14.8	37.6	.68	25.2	66.4	2.43	2.3	24.0	7.5	5.2	1.52	.....	...
7....	29.5	3.361	3.31	14.5	86.8	.68	25.2	62.0	2.44	2.5	24.0	7.7	5.6	1.61	.....	...
8 ...	31.2	3.449	3.31	15.2	38.0	.68	25.6	66.0	2.42	2.0	24.0	7.6	5.6	1.58	.....	...
9....	27.6	3.144	3.31	13.5	34.0	.68	23.1	56.8	2.44	2.0	22.4	7.0	4.8	1.60	.....	...
10....	29.0	3.271	3.31	14.3	36.4	.68	24.0	60.4	2.43	1.8	22.8	7.5	4.8	1.56	.....	...
11....	26.5	2.690	3.31	11.9	36.0	.68	23.9	61.2	2.43	1.9	22.8	7.4	5.2	1.65	.....	...
12....	28.5	3.215	3.31	14.0	36.0	.68	23.5	58.8	2.42	1.9	22.8	7.1	5.2	1.56	.....	...
Mean..	29.56	3.305	3.31	14.34	37.30	.68	24.64	62.57	2.43	2.017	23.4	7.408	5.233	1.578	.....	...

ABSOLUTE PRESSURES FROM INDICATOR DIAGRAMS AND CORRESPONDING PROPERTIES OF SATURATED STEAM.

April 13th, 1891.		Cut-off.	Release.	Beginning. Compression.	Beginning. Of Admiss'n.	Symbols. Thurston.	Symbols. Peabody.
Subscripts used..............		1	2	3	0		
Absolute Pressure.......	Head.	65.550	21.933	4.533	36.533	P	p
	Crank	62.567	23.400	5.233	37.300		
Heat of Liquid	Head.	268.1796	201.6474	126.2236	230.8799	S	q
	Crank	265.0437	205.1988	132.8893	232.1333		
Internal Latent Heat......	Head.	825.9524	877.3991	936.6752	854.6949	I	ρ
	Crank	828.3622	874.6360	931.7952	853.7249		
Latent Heat Evaporation.	Head.	....	....	....	....	L	r
	Crank	....	....	....	....		
Total Heat................	Head.	....	....	....	....	H	λ
	Crank	....	....	....	....		
Vol. 1 lb. Cu. Ft....... ..	Head.	6.4748	18.0536	80.4758	11.2054	C	μ
	Crank	6.6977	17.0280	69.8991	10.9860		
Volumes, Symbols..............		$V_c + V_1$	$V_c + V_2$	$V_c + V_3$	$V_c + V_0$	....	....
Volumes, Head, Cu. Ft		.042763	.144264	.097871	.01882	....	....
Volumes, Crank, Cu. Ft		.042705	.137563	.077740	.01616	....	....
Water per I.H.P. from Diagram...		W	W^1				
Water per I.H.P., Head, lbs.......		....	....				
Water per I.H.P., Crank, lbs		....	....				

MEAN PRESSURES AND HEAT EQUIVALENTS OF EXTERNAL WORK.

	Subscripts.	Head End.			Crank End.		
		Mean Pres'ures.	External Work. Foot lbs.	External Work. B.T.U.	Mean Pre'sures.	External Work. Foot lbs.	External Work. B.T.U.
Symbols		M.E.P.	W	$W \div 778$	M.E.P.	W	$W \div 778$
Admission	a	12.883	....	31.8356	14.342	....	34.3079
Expansion	b	25.978	....	64.1862	24.642	....	58.9474
Exhaust	c	1.2678	....	−3.1300	−2.017	....	−4.8242
Compression	d	8.891	....	−21.9711	−7.408	....	−17.7230
Total		28.700	....	70.9207	29.558	....	70.7090

The M.E.P.'s are computed with the entire length of the diagram as a base line.

APPLICATION OF HIRN'S ANALYSIS.

May 1, 1891.

DATA AND RESULTS.

Test of steam-engine, made by Daniel Royse, at Sibley College, C. U.

Kind of engine	Slide valve, throttling.
Diameter cylinder	6.06 inches.
Length stroke	8 inches.
Diameter piston rod	$1\frac{3}{16}$ inches.
Volume cylinder, crank end	.12921.
Volume, head end	.13354.
Volume clearance, cubic feet, head	.01744.
Clearance in per cent. of stroke	13.06.
Volume clearance, cubic feet, crank	.01616.
Clearance in per cent. of stroke	12.51.
Pressure by gauge, steam-chest	69.40.
Pressure absolute " "	83.700.
Revolutions per hour	12393.60.
Quality of steam in steam-pipe	
Quality of steam in compression	1.020.
Barometer	29.132 inches.
Boiling temperature, atmosphere pressure.	210.62.°
Steam used during run, pounds	586.7041.
Quality of steam in steam-chest	.9799
Quality of steam in exhaust	.86209.
Weight of condensed steam per hour	234 000.
Pounds of wet steam per stroke	
Head	.0091623.
Crank	.0097722.
Temperatures condensed steam	95.060° = Sg. + 32.
Temperatures condensing water, cold	52.065° = Si. + 32.
Hot	87.555° = Sk. + 32.
Pounds of condensing water, per hour	6091.62.
Per stroke	H .237114. C .254396.

SYMBOLS.

To denote different portions of the stroke, the following subscripts are used:

Admission, (*a*); expansion, (*b*); exhaust, (*c*); compression, (*d*).

To denote different events of the stroke, the following sub-numbers are used: Cut-off, (1); release, (2); compression, beginning of, (3); admission, beginning of, (0); in exhaust, (5).

Quality of steam denoted by X.

Cut-off, crank end, per cent. of stroke.....	21.843.
Release, crank end.......................	94.977.
Cut-off, head end, per cent. of stroke......	19.143.
Release, head end........................	95.714.
Compression, crank end, per cent. of stroke.	52.153.
Pounds of steam per I.H.P	36.0413.
Compression, head end, per cent. of stroke.	37.714.
Pounds of steam per brake H.P...........	47.8022.
I.H.P. $\begin{cases} H & 3.17631 \\ C & 3.33507 \end{cases}$	6.51138.
Brake horse-power..................	4.90943.

MECHANICAL LABORATORY, SIBLEY COLLEGE, CORNELL UNIVERSITY.

Test of Engine built by MESSRS. PAYNE, Elmira.

LOG OF TEST. ITHACA, N. Y., *May* 1, 1891.

