TRANSACTIONS

OF THE

AMERICAN SOCIETY

OF

MECHANICAL ENGINEERS.

VOL. XIII.

XXIVTH MEETING, NEW YORK CITY, NOV., 1891.
XXVTH MEETING, SAN FRANCISCO, CAL., MAY, 1892.

NEW YORK CITY:
PUBLISHED BY THE SOCIETY,
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NO. 12 WEST 31ST STREET.
1892.

OFFICERS

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

1891—1892.

FORMING THE STATUTORY COUNCIL.

PRESIDENT.

CHARLES H. LORING..Brooklyn, N. Y.

VICE-PRESIDENTS.

STEPHEN W. BALDWIN..New York City.
JNO. F. PANKHURST..Cleveland, Ohio.
ALEXANDER GORDON..Hamilton, Ohio.

Terms expire at Annual Meeting of 1892.

GEO. I. ALDEN..Worcester, Mass.
E. F. C. DAVIS..Richmond, Va.
IRVING M. SCOTT..San Francisco, Cal.

Terms expire at Annual Meeting of 1893.

MANAGERS.

JAS. E. DENTON..Hoboken, N. J.
CARLETON W. NASON..New York City.
H. H. WESTINGHOUSE..Pittsburgh, Pa.

Terms expire at Annual Meeting of 1892.

ANDREW FLETCHER..New York City.
WORCESTER R. WARNER..Cleveland, Ohio.
COLEMAN SELLERS, JR..Philadelphia, Pa.

Terms expire at Annual Meeting of 1893.

JAMES M. DODGE..Philadelphia, Pa.
ROBERT FORSYTH..Chicago, Ill.
JESSE M. SMITH..Detroit, Mich.

Terms expire at Annual Meeting of 1894.

TREASURER.

WM. H. WILEY..............................58 East 10th St., New York City.

SECRETARY.

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

HONORARY COUNCILLORS.

Past Presidents of the Society.

R. H. THURSTON	1880—1882	Ithaca, N. Y.
E. D. LEAVITT	1882—1883	Cambridgeport, Mass.
JOHN E. SWEET	1883—1884	Syracuse, N. Y.
J. F. HOLLOWAY	1884—1885	New York City.
COLEMAN SELLERS	1885—1886	Philadelphia, Pa.
GEO. H. BABCOCK	1886—1887	New York City.
HORACE SEE	1887—1888	New York City.
HENRY R. TOWNE	1888—1889	Stamford, Conn.
OBERLIN SMITH	1889—1890	Bridgeton, N. J.
ROBERT W. HUNT	1890—1891	Chicago, Ill.

[NOTE.—The former Presidents of the Society are members of the Council for life or during their retention of active membership in the Society.]

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.
ROBT. W. HUNT	"	Nov. 14th, 1890—Nov. 20th, 1891.
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. GILLMORE, 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, CHAS. B. RICHARDS, JOEL SHARP, GEO. W. WEEKS, and DE VOLSON WOOD.

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, MORRIS SELLERS, FRANK H. BALL, GEO. M. BOND, and WM. FORSYTH.

NOTE.

THE increasing bulk of the Annual Volume of *Transactions* has induced the Publication Committee to recommend the discontinuance of the practice of inserting 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, 1892, as the second edition of the Twelfth Catalogue. The following summary taken records the numbers in each grade :

Honorary members	17
Life members	21
Members	1227
Associates	62
Juniors	194
Total	1500

AMENDED.

[*November* 19*th*, 1889—*November* 17*th*, 1891.]

RULES

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

[*Adopted November* 5*th*, 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 \$25, and their annual dues shall be \$15, payable in advance. The initiation fee of juniors shall be \$15, and their annual dues \$10, payable in advance. A junior, being promoted to full membership, shall pay an additional initiation fee of \$10. Any member or associate may become, by the payment of \$200 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 unand 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.

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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.

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LIST OF ILLUSTRATIONS, VOLUME XIII.

PAPERS

OF THE

NEW YORK MEETING

(XXIVth)

NOVEMBER 16th TO 19th, 1891.

BEING ALSO THE TWELFTH ANNUAL MEETING OF THE SOCIETY.

CCCCLXV.

PROCEEDINGS

OF THE

NEW YORK MEETING

(XXIVth)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS,

November 16th to 19th, 1891.

EXECUTIVE COMMITTEE:—Chas. H. Loring, *Chairman;* J. F. Holloway, Carleton W. Nason, Wm. H. Wiley, F. R. Hutton.

FINANCE:—S. W. Baldwin, J. B. Edson, John C. Kafer, Wm. J. Logan.

EXCURSIONS AND ENTERTAINMENTS:—Jas. F. Lewis, John Thomson, E. V. Clemens, G. C. Henning, Jas. A. Denton, E. P. Stratton, E. A. Stevens, Stevenson Taylor.

RECEPTION:—Henry Morton, Andrew Carnegie, John Stanton, Andrew Fletcher, Chas. E. Emery, S. W. Baldwin, T. C. Clarke, C. W. Hunt, Charles Paine, C. C. Worthington, Chas. Kirchhoff, W. C. Kerr, William Kent, Horace B. Miller, F. Meriam Wheeler, W. H. Adams, W. S. Doran, T. Spencer Miller, C. M. Wales, H. DeB. Parsons.

FIRST DAY. MONDAY, NOVEMBER 16TH.

The Twenty-fourth Meeting of the American Society of Mechanical Engineers, which was also its Twelfth Annual Meeting, was convened in the City of New York pursuant to call. The business sessions were held in the house occupied by the Society as tenants of the Mechanical Engineers' Library Association; this meeting being the first which had been held since the title had passed from the former owners, the New York Academy of Medicine. The opening session, which was a mixed social and professional session, was found to require more accommodation than the cozy auditorium of the house afforded, and for this reason was convened in Sherry's Assembly Hall, corner of Thirty-seventh Street and Fifth Avenue.