		Revs.		Gauge Readings.					Temperatures.										Weights.			Head.		Crank.			Brake.		
				Pounds.		Inches Hg.									Calorimeter.														
Number.	Time.	Continuous Counter.	Speed Indicator.	% Entrained Water.	Steam-chest.	Exhaust.	Condenser.	Barometer.	External Air.	Engine Room.	Condensed Steam.	Exhaust Steam.	Injection Water.	Discharge Water.	Steam-pipe.	Steam-chest.	Cylinder.	Exhaust.	Condensed Steam.	Feed Water.	Injection Water.	M.E.P.	I.H.P.	M.E.P.	I.H.P.	I.H.P. Total.	Load.	D.H.P.	Scale of Spring.
1	7:15		206	2.66	73	23^1	24^2		58	80	100	152	53	19		228	261					26.5	3.2	29.7	3.5				
2	30		205	1.50	67	23^2	24^3			78	103	151	53	85		246	260					26.0	3.1	30.0	3.5				
3	45		207	1.70	72	23^3	25			80	97	152	52.5	85.5		245	261					26.4	3.2	27.8	3.3				
4	8:00		205	2.50	72	23^3	25			80	95	151	52.5	88.5		230	261					26.5	3.2	29.7	3.4				
																						Indicator Spring Slipped.							
5	15		206	2.45	73	23^2	25		55	80	95	152	52.5	87.5		232	263												
6	30		207	2.70	71	23^3	25	29.182		80	95	152	52.5	88		226	261					24.6	3.0	27.0	3.1				
7	45		207	2.16	67	23^3	25½			79	95	152	52.5	87		234	260					26.4	3.2	28.0	3.3				
8	9:00		207	2.05	67	23^2	25½		54	79	95	152	52.5	91		236	260					26.0	3.1	27.5	3.2				
9	15		208	2.05	67	23^2	25½			79	95	151	52.5	87		236	261					26.3	3.2	27.0	3.2				
10	30		207	1.33	68	23^2	25½			81	94	150	52.5	87.5		250	261					27.1	3.3	29.7	3.5				
11	45		207	1.00	70	23^3	25½		55	82	93	151	52.5	86.5		258	266					28.1	3.4	29.7	3.5				
Mean.			206.56	2.01	69.73	23.06			54.5	79.8	96.0	151.15	52.76	88.41															
Correct'd mean.			206.56	x=97.99	69.40 83.700 abs.			14.300 lbs.	54.50°	79.80°	95.06°	151.66°	52.065°	87.555°			261.40		585 lbs.		15,229.06 lbs.		3.17631		3.33507	6.51138	57	4.90943	40

DATA FROM DIAGRAMS—MAY 1, 1891.

No.	Total.			Admission.			Expansion.			Exhaust.			Compression.			Calor.	
	M.E.P.	I.H.P.	Length, in.	M.E.P.	P_0.	Length, in.	M.E.P.	P_1.	Length, in.	M.E.P.	P_2.	Length, in.	M.E.P.	P_3.	Length, in.	M.E.P.	Length, in.
Head 1	26.5	3.1803	3.50	13.8	35.6	.67	23.7	63.6	2.67	1.5	19.6		9.5	5.2	2.18	4.1	.51
2	26.0	3.1051	3.50	13.3	34.4	.67	24.0	62.3	2.70	2.0	18.8		9.3	5.2	2.18	4.2	.51
3	26.4	3.1836	3.50	13.5	34.4	.67	24.1	62.0	2.70	2.0	19.2		9.2	5.6	2.18	4.0	.51
4	26.5	3.1648	3.50	13.2	34.5	.67	23.8	64.0	2.68	1.5	19.2		9.0	4.4	2.18	4.2	.51
6	24.6	2.9666	3.50	12.7	33.6	.67	23.2	59.6	2.68	2.0	19.2		9.3	5.2	2.18	3.9	.51
7	26.4	3.1836	3.50	13.5	33.6	.67	24.2	62.4	2.69	2.0	19.2		9.3	5.2	2.18	3.9	.51
8	26.0	3.1354	3.50	13.2	33.6	.67	23.3	60.0	2.66	1.9	19.2		8.6	4.8	2.18	3.7	.51
9	26.3	3.1869	3.50	13.4	33.6	.67	23.6	61.6	2.67	1.9	19.2		8.8	5.2	2.18	3.8	.51
10	27.1	3.2681	3.50	14.0	34.0	.67	24.3	64.4	2.68	2.1	19.6		9.1	5.2	2.18	3.7	.51
11	28.1	3.3887	3.50	14.5	34.4	.67	24.6	65.2	2.67	1.7	19.2		9.3	4.8	2.18	3.8	.51
Mean	26.39	3.1763	3.50	13.51	34.17	.67.	23.88	63.56	2.68	1.86	19.24		9.14	5.08	2.18	3.94	.51
Crank 1	29.7	3.4805	3.46	15.7	38.4	.76	24.5	60.0	2.53	2.8	22.8		7.7	5.6	1.67	....	...
2	30.0	3.4684	3.48	15.8	38.0	.76	25.4	63.6	2.56	3.3	24.0		7.9	7.6	1.66	....	...
3	27.8	3.2455	3.48	15.0	37.2	.76	23.3	58.8	2.54	2.8	22.4		7.7	6.8	1.67	....	...
4	29.7	3.4338	3.49	16.1	38.4	.76	24.6	63.2	2.54	2.8	22.8		8.2	6.4	1.66	....	...
6	27.0	3.1121	3.48	14.3	35.8	.76	23.8	58.0	2.54	3.3	22.0		7.8	6.8	1.67	....	...
7	28.0	3.2688	3.49	15.3	36.4	.76	23.8	61.2	2.55	3.4	22.8		7.7	6.8	1.67	....	...
8	27.5	3.2097	3.50	15.3	37.2	.76	23.6	60.8	2.55	3.3	22.4		8.1	6.8	1.67	....	...
9	27.0	3.1673	3.49	14.7	36.8	.76	23.4	56.4	2.55	3.5	22.0		7.6	6.8	1.67	....	...
10	29.7	3.4673	3.50	16.6	38.0	.76	24.6	63.6	2.56	3.0	22.8		8.5	7.2	1.66	....	...
11	29.7	3.4673	3.47	16.4	40.0	.76	24.8	63.2	2.58	3.3	20.8		8.2	7.2	1.67	....	...
Mean ..	28.61	3.3351	3.484	15.52	37.62	.76	24.18	60.88	2.549	3.15	22.48		7.94	6.80	1.667	....	...

ABSOLUTE PRESSURES FROM INDICATOR DIAGRAMS AND CORRESPONDING PROPERTIES OF SATURATED STEAM.

May 1, 1891.		Cut-off.	Release.	Beginning. Compression.	Beginning. Of Admiss'n.	Symbols. Thurston.	Symbols. Peabody.
Subscripts used		1	2	3	0		
Absolute Pressure	Head.	63.560	19.240	5.080	34.170		
	Crank	60.880	22.480	6.800	37.620	P	p
Heat of Liquid	Head.	266.0096	194.5749	131.1900	226.6087		
	Crank	263.2160	202.9887	143.6506	232.6488	S	q
Internal Latent Heat	Head.	827.5392	882.9139	932.7829	857.7864		
	Crank	829.7642	876.3555	922.7808	853.3259	I	ρ
Latent Heat Evaporation	Head.	....	....	....	....		
	Crank	....	....	....	....	L	r
Total Heat	Head.	....	....	....	....		
	Crank	....	....	....	....	H	λ
Vol. 1 lb., Cu. Ft	Head.	6.6536	20.4612	71.6464	11.9256		
	Crank	6.9290	17.6848	54.5400	10.8964	C	μ
Volumes, Symbols		$V_c + V_1$	$V_c + V_2$	$V_c + V_3$	$V_c + V_0$	....	....
Volumes, Head, Cu. Ft		.043006	.145256	.10062[illegible]	.017440	....	....
Volumes, Crank, Cu. Ft		.044383	.138880	.077983	.016160	....	....
Water per I.H.P. from Diagram		W	W^1				
Water per I.H.P., Head, lbs		....	....				
Water per I.H.P., Crank, lbs		....	....				