The meeting was called to order by Chas. H. Loring, Chairman of the Local Committee, who introduced the President of the Society, Mr. Robert W. Hunt, of Chicago, who delivered his Annual Address, entitled "The Evolution of American Rolling Mills." At the close of this address, a conversazione and social reunion, with refreshments, was held in the hall, over three hundred persons being present, among whom was a large number of ladies.

The Tellers to count the ballots cast for officers of the Society for the ensuing year were appointed by the President, under Article 34 of the Rules, before the session adjourned.

SECOND DAY. TUESDAY, NOVEMBER 17TH.

The first session for business was convened in the auditorium of the Society's house, No. 12 West Thirty-first Street, at 10 A.M.

The Secretary's register at headquarters showed the following members in attendance during the Convention:

Alden, George I.	Polyt. Inst., Worcester, Mass.
Allderdice, W. H.	Washington, D. C.
Anthony, Gardner C.	Providence, R. I.
Archer, Edward R.	Richmond, Va.
Ashworth, Daniel	Pittsburgh, Pa.
Bailey, C. L.	Washington, D. C.
Bailey, W. H.	New York City.
Baldwin, Stephen W.	New York City.
Ball, Frank H.	New York City.
Bang, H. A.	New York City.
Bardwell, A. F.	Stamford, Conn.
Barnes, A. T.	Boston, Mass.
Barnes, D. L.	Chicago, Ill.
Basford, Geo. M.	Milwaukee, Wis.
Bates, Edw'd P.	Syracuse, N. Y.
Beach, C. S.	Bennington, Vt.
Beardsley, Arthur	Swarthmore, Pa.
Benham, E. B.	Providence, R. I.
Betts, Alfred	Wilmington, Del.
Billings, Chas. E.	Hartford, Conn.
Binsse, Henry Leon	Newark, N. J.
Bird, W. W.	Cambridge, Mass.
Blankenship, R. M.	Richmond, Va.
Bond, Geo. M.	Hartford, Conn.
Borden, Thomas J.	Fall River, Mass.
Boyd, John T.	Erie, Pa.
Brady, James	Brooklyn, N. Y.
Bristol, W. H.	Hoboken, N. J.

Brooks, Edwin C ... .Cambridge, Mass.
Brotherhood, Fred.......................Beaufort, S. C.
Brown, Robert A.........................New Haven, Conn.
Brown, R. S............ ....New Britain, Conn.
Bulkley, Henry W.... ....New York City.
Bullock, E. R ....Pawtucket, R. I.
Bullock, M. C............ ....Chicago, Ill.
Caldwell, A. J....................... ..New York City.
Camacho, Leopoldo A................. ...New York City.
Carr, C. A..............................Washington, D. C.
Carse, D. B................... ....Chicago, Ill.
Cartwright, Robert......................Rochester, N. Y.
Cary, Albert A..........................New York City.
Cavanagh, Joseph........................Philadelphia, Pa.
Chamberlain, P. M.......................Waynesboro', Pa.
Cheney, Walter L........................Meriden, Conn.
Christiansen, Alfred.............. ......West Troy, N. Y.
Church, E. D., Jr.......................Brooklyn, N. Y.
Cité, Jos. D............................Fishkill Landing, N. Y.
Clarke, Chas. L............ ....New York City.
Clarke, Samuel J........................New York City.
Clemens, Ernest V.......................New York City.
Cole, J. Wendell........................Columbus, Ohio.
Cole, L. W.............. ....So. Shaftesbury, Vt.
Comly, Geo. N...........................Wilmington, Del.
Conrad, H. V............................No. Tarrytown, N. Y.
Corbett, Chas. L........................Brooklyn, N. Y.
Covell, H. N............................New York City.
Cruikshank, Barton. ....New York City.
Cullingworth, Geo. R....................New York City.
Dallett, W. P...........................Philadelphia, Pa.
Darling, Edward A.......................New York City.
Davis, Chas. H..........................New York City
Davis, Isaac H..........................New York City.
Dean, Francis W.........................Boston, Mass.
Deane, Charles P ....Holyoke, Mass.
Delaney, Alexander......................Richmond, Va.
Denton, James E...... ....Hoboken, N. J.
Dinkel, Geo.; Jr........................New York City.
Dixon, C. A....... ....Newburgh, N. Y.
Dockam, E. H............................New York City.
Dodge, Jas. M....... ....Philadelphia, Pa.
Doran, W. S.............................New York City.
Draper, T. W. Morgan....................Portsmouth, Va.
Drewett, Wm. A..........................Brooklyn, N. Y.
Dripps, Wm. A...........................Philadelphia, Pa.
Durfee, W. F............................New York City.
Edson, Jarvis B.........................New York City.
Edwards, V. E...........................Worcester, Mass.
Ehlers, Peter...........................Albany, N. Y.
Emery, Charles E........................New York City.

Engel, Louis G........................Brooklyn, N. Y.
Ewer, Roland G........................Natrona, Pa.
Faber du Faur, A......................New York City.
Falkenau, Arthur......................Philadelphia, Pa.
Field, Cornelius J....................New York City.
Fish, Charles H.......................Manchester, N. H.
Fladd, Fred'k C.......................New York City.
Flagg, S. G., Jr......................Philadelphia, Pa.
Fleming, Wm. R........................New York City.
Fletcher, Andrew......................Hoboken, N. J.
Fletcher, W. H........................Hoboken, N. J.
Foster, Chas. F.......................St. Louis, Mo.
Fowler, Geo. L........................New York City.
Francis, Harry C......................Philadelphia, Pa.
Francis, W. H.........................Philadelphia, Pa.
Freeman, John R.......................Boston, Mass.
Fritz, John...........................Bethlehem, Pa.
Fry, C. A.............................New York City.
Fryer, Geo. G.........................Syracuse, N. Y.
Gantt, Henry L........................Nicetown, Pa.
Geoghegan, Stephen J..................New York City.
Gibson, Wm............................New York City.
Gill, John L..........................Philadelphia, Pa.
Gilmore, Robert J.....................Providence, R. I.
Glasser, Chas. H......................Ironwood, Mich.
Goodale, A. M.........................Waltham, Mass.
Gould, W. V...........................Norwich, Conn.
Graves, Erwin.........................Camden, N. J.
Gregory, Wm...........................New York City.
Grimm, Paul H.........................Glen Cove, L. I., N. Y.
Hague, Charles A......................New York City.
Hammett, Hiram G......................Troy, N. Y.
Hawkins, John T.......................Taunton, Mass.
Hayward, Fred. H......................New York City.
Hazard, Vincent G.....................Wilmington, Del.
Henderson, Alex.......................Boston, Mass.
Henning, Gustavus C...................New York City.
Hershey, Martin E.....................Harrisburg, Pa.
Higgins, M. P.........................Worcester, Mass.
Hoffecker, W. L.......................Elizabeth, N. J.
Holloway, H. F........................New York City.
Holloway, J. F........................New York City.
Hoppes, John J........................Springfield, Ohio.
Hough, D. L...........................Camden, N. J.
Howell, Edward I. H...................Philadelphia, Pa.
Humphrey, John........................Keene, N. H.
Humphreys, Alex C.....................Philadelphia, Pa.
Hunt, C. W............................New York City.
Hunt, Robert W., *President*..........Chicago, Ill.
Huson, Winfield Scott.................Taunton, Mass.
Hutton, F. R., *Secretary*............New York City.

Hyde, Charles E. . . . . Bath, Me.
Idell, Frank E. . . . . New York City.
Jacobi, Albert W. . . . . Newark, N. J.
Jacobus, D. S. . . . . Hoboken, N. J.
Johnson, A. E. . . . . Stamford, Conn.
Kafer, John C. . . . . New York City.
Kent, Wm . . . . New York City.
King, Chas. C. . . . . West New Brighton, N. Y.
Kirchhoff, Charles. . . . . New York.
Kraus, Julius. . . . . Paterson, N. J.
Laforge, F. H. . . . . Waterbury, Conn.
Laird, John A. . . . . St. Louis, Mo.
Lane, Harry M. . . . . Cincinnati, Ohio.
Lanza, Gaetano. . . . . Boston, Mass.
Laval, Geo. de. . . . . Warren, Mass.
Lemoine, Louis R. . . . . Philadelphia, Pa.
Leonard, Samuel H. . . . . Baltimore, Md.
Le Van, W. B. . . . . Philadelphia, Pa.
Lewis, James F. . . . . New York City.
Locke, Sylvanus D. . . . . Hoosick Falls, N. Y.
Lord, Hiram F. . . . . Yonkers, N. Y.
Loring, Charles H. . . . . Brooklyn, N. Y.
Low, F. R. . . . . New York City.
Lyall, W. L. . . . . New York City.
MacFarren, S. J. . . . . Pittsburgh, Pa.
McBride, Jas. . . . . Brooklyn, N. Y.
McElroy, Samuel. . . . . New York City.
McEwen, J. H. . . . . Ridgway, Pa.
McRae, John D. . . . . Baldwinsville, N. Y.
Main, Charles T. . . . . Lawrence, Mass.
Manning, Chas. H. . . . . Manchester, N. H.
Meyer, J. G. A. . . . . New York City.
Miller, Alex. . . . . Jersey City, N. J.
Miller, Fred. J. . . . . New York City.
Miller, Horace B. . . . . New York City.
Miller, T. Spencer. . . . . New York City.
Monaghan, Wm. F. . . . . New York City.
Montgomery, H. M . . . . Boston, Mass.
Moore, D. G. . . . . Elizabeth, N. J.
Morgan, Chas. H. . . . . Worcester, Mass.
Morse, Chas. M. . . . . Buffalo, N. Y.
Moulthrop, Leslie . . . . New Haven, Conn.
Müller, Teile H. . . . . Philadelphia, Pa.
Mumford, E. H. . . . . Detroit, Mich.
Murphy, Edward J. . . . . Hartford, Conn.
Murray, S. W. . . . . Milton, Pa.
Nash, Lewis H. . . . . So. Norwalk, Conn.
Nason, Carleton W. . . . . New York City.
Nicoll, Chas. H. . . . . Newark, N. J.
Nicolls, Wm. J. . . . . Philadelphia, Pa.
Norris, R. Van A. . . . . Wilkesbarre, Pa.