MEAN PRESSURES AND HEAT EQUIVALENTS OF EXTERNAL WORK.

	Subscripts.	Head End.			Crank End.		
		Mean Pressures	External Work. Foot lbs.	External Work. B.T.U.	Mean Pressures	External Work. Foot lbs.	External Work. B.T.U.
Symbols		M.E.P.	W	$W \div 778$	M.E.P.	W	$W \div 778$
Admission	a	13.510	....	33.3900	15.520	....	37.1176
Expansion	b	23.880	....	59.0194	24.180	....	57.8289
Exhaust	c	1.860	....	−4.5970	3.150	....	−7.5335
Compression	d	9.140	....	−22.5895	7.940	....	−18.9893
Total		26.390	....	65.2229	28.610	....	68.4237

The M.E.P.'s are computed with the entire length of the diagram as a base line.

DATA AND RESULTS PER 100 STROKES.

QUANTITIES.	SYMBOLS.	FORMULÆ.	APRIL 13.		MAY 1.	
			Results given in B.T.U.			
			HEAD.	CRANK.	HEAD.	CRANK.
Steam used by calorimeter, lbs.	M_x		.00490		.00545	
Steam from boiler, lbs.	M		1.09707	1.09883	.91693	.97722
Steam in clearance, lbs.	M_0	$100 \frac{V_c + V_0}{C_0 X_0}$	.16458	.14414	.14346	.14540
Steam, total, lbs.	$M + M_0$		1.26165	1.23797	1.05969	1.12262
Heat of condensed steam	K'	$(M - M_x) S_g$	78.0629	78.1815	57.4338	61.6235
Condensing water, lbs.	G		21.2016	21.2274	23.7114	25.4396
Heat given to condensing water	K	$G (S_h - S_i)$	1048.6523	1049.9284	844.5176	902.8514
Heat supplied to engine	Q	$M (XL + S)$	1265.2743	1281.4785	1062.8957	1133.6487
Sensible heat at admission	H_0	$M_0 S_0$	87.9932	83.4579	82.5093	83.8271
Internal heat at admission	H'_0	$100 \frac{V_c + V_0}{C_0} I_0$	143.5501	125.5798	125.5173	126.5534
Sensible heat at cut-off	H_1	$(M + M_0) S_1$	338.3488	328.2161	231.9631	295.4915
Internal heat at cut-off	H'_1	$100 \frac{V_c + V_1}{C_1} I_1$	545.5026	528.1695	534.8858	581.3785
Sensible heat at release	H_2	$(M + M_0) S_2$	254.4084	254.0800	206.1891	227.8784
Internal heat at release	H'_2	$100 \frac{V_c + V_2}{C_2} I_2$	701.1183	706.5868	626.7889	688.9081
Sensible heat, beginning of compression	H_3	$(M_0 + M_x) S_3$	21.3924	19.0826	19.5355	20.8868
Internal heat, beginning of compression	H'_3	$100 \frac{V_c + V_3}{C_3} I_3$	113.9140	108.6318	131.0014	131.9348
Cylinder loss during admission	Q_a	$Q + H_0 + H'_0 - H_1 - H'_1 - \frac{W_a}{778}$	551.1356	549.8245	370.6544	430.0416
Cylinder loss during expansion	Q_b	$H_1 + H'_1 - H_2 - H'_2 - \frac{W_b}{778}$	−135.8615	−163.1786	−75.1285	−147.0454
Cylinder loss during exhaust	Q_c	$H_2 + H'_2 - H_3 - H'_3 - K - K' - \frac{W_c}{778}$	−303.3649	−285.3828	−214.9183	−193.6765
Cylinder loss during compression	Q_d	$H_3 + H'_3 - H_0 - H'_0 - H_x - H'_x - \frac{W_d}{778}$	−29.7279	−18.6086	9.0961	11.4304
Heat discharged, and work	B	$K + K' + H_x + H_x' + \frac{W}{778}$	1203.0030	1198.8780	973.1920	1062.8986
Loss	D	$Q - B$	82.1813	82.6596	89.7087	100.7501
Loss	D'	$Q_a + Q_b + Q_c + Q_d$	82.1813	82.6595	89.7087	100.7501

SUMMARY AND RESULTS.

Quantities.	Symbols.	Formulæ.		April 13. Head.	April 13. Crank.	May 1. Head.	May 1. Crank.
Quality of steam entering	X	Per calorimeter	%	99.41	99.41	97.99	97.99
Quality of steam at cut-off	X_1	$100 \dfrac{Vc + V_1}{(M + M_0) C_1}$	"	52.84	51.50	60.966	57.045
Quality of steam at release	X_2	$100 \dfrac{Vc + V_2}{(M + M_0) C_2}$	"	63.84	65.26	66.048	69.953
Quality of steam at compression	X_3	$100 \dfrac{Vc + V_3}{(M_0 + Mx) C_3}$	"	71.75	77.16	94.318	98.838
Quality of steam at admission	X_0	$100 \dfrac{Vc + V_0}{M_0 C_0}$	"	102.05	102.05	102.00	102.00
Quality of steam at admission	X_0	Per calorimeter	"	102.05	102.05	102.00	102.00
Quality of steam in exhaust	X_5	$\left(\dfrac{K + K'}{M - Mx} - S_5\right) \div L_5$	"	90.214	90.195	86.328	85.996
Heat lost, admission *	a	$Qa \div Q$	"	42.881	42.905	34.872	37.984
Heat restored, expansion	b	$Qb \div Q$	"	10.571	12.738	7.068	12.962
Heat rejected, exhaust	c	$Qc \div Q$	"	23.603	22.270	20.220	17.081
Heat lost, compression	d	$Qd \div Q$	"	−2.818	−1.452	.856	1.008
Heat utilized, work	w	$\dfrac{W}{778} \div Q$	"	5.518	5.518	6.136	6.086
Heat lost, radiation	r	$D \div Q$	"	6.394	6.450	8.440	8.887
Ratio, radiation to work		$r \div w$	"	1.1588	1.1689	1.3755	1.4728
Ratio, cylinder condensation to work		$a \div w$		7.7711	7.7757	5.6318	6.2843
Thermodynamic efficiency †	E	$(t - t_2) \div (461 + t)$	"	19.921	19.031	19.715	16.546
Actual efficiency	E_1	$\dfrac{W}{778} \div Q$	"	5.518	5.518	6.136	6.036
Efficiency compared with ideal	E'	$E_1 \div E$	"	27.67	28.90	31.12	36.48

* Initial cylinder condensation.
† Calculated from indicator diagrams.

Mr. E. F. C. Davis.—This discussion seems to have taken a turn as to its relations to non-conducting heat in steam-cylinders. I would like to ask what experiments have been made in regard to pickling plates for boilers. That is a case in which the acid would probably have an injurious effect. The Government require all plates for their use to be pickled, and if it is going to have this effect in rendering the plates non-conductive, it is going to reduce the horse-power.