O'Connell, John C........................Montgomery, Ala.
Odell, W. H..............................New York City.
Parks, Edward H.......... ....Providence, R. I.
Parsons, Harry de B......................New York City.
Partridge, Wm. E................. ....New York City.
Payne, David W..........................Elmira, N. Y.
Payne, S. F..............................Naugatuck, Conn.
Pearson, Wm. Anson, Jr..................Scranton, Pa.
Percival, G. S............................New York City.
Phillips, Franklin.... ....Newark, N. J.
Phillips, George H............. ....Newark, N. J.
Philp, C. von............................Bethlehem, Pa.
Pitkin, Stephen H................. ......Akron, Ohio.
Platt, Geo. H............................New York City.
Platt, Jos. C.............................Waterford, N. Y.
Porter, Chas. T..........................Montclair, N. J.
Post, John, Jr...........................Boston, Mass.
Pusey, Chas. W.... ... .Wilmington, Del.
Redwood, I. I............................Brooklyn, N. Y.
Reeve, C. D..............................Bridgeton, N. J.
Rice, Richard H..........................Providence, R. I.
Richards, Francis H......................Hartford, Conn.
Richmond, George........................New York City.
Riesenberger, A..........................Hoboken, N. J.
Ridsdale, T. W...........................New York City.
Roberts, Percival ... .Pencoyd, Pa.
Robinson, J. M...........................New York City.
Rogers, W. S.............................West Troy, N. Y.
Ross, E. L................................Indian Orchard, Mass.
Rowland, Amory E.......................New Haven, Conn.
Rowland, C. B............................Brooklyn, N. Y.
Rowland, George..........................Brooklyn, N. Y.
Rowland, Thomas F., Jr...... ....Brooklyn, N. Y.
Sargent, John W..........................Scranton, Pa.
Scheffler, F. A...........................Cleveland, Ohio.
Schuhmann, George.......................Philadelphia, Pa.
Schutte, Louis...........................Philadelphia, Pa.
Schwanhausser, William..................Brooklyn, N. Y.
See, Horace..............................New York City.
Seemuller, H. E..........................Stamford, Conn.
Sewall, M. W.............................New York City.
Shaw, T. Jackson..................... ...Wilmington, Del.
Shirrell, David.... ....Richmond, Va.
Sinclair, Angus...........................New York City.
Slater, Alpheus B............. ....Providence, R. I.
Smith, Charles F......... ....Brooklyn, N. Y.
Smith, George H..........................Providence, R. I.
Smith, Oberlin...........................Bridgeton, N. J.
Spangler, H. W...........................Philadelphia, Pa.
Spies, A..................................New York City.
Stearns, Albert... ....Brooklyn, N. Y.

Stiles, Norman C........................Middletown, Conn.
Stillman, Francis Hill ....New York City.
Stone, W. M...............................Hartford, Conn.
Stratton, E. P............................College Point, N. Y.
Stratton, W. H............................Providence, R. I.
Stryker, Francis ....Westerly, R. I.
Suplee, Henry H...........................Stamford, Conn.
Swasey, Ambrose...........................Cleveland, Ohio.
Swenson, W................................New York City.
Tabor, Harris.............................New York City.
Taylor, John T............................New York City.
Taylor, Stevenson.........................New York City.
Thomas, Charles W.........................New York City.
Thompson, Edward P........................New York City.
Thomson, John.............................New York City.
Thorne, W. H..............................Philadelphia, Pa.
Thurston, Robert H........................Ithaca, N. Y.
Tobey, George A...........................Newark, N. J.
Tompkins, S...............................Charlottesville, Va.
Torrey, H. G..............................New York City.
Trask, G. F. D............................Staten Island, N. Y.
Trautwein, Alfred P.......................Carbondale, Pa.
Tremaine, E. G............................Brooklyn, N. Y.
Trump, C. N...............................Wilmington, Del.
Tucker, W. B ....Elizabeth, N. J.
Vandegrift, J. A..........................New York City.
Victorin, A...............................West Troy, N. Y.
Voorhees, Philip R........................New York City.
Vose, Clarence............................New York City.
Wagner, J. R..............................Drifton, Pa.
Wales, Charles M..........................New York City.
Walworth, Arthur C........................Boston, Mass.
Ware, Justin R............................Worcester, Mass.
Warren, B. H..............................Stamford, Conn.
Watson, William...........................Boston, Mass.
Webster, John H...........................Boston, Mass.
Weeks, George W...........................Clinton, Mass.
Weightman, William H......................New York City.
Wellington, A. M..........................New York City.
Wellman, Samuel T.........................Thurlow, Pa.
Wells, J. Leland..........................New York City.
Wheeler, F. Meriam........................New York City.
Wheelock, Jerome..........................Worcester, Mass.
White, H. C...............................New York City.
Whitehead, George E ....Providence, R. I.
Whitney, B. D.............................Winchendon, Mass.
Whitney, W. M ....Winchendon, Mass.
Whittier, Charles.........................Boston, Mass.
Wiggin, W. H..............................Worcester, Mass.
Wiley, William H., *Treasurer*............New York City.
Wilkin, W. M..............................Erie, Pa.

Willcox, Charles H.	New York City.
Willcox, E. B.	Brooklyn, N. Y.
Wilson, J. E.	New York City.
Winship, J. G.	New York City.
Wittman, N. B.	Birdsboro', Pa.
Wolff, Alfred R.	New York City.
Wood, De Volson	Hoboken, N. J.
Wood, Matthew P.	New York City.
Wood, Walter	Philadelphia, Pa.
Woodbury, C. J. H.	Boston, Mass.
Woolson, O. C.	Newark, N. J.
Total	298.

There was also a number of guests present, and the ladies' delegation was over 100.

The first order of business was the report of the Council, presented by the secretary as follows:

ANNUAL REPORT OF THE COUNCIL.

The Council beg leave to present its annual report as follows:

The Council has held nine 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 passing upon applications for membership.

Applications have been received from a number of libraries requesting that such institutions might be put upon the free list of those who received our volumes of *Transactions* annually as issued, and in many cases asking for the back volumes. These requests have usually been granted, to begin with the less scarce volumes later than No. IV.