Mr. H. H. Suplee.—I think the last paragraph of the paper is interesting reading, from the fact that the author of the paper does not seem to think that the pickling amounts to much anyhow, but relies principally upon the varnish. It really is a question, therefore, of varnishing the plates.

The President.—These plates, I take it, are cast-iron plates?

Mr. Carleton W. Nason.—On the third page it will be noticed that he states the plates are of variable thicknesses, ranging from one-quarter to one-half inch. In his tables the thickness of the plate experimented upon is not given. I think, to be of any practical value, an experiment should be made in condensation on a plate first without pickling, and then the same plate should be taken and tested after pickling. Then, again, he has used no means in his experiment for keeping his water currents even and flowing constantly over the surfaces. He admits that his variations in temperature are, he thinks, due to this point. Here you have water on one side of the plates, and steam on the other; the condensation becomes so very active that a very small modification of the current on that surface will affect the condensation result largely. For that reason particular attention should be paid in experiments of this nature, and in which the conditions are similar to those of surface condensers, to see that the water currents are in all cases running at precisely the same velocities and in the same directions. I think that the experiments given are too crude, and should be made more carefully to have much value.

*Mr. Daniel Royse.**—Prof. Rankine, in the *Steam Engine*, discussing the transmission of heat through boiler plates, states that the resistance offered by the metal is so small compared with that offered by the surfaces that the former may be neglected. This is my authority for not considering the thickness of plate in my experiments.

* Author's Closure.

It has also been objected that there was no device for maintaining the water currents "even and constantly flowing" during a calorimeter run. The temperature of the outflow, as shown by the thermometer t', varied, the cause presumably being uneven water currents. However, by observing the temperature of the outflow at regular and frequent intervals during the run, a mean was found and the effects of uneven currents thus eliminated.

CCCCLXII.

TOPICAL DISCUSSIONS AND INTERCHANGE OF DATA.

XXIIID MEETING, PROVIDENCE, JUNE, 1891.

No. 462–86.

What limits are there to the speeds of hot-air or caloric engines? What is the least and greatest number of revolutions per minute known to you for such engines?

Mr. Geo. W. Bissell.—Two years ago I made a test of a caloric pumping engine having a stroke of piston of 2½″ and a diameter of cylinder of 6″. The pump was arranged to deliver the water at various heads from 5 to 70 feet. The speed diminished with increase of head, and ranged from 120 revolutions per minute at 5 feet head to 35 revolutions per minute at 70 feet head. These figures correspond to piston speeds of 50 and 14½ feet per minute respectively.

Mr. Fredk. Meriam Wheeler.—There is no reason why we cannot get as high a rotative speed in hot-air pumping engines as in any form of pumping engines. We all know, of course, that water cannot be hurried beyond a certain piston travel. The power of the engine being sufficient to handle the water, I do not see why the hot-air pumping engine cannot be run as high as 200 feet per minute.

No. 462–87.

Which is better economy in a foundry cupola, to melt rapidly, producing a relatively cool iron, or to melt more slowly, producing hotter iron?

Mr. W. O. Webber.—I have a few data in this connection which I would like to offer. From the first of May to the last of December, 1890, we were getting about an average of 100 tons of clean castings a week from a 60-inch cupola, and melting about 8 tons per hour at that rate. The ratio of iron melted per pound of fuel was seven and a half. The number of pounds of clean castings which we got out of the total amount of iron put into the cupola was 75¾%. We wanted to increase the amount which we could melt in that cupola. Our foundry foreman said of course it could be done, but at a sacrifice in the per cent. of good castings which we would obtain; that is, in melting it faster we would not melt it so hot, and we would lose more castings. I did not agree with him, and from January 10th to May 20th of this year we crowded the cupola so that the average melting for that period was at the rate of $11\frac{1}{3}$ tons per hour. The ratio of melting in pounds of iron melted to pounds of fuel was nine and a half. The difference between the efficiency of the cupola over the previous one is three-fourths of one per cent. We got 75% of good castings. So that the difference is very slight when you come to reduce it down to figures and take a record for a certain period, which is the only fair way. It makes a difference also of about three dollars a ton on the economical side of the question in the quicker melting. We have had some remarkably good weeks. During the week ending May 7th we melted $13\frac{1}{4}$ tons per hour in a common 60-inch cupola. We melted $10\frac{3}{10}$ lbs. of iron per pound of fuel, and the efficiency that week was 76.4 per cent.

Mr. E. C. F. Davis.—I would like to ask Mr. Webber how he obtained this increase.

Mr. W. O. Webber.—It was simply by using a stronger blast and putting in another row of tuyeres.

Mr. H. L. Gantt.—I would like to ask what the blast pressure was.

Mr. W. O. Webber.—Fourteen ounces, using just common coke.

Mr. Gus. C. Henning.—The question, I think, has to be analyzed a little in order to give a proper answer. A cupola may be used for two purposes, as a converter or as a melter. If as a melter simply, the quicker the metal is melted the less fuel will probably be used. If this is secured by a high temperature, then the metal must be held before casting. In the Herbertz cupola, which is now extensively introduced in Europe and in England, and which will soon have representatives here, a new departure has been introduced. On the accompanying sketch (Fig. 332) it will be noticed that at *P* there is marked a jet. Instead of driving the air in under pressure at the tuyeres there is a steam-jet put in at that point. The result is a large increase in economy, amounting from 14 to 18% of cost of castings. The fuel consumption is very much decreased by that plan, using cold air for ordinary purposes, having open tuyeres, or permitting the air to enter between the movable hearth and stack. In another case, when it is desired to have very high temperatures for melting steel or irons which require them, there are the channels or duct pipes C, which come from a chamber in the top of the stack for heating the air; the jet, however, is exactly the same as before. The air is drawn into P and through the channels or duct pipes C into the tuyeres, which are similar to those generally in use, and into the cupola. Under these conditions much greater fuel consumption is obtained, with less waste of metal. There is one radical difference in this cupola from all others. It does not convert metal like others; and when different metals are taken and mixed, the chemical or metallurgical effect in the furnace is comparatively slight. It has been found that in this furnace melting goes on so fast that it is merely a mixing furnace. There does not seem to be enough time to allow the carbon of the gases to combine with the iron or other materials to form fluxes, to change the material radically. At first, in trying to cast chilled car wheels the metal was so little changed from the original pig that they would not chill. A few experiments, however, demonstrated which crude metals had to be introduced to produce the desired effect.

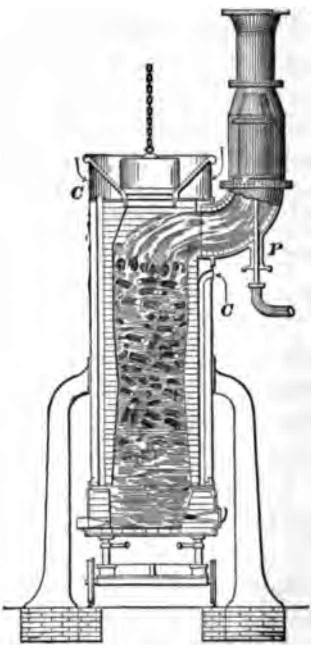


FIG. 332.

Mr. L. C. Jewett.—It is well known that when the bed is low in the cupola the melting proceeds faster; but there are greater chances for accident to cupola-ladles and to castings to be poured. The proportion of fuel to iron is a little less than when the bed is higher, and charges of fuel are gauged to maintain the bed at the point where melting is slower and hotter. When the melting-point is low and the production of melted metal is fast, the accumulations of "bugs" (so called in foundry parlance) are greater than when the reverse is the case. By bugs are meant the particles of metal which find lodgement and adhere to the spent fuel and cinders inside the cupola during the process of melting the iron. These bugs are recovered to a certain extent in most foundries by crushing the cinders in a tumbling mill, but that takes time, with wear and tear, and much goes to waste through the crevices in the mill. When the melting is done slower but hotter, there is less lodgement of metal among the spent fuel. Almost all foundries find that several different brands of iron bring best results when mixed in a charge or burden, and these various brands vary in quantity in each charge. To melt with a low bed and low temperature there is not produced as homogeneous metal in the casting as there will be if the mixture is melted at higher temperature and consequently slower.

To have the bed reasonably high—not too high, but just high enough—to insure good white-hot liquid metal, not only insures a thorough mixing of the various brands, but it prevents to a certain extent the changing of the nature of the carbon in the iron from the graphitic to the combined state, and it is believed to increase the amount of graphite in the resulting castings, thereby rendering them softer and more easily machined.

A few years since it was considered the best economy to have the tuyeres in foundry cupolas low, and many are still found with but 8 to 10 and 12 inches depth from bottom of tuyeres to sand bottom of cupola. To-day, however, it is generally conceded among the most experienced that the iron is improved if tuyeres are high, and 26 and 36 inches above the sand bottom will now be found in several foundries, with evident improvement in the quality of the castings. The improvement is undoubtedly due to the fact that the transit of the metal from the melting zone, some 20 to 22 inches above the tuyeres, to the sand bottom—if tuyeres were 4 inches diameter or wide—would cause the iron to travel through $22 + 4 + 36 = 62$ inches, while all the while in contact with the fuel, and give a long chance to absorb carbon. Consequently the evidence is that, with the high bed and liquid-hot iron, the condition of the resulting castings is likely to be improved.

With hot metal fewer castings are lost from not running well; one does not have so many cold shuts, nor are so many poured short from the ladles, being what moulders term bunged up, etc. To sum up the results of the two methods: With low bed and quick melting, you have apparent lower fuel consumption, less time for the running of fans, and thus less wear and tear, with less consumption of power, etc.

With higher bed and a little more time in melting, you use a little more fuel, but have better iron, lower percentage of bad castings, less percentage of waste of metal in the form of bugs and from pouring on the floor to save bunging up the ladles; less daubing used to reline the ladles and fuel to dry them, and less time of operator to keep them in repair; less chipping out of cupola, and consequent saving of lining, etc.

After twenty years' experience as a practical foundryman, having melted iron with all kinds of fuel, cupolas, and blast producers, I feel satisfied that a little slower melting, with hot liquid iron, with the consequent confidence which it inspires, is to be preferred to the quick-melted, low-tempered dull iron, which carries doubt in every ladle poured as to whether the piece is run up good and sharp in the corners, or cold shut, etc.

If the metal has to be churned or fed to prevent shrinkage, with the feeding metal too dull to keep the riser head open, then there is more doubt if the piece will be sound under the riser, etc., etc. Give me hot iron "every and all the time."

No. 462–88.

Can a water pyrometer be made to work successfully for high temperatures?

Mr. A. F. Nagle.—Perhaps it may not be inappropriate for me to say that the question is of my propounding, but in the form in which it is put above it does not solicit the information I want.

The present water pyrometer is not, strictly speaking, a water pyrometer. It is a metal pyrometer. The metal is heated by the hot gases and then cooled in water, and from the units of heat imparted to the water the original temperature of the hot metal is calculated.

It is now proposed that water be enclosed in a thin metal sphere and exposed directly to the hot gases for a given time, and from the heat imparted to the water in said time, with the aid of experimental data previously established, the surrounding temperature will be known.

One of the most important features of this type of pyrometer is that we can obtain temperatures above the fusing temperatures of metals.

It is thought desirable to limit the highest temperature of the water to 110° Fahr., and the least time of exposure to five minutes.

These proportions would make the error of observation in temperature, if the increase were 50° and the thermometer graduated to $\frac{1}{5}$ of a degree, about $\frac{1}{250}$; and the error in time, if the operation could be effected in three seconds, about $\frac{1}{100}$. The time is more difficult to obtain with accuracy than the temperature, but to prolong the time implies too large spheres. Are these safe limitations? If they are, the size of the spheres must be governed by the temperatures to be measured.

Probably spheres 4, 8, and 12 inches in diameter will answer all the requirements of practice.

What are the objections to such an instrument?

Can we obtain the experimental data which, if once obtained, can be applied to new conditions without serious errors resulting?

Let me relate some of the points to be established:

1. The absorbing power of the water depends somewhat upon its being in either a quiescent state or in motion; but for the low temperatures involved will that be a disturbing factor of any magnitude?

2. Is copper the best metal to use?

3. A standard thickness of metal should be established. Is No. 25 B. & S. gauge a safe thickness?

4. The sphere when full of water must be without any air space, and the screw plug for the insertion of a thermometer must be thin and hollow, so that no part may be overheated.

5. To protect the sphere from being ruptured by the expanding water, is it a sufficient safeguard simply to flatten it slightly at several points, or will it be necessary to provide some other means of compensation?

6. Is it better to attempt to preserve a bright or a dull surface on the spheres, and how greatly does one differ from the other in its absorbing power?

7. How greatly will the velocity of the hot gases affect the heat imparted to the sphere? While extremely low or very rapid velocities will undoubtedly have less or greater effect, still, within the limits prevailing in practice, will this be a factor so great as to necessitate the determination of the velocity in each case?

If it should prove that it will not be necessary to ascertain the velocity, then it will be a very simple instrument to use. We need but know the units of heat absorbed per minute by the standard size sphere, and, from data once correctly obtained, the temperature of the hot gases will be known at once. The only care required by the operator will be the correct reading of the temperatures and time.

I do not think it necessary to refer to the manner of conducting the experiments, or precautions to be observed, for they are such as will appear evident to experienced experimenters.

I am not aware of any experimental data extant which could be safely applied to this proposed instrument. Are there any such in existence?

In the foregoing I have disclosed my line of thought on this subject. If any one chooses to pursue it for himself I shall be much pleased, or if any one has anything to suggest which will aid the development of the instrument, or has already knowledge on the subject which would seem to make it useless to pursue the scheme any further, I shall be glad to have it disclosed.

Mr. David L. Barnes.—A water pyrometer such as is described would certainly be successful as a measure of temperature (if it were properly calibrated) for all those uses where the heat which was the origin of the temperature was radiant, or where the gases or liquid in contact with the pyrometer were not in motion, or when in the same uniform motion. The manner of exposure of the pyrometer bulb would affect the readings, so that while the pyrometer would give identical results under identical conditions, yet the results under different conditions would not be comparable. For any given class of work, undoubtedly a water pyrometer would be satisfactory when calibrated for the conditions under which it was to be used, but as a universal pyrometer I doubt its value. This may be said of it: When properly calibrated for a given set of conditions, it would undoubtedly be more delicate and accurate than any form of metallic pyrometer that I have ever seen.