The matter was presented to the Council early in the season of holding informal reunions at the house of the Society, for the purposes of social and professional enjoyment, such reunions to be under the auspices and to be controlled by the members of the Society who might be able to attend them. By this means the entire expense of such evenings would be independent of the Society's funds, and be to no extent a charge upon its regular income. The first of these meetings was called after the Richmond Convention, and was made the occasion for the unveiling of the portrait of Alexander L. Holley. During the winter similar ones were held, the subjects forming the centre of the evenings' entertainments being Robert Fulton, Growth of the Railroad, Egypt, New and Old; Electricity previous to Galvani, and on the last evening the oil portrait of Henry R. Worthington, the gift of his son, C. C. Worthington, was made the concluding feature. The Society's auditorium, therefore, has hanging upon its walls the portraits of the two gentlemen who were created in 1882 by vote "Honorary Members in Perpetuity," as founders of the Society.

By gift of Prof. Thos. Egleston, Ph.D., the Society became the owner of a handsome mahogany dining-table belonging to the Colonial period of American history, and once the property of Robert Fulton, and believed to have been his dining-table. It consists of three parts, forming a central square and two oval ends. There has been inset into the top of this table a polished brass plate with the following inscription: "This table was presented to Mrs. Egleston House by Robert Fulton, Engineer. It passed from her hands to those of Geo. W. Egleston, and from him to his brother, Thos. Egleston, who presented it to the American Society of Mechanical Engineers, January, 1891."

At the request of Professor Egleston, the Council has appointed a committee on Standard Units to serve as a standing committee while its duties remain before it; and has appointed as such committee Messrs. Egleston, Thurston, and Bond.

The desirability of protecting still further the procedure of candidacy to membership in the Society has induced the Council to direct a committee to draft a new form of application blank, the object being first, to add dignity and distinction to the procedure, and second, to favor the policy of having the candidates proposed by their friends rather than that they should come before the Council as the result of their own desire; third, to emphasize the qualifications necessary for membership, and to secure from the proposers more detailed statements upon those qualifications, and fourth, to render the blank more convenient both as to size and with reference to the Society's rules and precedents.

The Council has received a gift of $513 from the committee of local members at Providence, it being the surplus after the local expenses chargeable to the fund for the meeting had been paid. This fund has been expended under the direction of the Library Association, and the books purchased are to be known as the Providence Gift.

At the request of the Mechanical Engineer of the Columbian Exposition a committee has been appointed to formulate a series of engineering tests to be made upon the exhibits at the Fair. In view of the fact that there may be a question of jurisdiction between the Mechanical Engineer's office and that of a Committee on Awards appointed by the office of the Director General, the action of the committee is held in abeyance until further action is taken in Chicago.

The Council has undertaken the consideration of two important amendments to the Rules, which are to come up for discussion at this meeting, and in view of the important nature of these amendments, which affect the rate of the dues, the Council has directed that a circular should be issued to all members, stating the reasons inducing the Council to give favorable consideration to these amendments, and to elicit from the entire membership an expression of opinion on these points, so that those who attend the meeting, and upon whose votes the decision must rest, might be advised as to the sentiment of the Society at large.

Similar action was taken by the Council in reference to the proposition to hold the spring meeting of 1892 in the city of San Francisco. It was found that the entire number heard from was 796, and of this number 723 favored the plan of the San Francisco meeting, and but 73 votes were recorded against it, an affirmative decision of over ten to one. The Council, therefore, has accepted the invitation that the next meeting should be held in San Francisco.

Pursuant to the action of the Society, the Council has appointed as its Committee on Standard Flanges for Valves and Pipes, etc., Messrs. Nason, Caldwell, Jarecki, Frank H. Ball, and John E. Sweet.

The Council, feeling also that it was not too early to initiate steps looking towards the invitation to foreign engineers who had been the hosts of the Society on the occasion of its European trip in 1889, to allow the American societies to reciprocate this attention, has directed that a committee of conference be requested from the four engineering societies, each society to supply three members, as a committee to consider the question and arrange for the transmittal of a formal invitation as soon as convenient. The report of the Council's committee and draft of the letter which requests coöperation are appended to this report.

The question of an Engineering Headquarters and Congress at the Columbian Fair has been before the Council at several sessions, the last communication being dated October 16th. The Council has directed that action on this matter be deferred for the present.

The Council has passed favorably during the year upon 323 applications for membership in the several grades.

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

Frank W. Padgham,
Francis C. Blake,
Thos. P. Conant,
Franklin E. Worcester,
James A. Crouthers,
Luke Chapman,
Geo. B. Reynolds,
Geo. A. Porter,
Alfred C. Hobbs.

The present membership of the Society, including those joining at this meeting, and favorably acted upon by the voting membership, is 1,443, and is distributed among the grades as follows:

Honorary Members	18
Life Members	13
Members	1,190
Associates	56
Juniors	166
	1,443

The Council would also present the report of its Tellers of Election, as follows:

REPORT OF TELLERS OF ELECTION.

The undersigned were appointed a committee of the Council to act as Tellers (under Rule 13), to scrutinize and count the ballots cast for and against the candidates proposed for membership in the American Society of Mechanical Engineers, and seeking election before the XXIVth Meeting, New York, 1891.

They have met upon the designated days, in the office of the Society, and have proceeded to the discharge of their duty. They would certify for formal insertion in the records of the

Society, to the persons whose names appear on the appended list, to their respective grades.

There were 484 votes cast of the blue ballot, of which 25 were thrown out because of informalities (the members voting having neglected to indorse the sealed envelope).

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

NEW YORK, *September* 23d, 1891.

There were 536 votes cast on the violet ballot, of which 20 were thrown out by reason of informalities, the voter not having affixed his name thereto.

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

AS MEMBERS.

Abbott, Wm. L.
Angus, Robert.
Baker, Chas. F.
Bates, Edward P.
Beaman, Elmer A.
Behr, Hans C.
Blessing, James H.
Bourne, Stephen N.
Bowden, James H.
Boynton, E. C., Jr.
Bryan, Wm. H.
Chase, Wm. L.
Cobb, George H.
Cook, E. J.
Corthell, Elmer L.
Cronise, Ernest S.
Cushing, George H.
Dodds, William E.
Dunlap, Frank M.
Evans, Quimby N.
Flach, Emil J.
Flad, Edward.
Flagg, Stanley G., Jr.
Ford, Edward L.
Fryer, George G.
Gemmell, James C. R.
Gillis, Albert R.
Grieves, E. W.
Hartness, James
Haskins, John F.
Henderson, George.
Herr, Edwin M.
Hibbard, John D.
Holloway, H. F.
Jenks, L. H. H.
Kaven, Moses B.
Kendricken, Paul H.
Lattin, Judson.
Lewis, G. T.
Lindstrom, Charles.
McKinney, W. S.
Meatz, John T.
Meier, E. D.
Nash, Lewis L.
Pierce, Herbert H. D.
Platt, Joseph C.
Rand, Addison C.
Raqué, Philip E.
Reber, Louis E.
Richards, Frank.
Richards, John.
Robinson, Edward P.
Rockwood, Geo. I.
Ross, Cornele G.
Rowland, Amory E.
Ruttmann, Ferd. S.
Siner, J. B.
Slack, Joseph A.
Torrey, Willard C.
Vandegrift, James A.
Walker, Thomas C.
Washburn, William S.
White, Edward F.
Wilson, J. Fred.
Wood, William M.

FOR PROMOTION TO FULL MEMBERS.

Armstrong, E. J.
Bardwell, A. F.
Basford, Geo. M.
Bird, Wm. W.
Merrill, Allyne L.
Miller, Edward F.
Norris, R. Van A.
Stone, Wilbur M.

FOR PROMOTION TO ASSOCIATE.

Dockam, Edward H.

AS ASSOCIATES.

Bagaley, Ralph.	Hopkins, Harvey L.	Seemuller, H. B.

AS JUNIORS.

Ackerman, William S.	DePuy, C. E.	Morgan, Paul B.
Adams, David.	Doty, P. A.	Norris, J. H.
Atwater, Chris. G.	Estes, William W.	Noyes, Henry F.
Beach, Edward L.	Foster, Jed.	Payne, Stephen H.
Bird, John D.	Hawkins, William C.	Petterson, Alfred.
Brand, Horace L.	Hay, Alex. S.	Preston, Geo. B.
Buttolph, Benjamin G.	Hopton, Walter E.	Robinson, Edward.
Cole, Winthrop.	Huff, S. W.	Schoenborn, W. E.
Collins, B. R. T.	Jones, Harry P.	Seymour, Joseph W.
Cowan, F. B.	Kress, James E.	Shapleigh, William C.
Davis, Charles A.	Moeller, Franklin.	Whitaker, Henry E.
	Wildman, Leonard D.	

APPENDIX TO THE REPORT OF THE COUNCIL.

(Copy.)

TO THE DIRECTORS OF THE AMERICAN SOCIETY OF CIVIL ENGINEERS.

By direction of the Council of the American Society of Mechanical Engineers, we have the honor to submit the following for your consideration :

In the summer of 1889 a joint party of American engineers visited England and France, and were the recipients there of large hospitality at the hands of the English and French engineers. The American visitors, individually and collectively, repeatedly expressed their earnest desire that the visit should be reciprocated by their hosts. The occurrence of the World's Columbian Exposition in 1893 will evidently be the most fitting time for such visit, and it is undoubtedly the desire and expectation of American engineers generally that an invitation should be extended to foreign engineers to visit the United States during the period of the Exposition.

The hospitalities of our English and French hosts in 1889 were extended equally to the representatives of our four national societies of engineers, and it would seem fitting, therefore, that the return invitation should emanate from some body representative of the same four societies. Guided by the former experience, the Council of this Society has thought it desirable to give early consideration to the matter, for which purpose a sub-committee was recently appointed to consider and report thereon, a copy of whose report is appended hereto. The recommendations of that report have been approved by the Council of this Society, which hereby invites you to unite in the appointment of members upon a joint Committee of Conference. The suggestion that the commissioners from each society should be chosen from a list of those who have held the presidential or vice-presidential office commends itself to the council and will doubtless be followed by it. Should this suggestion be generally accepted, the joint commission would be wholly constituted from among the past-presidents and vice-presidents of the several societies.

If the plan thus outlined commends itself to your approval, will you be kind enough to notify the Secretary of this Society of any action you may take, and of the persons appointed as commissioners, should such appointment be made, in order that, at the earliest convenient opportunity, arrangements may be made for the first meeting of conference?

For the Council of the American Society of Mechanical Engineers.

ROBERT W. HUNT, *President.*
F. R. HUTTON, *Secretary.*

TO THE COUNCIL OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

Your Committee, appointed to consider the question of extending an invitation to foreign engineering societies to visit this country at the time of the Columbian Exposition, beg leave to report as follows:

I.