Mr. Geo. H. Barrus.—In reply to the query upon the water pyrometer, I very much doubt whether such an instrument would be reliable, for the reason, which I fear, that the rate of transmission on the second and later trials, after the surfaces had become foul, might not be the same as it was at first.

Still, it would be an interesting thing to experiment on, and I should be glad to know what results can be obtained with this idea.

Mr. W. F. Durfee.—The subject is an interesting and important one; but it is my fear that a water pyrometer on the plan you suggest would not be a practical piece of apparatus. The pyrometer of Dr. Siemens depends substantially upon the principle which you suggest—the rise of temperature in a known weight of water; but in his instrument (or, rather, method) the element of time does not enter. There have been several attempts to measure high temperatures by the varying conductivity of a platinum rod or wire to an electric current, but I am not aware that any of them have been regarded as perfectly satisfactory. It would be very interesting to know just what kind of a pyrometer was used by King Nebuchadnezzar to determine with precision when his furnace was "heated seven times hotter than it was wont to be heated," but I fear that no delver among the potsherds of Babylonia will ever discover that instrument.

Mr. Henry M. Howe.—I fear that the indications of the pyrometer suggested would be extremely rough, probably less accurate than the judgment of an experienced eye. If few and isolated observations only are needed, the old method of mixtures leaves little to be desired—*e.g.*, dropping an iron ball (initially at the temperature which is to be determined) into a known weight of water, and noting the rise of temperature of the water. But here the enormous variations in the specific heat of iron must be taken into account. The Seger pyrometric cones offer another ready means of determining intermittently and roughly even the highest temperatures, and are used in the royal porcelain works at Berlin with great satisfaction.

But of all pyrometers, the most convenient and trustworthy under most conditions, where a large number of observations is to be made, I believe to be the Le Chatelier instrument. By it we get not only accurate but continuous and extremely rapid readings, up to the melting point of platinum, probably with an accuracy of 10° C. It measures the temperature by means of the thermo-electric current set up by the junction of a platinum and a rhodio-platinum wire, heated to the temperature to be determined, using a dead-beat galvanometer. Any good laboratory assistant can not only operate, but, after a little teaching, even calibrate it. Its cost is about $108.

Mr. Carleton W. Nason.—I think that one of the chief objections to a water pyrometer, if made as proposed, is as follows: The temperature of the shell holding the water will be approximately at the temperature of the water. It would be exposed to two conditions—one, in which the surrounding heated gas is quiet, or nearly so; the other, it may be travelling very rapidly. Now, as the pyrometer is dependent for its reading entirely upon the time in each case, the heating of the water will be very much greater in one than it will in the other case, and it would therefore be necessary to know at what velocity the gas currents were moving, which in many cases could not be obtained, and I should therefore regard a reading of that sort as unreliable and the proposed form of construction as not practical.

Prof. W. A. Rogers.—As the query is stated, the use of the water pyrometer is limited to high temperatures. If this instrument will indicate high temperatures correctly, it certainly ought to give correct indications for lower temperatures. I am quite sure that it will not do this. The question of the time required for water to pass from a constant air temperature to another temperature also constant is a very important consideration in this connection. I have made an extended series of observations for determining the time required for mercury, for a bar of steel having a length of 40 inches with a width of half an inch, and for varying volumes of water, with the following results:

	H.	M.
For mercury	0	30
For steel	5	30
For water having depths varying between ½ inch and 4 inches	17	00

While the greater part of the changes occur within moderate intervals of time, the changes which take place after the lapse of considerable time cannot be neglected.

It will be seen from these experiments that the slow thermal action in the case of water forbids its use as a means of measuring ordinary temperatures.

But there is another very important consideration in this connection. If I read the indication of a thermometer placed in a tumbler of water, I naturally conclude that this reading indicates the real temperature of the water. To a certain extent only is this true. Water never rises to the temperature of the surrounding air, on account of the cooling effect due to the evaporation which is continually taking place. The indicated temperature of water, therefore, is the temperature as affected by the evaporation going on at the particular instant when the observation is made. The cooling effect due to this cause is very slight in the neighborhood of the freezing point, but it is surprisingly great in the case of high temperatures. As the result of a large number of experiments, I find that at 50° Fahr. water is always 0.7° lower than the surrounding air, while at 85° it is 2.5° lower. At still higher temperatures it will probably be found that the cooling effect of evaporation increases very rapidly.

I should further say that, in my judgment, a simple metal pyrometer will give as accurate an indication of the temperature as can be obtained. I base this opinion upon the results which I have obtained with an interference reflectometer invented by Prof. Edward Morley, of Cleveland, Ohio. With the aid of this instrument I have found it a very simple matter to count the number of wave lengths of sodium light which correspond to the difference in lengths of bars of steel and bronze having a length of about 42 inches. The precision with which these optical comparisons can be made is very great. The limit of error is certainly not greater than one five-hundred-thousandths of an inch; and often closely approaches a millionth of an inch. The temperature of the bar was obtained from thermometers having their bulbs in contact with the vertical face of the bar.

It appears from these experiments that the bars follow the indications of the thermometers at least between 10° to 90° Fahr., within about one twenty-five-thousandths of an inch, that is, the observed difference in length agrees with the computed difference within this limit. All of my observations show that metals in the form of a thin sheet, having, for example, a thickness of two or three hundredths of an inch, take the temperature of the surrounding air rather more quickly than the mercurial thermometers of good quality.

CCCCLXIII.

MEMORIAL NOTICES OF MEMBERS DECEASED DURING THE YEAR.

FRANK W. PADGHAM.

Mr. Padgham was born December 30, 18[illegible]4, in Syracuse, N. Y. After graduating from the high school he served four years in the Straight Line Engine Co., in the machine and pattern shops and in the drawing room. In 1885 he entered Sibley College, whence he graduated in 1888 with the degree of M.E. From 1888 to 1890 he acted as chief draughtsman and engineer for the C. W. Hunt Co., of New York, and in October of the latter year he accepted a position as chief engineer of a division of the Pipe Line under the control of the National Transit Co, with headquarters at Oil City. This position involved the care of 250 miles of pipe and the machinery of the pumping stations. He connected himself with the Society at the Richmond meeting, November 11, 1890, but an illness of the typhoid type prevented his enjoying any of the advantages of membership. He passed away January 26, 1891.

FRANCIS C. BLAKE.

Mr. Blake was born February 23, 1854, in Connecticut. He graduated in 1876 from the Worcester Institute, having previously served three years as machinist in the Washburn & Moen machine shop. From 1877 to 1879 he held the position of adjunct professor of chemistry at Lafayette College, Easton, Pa., and from 1879 to 1881 he was assistant superintendent of the Pennsylvania Lead Co. From this latter date he was their superintendent and manager, engaged in controlling the work of smelting and refining silver lead bullion. Under his management extensive electric plants were put in for lighting and for the parting of silver and gold, and he also made important changes in the mechanical appliances for handling and treating the material of the works. He joined the Society at the New York meeting of 1886, and was an interested member until the time of his death. He had gone to Helena, Mont., on business for the firm, but a heavy cold caught during the severe weather developed into pneumonia and he died far away from home, February 21, 1891.