At the threshold of its inquiry, the Committee was confronted with the question as to what societies should be included in such invitation. Having in mind the hospitalities extended to the American engineers visiting Europe in 1889, and the fact that their hosts in England were the Institution of Civil Engineers, and in France the Société des Ingénieurs Civils, it is evident that the invitation should certainly include the membership of those two societies. The relations of the Institution of Civil Engineers to the other English engineering societies are such as to make it unnecessary, in the opinion of the Committee, to extend an invitation to any of the others, none of which took official part in the entertainment of the American engineers in 1889, and the more prominent members of all such societies being also members of the Institution of Civil Engineers. The British Iron and Steel Institute did not, as such, take part on that occasion, and has, moreover, been the recipient recently of hospitalities from the American engineering societies, under the leadership of the Institute of Mining Engineers. In France the Société des Ingénieurs Civils is the only national engineering organization, and is distinctly representative of the whole body of French engineers. A small party of the American engineers in Europe in 1889 visited Germany, and were the recipients there of hospitalities from several engineering societies, but chiefly from the Verein Deutscher Eisenhuttenleute. As the latter society, jointly with the British Iron and Steel Institute, participated in the recent visit to this country, and in the hospitalities extended at that time, the obligation of American engineers to that society was fully and handsomely discharged. In view of this, and for the further reason that an invitation to German engineers could hardly be limited to the membership of that society, it is not deemed expedient that the proposed invitations include any of the German societies. A further and final reason for this conclusion is the fact that, if such an invitation be given, it is difficult to see why similar invitations should not be extended to the engineering societies of Austria, Italy, Russia, and all other European countries. The Committee therefore recommends that the proposed invitation be limited to our hosts in England and France—the Institution of Civil Engineers and the Société des Ingénieurs Civils.

While it seems expedient that the special invitation be thus limited, the Committee suggests the propriety of extending a general welcome from American engineering societies to individual members of engineering societies, in all parts of the world, who may visit the United States during the exposition, including therewith an invitation to avail of the hospitalities of the houses of the respective societies in New York and of the engineering headquarters at the exposition. This more general invitation, if given, will have added value if conveyed by separate notification to each foreign society of which the name and address can be learned.

II.

The second question before the Committee relates to the manner in which the invitation shall originate. No society of engineers in the United States is representative of the whole profession. The invitation given by our French and English hosts in 1889 was extended equally and jointly to

The American Society of Civil Engineers ;
The American Society of Mechanical Engineers ;
The American Institute of Mining Engineers ;
The American Institute of Electrical Engineers.

It would seem, therefore, that all of these societies should unite in extending a return invitation to members of the English and French societies to visit the United States during the period of the Columbian Exposition. This implies primary action by the councils of the four societies in favor of the proposed invitation, and, preferably, the appointment of representatives on a joint committee of invitation. If this plan be adopted, it would probably follow that the joint committee thus appointed would be authorized to take charge of the whole matter and arrange all the details. An invitation emanating from such a source would doubtless recite at its opening the fact that all four of the American national Societies of Engineers had united for this purpose, and that the invitation was given by their united representatives in their joint behalf. The work of this committee would be such as to make it eminently fitting to have it composed of the President and one or more vice-presidents of each of the societies indicated, unless it were thought better that it be composed wholly or in part of past-presidents and vice-presidents.

III.

The third question before the Committee relates to the character of the invitation and the responsibility thereby assumed by those who give it. Referring to the precedents of the English excursion of 1889, it may be noted that the first communication on the subject was addressed by the President of the Institution of Mechanical Engineers to the President of the American Society of Mechanical Engineers, and extended an invitation to the Society "to hold a week's meeting in London," and stating that the visiting Society "would be warmly welcomed by the Institution of Civil Engineers, the Institution of Mechanical Engineers, and others." Subsequently, a letter was received from the Institution of Civil Engineers, asking whether many American engineers would visit Europe during the Paris Exposition, and intimating the desire of the Council "to consider the possibility of making such arrangements as may best tend to further the objects which the visitors have in view, and to render their visit generally as useful and agreeable as possible. The Council desires to embrace this opportunity of manifesting its friendly feeling to the utmost of its power. Of course, in any case, the facilities furnished by this Institution are always at the disposal of your members." A similar communication was duly received from the Society of Arts in London. The invitation thus given was subsequently accepted by each of the American societies, acting through their respective Presidents. Following this precedent, it would seem that the American invitation for 1893 should include a cordial offer of hospitality to English and French engineers; a suggestion that they could well follow our example by coming in a body, preferably by special steamer; that the houses of our several Societies in New York will be at their service; that, if agreeable to them, arrangements for a joint convention in Chicago would be made in advance; that committees will be appointed to formulate the lines and time-tables for excursions, and for otherwise facilitating the movements of the visitors while here, and especially for securing for them the privilege of visiting such American industrial works as they may wish to see. The Committee believes that it will not be expedient or possible to assume any obligation to entertain the visitors in the manner nor on the scale of hospitality extended to the British Iron and Steel Institute in 1890, and that therefore no reference should be made which would imply the intention of furnishing free transportation or of otherwise relieving the visitors from the usual expenses of travel. At the same time, the Committee recommends that the plan of arrangements should contemplate the entertainment of the visitors in New York or Chicago, or both, on at least one formal occasion, the expenses thereof being borne by contributions from the members of the American societies.

In conclusion, the Committee recommends that whatever action be proposed should be taken at an early date. The first suggestion from English engineers, relating to the visit of American engineers in 1889, was dated October 6, 1888, or only seven months before the sailing of the party from New York. The intervening time was occupied with constant and active correspondence, and was shorter than was convenient for the transaction of the business involved. It would be better to have such invitation issued ten or twelve months previous to the time of the proposed visit, which, in this case, involves its being sent in the summer of 1892. As much preliminary discussion will be involved in adjusting the relations of the four societies involved and in determining upon a plan of action, the time remaining would seem to be no longer than required.

All of which is respectfully submitted.

HENRY R. TOWNE,
ALFRED E. HUNT, } *Committee.*
F. R. HUTTON,

COPY OF LETTER FROM COUNCIL TO MR. C C. WORTHINGTON.