THOMAS PETERS CONANT.

Mr. Conant was born at Paris, France, July 11, 1860. He prepared for college at the Polytechnic Institute of Brooklyn, in which city his boyhood was spent, and received his professional training from the Columbia School of Mines, from which he graduated in June, 1882. He acted as assistant engineer for the Tilly Foster Iron Mines for a few months, but in the autumn of that year he began his preparation for his electric acquirements by a year's study in the laboratory of Thomas A. Edison. At the expiration of this time he undertook regularly the business of constructing electric-light stations for the Edison Company, and left his impress on various plants in Pennsylvania and New England. In February, 1886, he was sent, in the interests of the Electrical Accumulator Co., to England, and on his return was made their engineer, working for them upon accumulator installations until 1889. In July of that year he became superintendent of construction with the Edison Company, and at the time of his death was their district engineer in charge of the Eastern District, and his work comprised railway work as well as central stations and isolated plants. He died on the 24th day of February, 1891, after a painful illness of four months. He joined the Society as a junior at the Cleveland meeting in 1883, and was promoted to full membership at the New York meeting in 1889.

FRANKLIN E. WORCESTER.

Mr. Worcester was born at Albany, September 1, 1860. He was prepared for college at the Albany Academy, entering Yale College, to graduate therefrom in 1882. He then entered the Sheffield Scientific School, whence he graduated as Ph.B., part of the time having been spent in study in Europe. He received the full degree of Dynamic Engineer in 1886. His father being vice-president of the Michigan Central, as well as secretary of the New York Central & Hudson River R. R., he had an opportunity to enter the car shops at Jackson, Mich., of the Michigan Central roads, remaining there as machinist's apprentice from 1885 to 1887. After acting as locomotive fireman on the Michigan Central and Hudson River roads, he became, in February, 1888, assistant superintendent of motive power for the Duluth, South Shore & Atlantic R. R., with headquarters at Marquette, Mich. In 1889 he resigned this position, and after acting for a short time as inspector for the United States Government upon the new breakwater at Marquette, he connected himself with the Iron Bay Company, at first at Marquette and later at West Duluth. In July, 1890, he became general agent for the Montana region for firms making a specialty of mining machinery, which compelled him to make his headquarters at Helena. He was attacked during the winter with the severe pneumonia which appeared at that time almost to be an epidemic in Montana, and he died suddenly on March 3, 1891. He joined the Society at the Scranton meeting, October, 1888.

JAMES A. CROUTHERS.

James A. Crouthers was born at Astoria, N. Y., May 18, 1846. He served an apprenticeship and worked under instruction in the Novelty Iron Works, New York, and in other shops; entered the United States Navy as 2d assistant engineer, U. S. S. *Saco*, under J. Henderson as chief; and afterward served on U. S. S. *Chenango*, with Samuel Shear as chief, whom he afterward succeeded. He was also assistant engineer on U. S. S. *Hartford*, under Abraham Lefton as chief. In 1867 he was made chief engineer for ocean steamers, and as such was in charge of several merchant vessels. He connected himself with this Society at the Atlantic City meeting, in May, 1885, and at that time was agent and manager for certain firms engaged in the manufacture and sale of steam specialties. He was an active manager of the Board of the American Institute, of this city, and died at his home, January 15, 1891.

LUKE CHAPMAN.

Mr. Chapman was born at Willington, Tolland County, Conn., in 1835. He began his mechanical career in work connected with the manufacture of bayonets, under a contractor for the Collins Company, at Collinsville, Conn., early in the Civil War. Owing to his ability and judgment he was soon appointed inspector of the finished bayonets. In 1868 he was promoted to the position of master mechanic for the Collins Company, which position he held to the time of his death.

He improved and perfected many machines for machine forging and the working of metals in connection with the manufacture of edged tools, and obtained numerous patents for improvements on ploughs of several kinds. His mechanical life, covering a period of about thirty years, was all spent in the employ of the Collins Company, at Collinsville, Conn., where he died July 16, 1891, from a chronic disease from which he had suffered many years. He connected himself with the Society in its first year.

CCCCLXIV.

APPENDIX.

ADDRESS AT THE UNVEILING OF A PORTRAIT OF ALEXANDER L. HOLLEY,

LATE MEMBER OF THE SOCIETY, AND HONORARY MEMBER IN PERPETUITY, AS FOUNDER.

BY JAMES C. BAYLES, NEW YORK CITY.

(Delivered November 28, 1890.)

MR. PRESIDENT AND GENTLEMEN:

By a curious coincidence, which I cannot regard as other than very flattering, I come before you this evening charged with a double duty, which I shall have great pleasure in performing as well as I am able. That I have been selected is due to the surprising vitality of a tradition which has outlived by many years whatever basis of truth it may have had in the beginning. When I was young, and my time was devoted to the blithesome and variegated profession of journalism, I probably possessed a certain happy talent for so manipulating words as to conceal, like beads on a string, the slender thread of thought which gave them a semblance of continuity. But such a talent, if talent it be, cannot long survive the sharp focalization of thought and purpose upon the troublesome problems of a manufacturing business. If, therefore, my effort to serve you in lieu of an orator on this pleasant occasion shall result in disappointment to all concerned, please remember that to me, as to all creatures, applies the law of differentiation due to change of environment. My present environment is not conducive to literary fertility.

I have alluded to a double duty as constituting my service this evening. By request of the donor, I shall have pleasure in presenting to the Society a gift of great interest and value. By request of the management, I shall receive it on behalf of the Society. This arrangement, though unusual, has many advantages. When, on such occasions, one person makes a speech of presentation and another a speech of acceptance, it is almost inevitable that one will suffer from contrast with the other. It is usual, moreover, that one speech does not quite fit the other, each speaker having labored under a misapprehension as to what the other was likely to say. I had an interesting illustration of this at a recent meeting of the Institute of Mining Engineers in a western city. I was told in advance that the mayor of the city would welcome the institute; and as it was my duty, *ex officio*, to respond for the visiting membership, I first roughed out such a speech as his honor would be likely to make, and then framed a suitable response. But the mayor did not say what I had expected he would. I am told he never does. On the contrary, he told how, as a boy, he had mistaken one mineral for another, and dwelt at length upon the importance to the mining engineer of knowing "the difference between true silver and pyrites of lead," which so paralyzed me that I really could think of nothing appropriate to say in reply, unless, indeed, that for the same reason the mathematician should know the difference between calamus root and the cube root of minus zero.

In view of this and sundry similar experiences, I consider it a great cause for congratulation—or, rather, I consider that I have good reason to congratulate myself—that, having to make the speech of presentation, I am also to make the speech of acceptance.

On behalf of a gentle and estimable lady to whom many of us are very deeply and sincerely attached, and in whom we recognize a high exemplification of the virtues and graces of true womanhood—Mrs. Mary H. Bunker—I have great pleasure in presenting to the American Society of Mechanical Engineers the portrait of Alexander L. Holley, which will in future adorn the walls of our permanent home. It is not the wish of the giver that the occasion should be one of ceremony; much less that in conveying her gift to the Society I shall much exceed the mere statement of fact in the fewest and simplest words which will express it. To select from the many portraits of Holley those which best expressed his true personality; to shape the thought and guide the hand of the artist, who had never seen the one portrayed; and to place the finished work where it will be most valued and best preserved, is the last tribute of love to the memory of one who was, perhaps, better loved than any man of his time. That the portrait is faithful, those are agreed whose relations with Holley were the closest and most intimate. That it will perhaps disappoint many, at first glance, is to be expected. We have become familiar with a picture which, however charming, is not a portrait. The mischievous skill of the retoucher of negatives has idealized the face by obliterating the modelling, and from our long acquaintance with the photograph with which we are most familiar, we have probably forgotten how Holley really looked. As this portrait, with its strong modelling, is studied, memory will recall the living original with startling but pleasing vividness. What we shall miss no skill could reproduce—the indescribable "sweetness and light" which beamed from Holley's eyes and ever played upon his lips. We must let memory supply this deficiency. For those who knew him well this will not be difficult. We no longer sorrow for him. As the curtain falls from the face of the portrait, so from our eyes fades the mist of tears, and we see the man we loved so well as we knew him in the bright years of the brilliant professional work which is his noblest and most enduring monument.

Gentlemen of the Society of Mechanical Engineers, I have pleasure in assuring you that, in the opinion of the lady to whose liberality we are indebted for this most welcome and valued gift, no greater honor could be paid to the memory of Alexander Lyman Holley than that his portrait should be the first to be hung in what we hope will be the Parthenon of American Engineering. If I refrain from saying more, it is because I respect the wish so strongly expressed to me, that the presentation be as informal as possible, and that not the giver but the gift be the subject of our thought.

In receiving it on behalf of the Society, it is unnecessary that I should repeat the formal expression of thanks which has already been communicated to Mrs. Bunker by direction of the Council. Let me rather use this opportunity to point out, for the benefit of those of our membership who knew him not, and especially for those who represent a later generation of the profession—the young men who are already crowding us of earlier date into the dignified, but not always welcome, leisure suited to the afternoon of life—what his career and character teach those who are moved to study worthy examples and profit thereby.

It is not my intention to review his life work or trace the steps of a professional career so successful, and yet so different from what his youthful imagination pictured. This has been done many times by loving biographers. Dr. R. W. Raymond, in his masterly tribute to the memory of Holley, since embodied in the memorial volume, has told the story of his life with the skill of a master. By invitation of this Society, it was my pleasure and privilege at our memorial session to place upon record a consecutive narrative of his professional career. More recently, Mr. James Dredge, of London, in the memorable address before the representatives of the profession of two continents, has completed the record, and paid to Holley, as an engineer and a man, a tribute which seems to leave nothing unsaid.

We will not, however, find in the record of Holley's work, nor in the catalogue of his inventions, an explanation of his phenomenal career, still less of the spontaneous outpourings of sorrow evoked by the announcement of his death. Greater men than he have passed away, almost without attracting notice. Lives characterized by every endearing trait have failed to leave a lasting impress upon the generation which knew them. Why was Holley's career so like the flight of a meteor, and why did its termination leave the earth and the heavens dark for more people than the most famous of us know, or are known to? It was not alone because nature had so richly endowed him with talents and made him so attractive and lovable, or that he gave pleasure to those with whom he came in contact by his quick wit and his rare power of pleasing. To his unconquerable enthusiasm, triumphing over difficulties greater than most of us have ever known; his earnestness of purpose, which nothing could swerve or discourage; his patience and forgiveness when wronged, and his absolute honesty, more than to his talent and his power to charm, I attribute his hold upon the respect and affection of his contemporaries. The measure of success which came to him in the later years of his life, and which enabled him to realize the dearest wish of his heart, that his death might not bring deprivation to those he loved best, I attribute solely to the fact that he inspired and merited the absolute confidence of those who employed him professionally—men who had taken the print of the golden age, and into whose plans no sentiment entered. Whatever Holley undertook absorbed him. To that which he had to do he devoted himself with an utter disregard of all selfish considerations. To his wife, the sharer of his sorrows and disappointments, as of his joys and successes, ever his wisest counsellor and best friend, he said, many times and in many ways, "To those who employ me, I give the best I have." No one ever employed Holley's professional services who did not have reason to feel that to the work given him he brought zeal, enthusiasm, and fidelity. His was no half-hearted service, concealing the hope that in it, or behind it, lay something of present or future advantage for himself.

In his association with engineers, Holley made friends easily. But in these days of keen business and professional competition, something more than a winning smile and a happy facility of graceful speech and gracious compliment are needed to grapple to one's heart with hooks of steel the friends one makes. Holley did it by his unselfishness and honesty. Never seeking to advance his professional interests at the expense of another; always ready and eager to give credit where credit was due; always frank and truthful, and as ready to impart knowledge as to gain it, he won hearts wherever he went. Those who knew him were eager to be admitted to his confidence, and freely gave him theirs. Nothing mean or selfish or false was ever suspected of Holley; and in the intimate story of his life, there is nothing over which a friendly hand would wish to draw a veil. Probably no American engineer ever had more opportunities to turn to personal account the service he had engaged to render for others. It is because truth, loyalty, and sincerity were the governing principles of his life, that his career was so conspicuously brilliant, and that America and Europe have competed in doing honor to his memory.

Holley's standard of professional ethics was one which left no room for quibbles or questions. It was the unwritten law of duty, which admits of no misinterpretation. Personal advantage and loyalty to his employers were to him utterly incompatible, and his clear mind rejected the sophistries by which weaker and more plastic natures are corrupted. The subtle temptations which shrewd men of business too often devise to influence the judgment of the engineer had for him no attractions. However disguised, the bribe was, in his judgment, a dishonor; its tender, an insult. No man made merchandise of his reputation, or influenced his judgment by appeals to his self-interest. On these high qualities securely rests the reputation which has survived him, and these alone would warrant the honors which have been paid to his cherished memory.

As a member of this Society, one of the first on its original roll, a participant in the initial steps looking to its organization, and—as I may mention for the information of one of my friends—its first secretary, I am proud of its success, proud that it ranks A. L. Holley among its founders, and proud that its walls are adorned with his portrait. On behalf of the Society, I receive the gift with sincere gratitude and profound appreciation.

The love which consecrates the gift makes it, for us, as sacred as a devotion upon an altar. While those of us who knew and loved him live, its presence will make our home more home-like, and be at all times an inspiration. When we are gone, let us hope it will still be cherished for our sakes, as well as for his; and when we look at it, as we shall often be moved to do, let us remember that the man whom we delighted to honor living, and to remember dead, as the first among American engineers, was no gloomy recluse, impatient of the wholesome pleasures which rest mind and body, but one whose joyous nature welcomed relaxation, and who could surround even the most serious concerns of life with a luminous photosphere of scintillating wit. If, as I hope, we shall find this house the home of pleasure as well as a place for serious work, the memory of Holley will never be in discord with our mood.

GENERAL INDEX TO VOLUME XII.

[NOTE.—Titles of papers are printed in *italics*; names of authors and participants in debate are printed in SMALL CAPS.]