The Committee of the American Society of Mechanical Engineers hereby tenders, on behalf of the membership, the cordial thanks of the Society to C. C. Worthington for his generous gift of a portrait of his honored father, the late Henry Rossiter Worthington, a gift prized not only for its intrinsic value but also as commemorative of one of the founders of the Society, who, during his lifetime, was ever one of its warmest and wisest friends. Recognizing the great and lasting obligation which the Society owes to Mr. Worthington for his part in its organization, and recognizing also his high professional attainments and reputation, known and appreciated throughout the world, and rivalled only by his personal character and moral worth, the Council of the Society has given permanent place to the portrait on the walls of its Auditorium, where, in company with that of his friend and associate, Alexander L. Holley, it will lend dignity to the surroundings of those who in future may preside over the meetings of the Society, and will make familiar to the membership attending those meetings the features of one of its most illustrious members, and whose na· in evidence of that appreciation, has long been recorded upon our rolls under the title of

"HONORARY MEMBER IN PERPETUITY, DECEASED FOUNDER OF THE SOCIETY."

By Order of the Council,

F. R. HUTTON, *Secretary*. ROBERT W. HUNT, *President*.

At the close of the Report of the Council, the second order of business were the Reports of the Finance and House Committees of the Society, which were presented as follows, and were ordered on file:

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, 1890, to November, 1891:

ANNUAL REPORT.

RECEIPTS.

Initiation Fees	$3,180 00
Annual Dues, Current	11,738 00
" " Past	110 00
" " Advance	70 28
Paper Sales	1,624 91
Binding	775 94
Badges	936 50
Engraving	343 20
Rent of Hall and Parlor	425 00
Life Membership (One Bond, $100, Surrendered)	500 00
Furniture and Fixtures	28 30
Interest on Balance in Savings Bank	73 46
Balance, November 1, 1890	37 03
	$19,842 62

EXPENDITURES.

Printing and Stationery	$2,417 95
Publications	8,090 23
Postage	1,121 49
Library	63 72
Salaries	4,152 42
Expense of Office	629 86
Engraving	1,825 45
Contingencies	29 50
Binding	844 20
Meetings	885 30
Furniture and Fixtures	620 90
Badges	707 00
Travelling	121 33
Rent	2,199 96
Unveiling of Holley Portrait	212 14
Work of Committees	37 21
Sinking Fund	808 50
Interest on Balance in Savings Bank	73 46
Balance in Treasurer's Hands, November 1, 1891	2 00
	$19,842 62

There stands also to the credit of the Society, for its Library Fund, the sum of $2,272.67, deposited in savings banks and drawing interest.

There are, however, outstanding bills against the Society for work done in its interest amounting to $3,586.15, the items as follows:

J. J. Little & Co., printing Volume of Transactions	$2,730 10
Bailey, Banks & Biddle, Pin Badge Account	198 00
American Bank Note Co., Engraving Account	70 00
Sundries (small accounts)	145 00
F. R. Hutton, on account of advances	443 05
Total	$3,586 15

for the payment of which at this time there are no funds. There is one bond of the Mechanical Engineers' Library Association transferred to the Society in payment of a Life Membership which the Committee holds as cash invested, the par value of which is $100, bearing interest at five per cent.

There remains uncollected from the membership of over 1,300 only the sum of $129.90 from eleven of its members. The Committee has directed that members over eleven months in arrears be drawn upon for the amount of their indebtedness, and as a result of this practice the Committee attributes the unusually small number of delinquents in so large a society. It speaks well, also, for the healthy interest of all members in the work of the organization.

Respectfully submitted

By the Finance Committee.

November 12, 1891.

REPORT OF THE LIBRARY AND HOUSE COMMITTEES.

The formation of the Mechanical Engineers' Library Association rendered the continued existence of the Library Committee of the American Society of Mechanical Engineers a superfluity as a separate body, and it was therefore consolidated with the House Ccmmittee of the Society entrusted with all matters pertaining to the carrying on of the Society's work in its enlarged quarters.

Five bedrooms have been fitted up in the third and fourth stories of the house, which have been made use of by the members during the year, bringing in an income which has been credited to the Library Association amounting to nearly a thousand dollars. The movement of having these bedrooms at the service of the non-resident members has been very popular, and will be continued as a feature of the Society's life.

The Reunions which were held in the Society's house, while in no sense a Society affair, but carried on only by the resident members, who bore their entire expense, were under the control of the House Committee, who assumed the burden of seeing after their details. They were voted so popular a feature that they will be probably continued through the coming winter, as a means of cementing the professional bond between the members and their families who are able to be present. A large proportion of non-resident members was on hand at nearly every meeting.

The report of the finances of the Library Association is made to the entire Society of Mechanical Engineers, through the channel of the House and Library Committee, and is appended to this report.

LIBRARY ASSOCIATION.

The summary of the receipts and expenditures of the trustees from November, 1890, to November, 1891, is appended below:

RECEIPTS.

Fellowship Fund	$349 00
Sinking Fund	1,955 92
Library, Permanent	538 00
" Current	575 50
Rent Office	2,676 63
" Rooms	996 13
Equipment	25 00
Interest	9 58
Bonds	1,400 00
Balance on hand, November 1st, 1890	22 74
	$8,548 50

EXPENDITURES.

Interest on Mortgage	$1,485 00
" " Bonds	1,634 58
Salaries	490 00
Supplies	84 25
Fuel	165 95
Lighting	418 96
Equipment	685 91
Laundry	199 25
Insurance and Taxes	77 95
Binding	158 05
Repairs	147 06
Bonds	1,100 00
Bills payable	1,000 00
Balance on hand, November 1st, 1891	901 54
	$8,548 50

The Committee of the Society on Standards presented its report of progress, as follows:

REPORT OF COMMITTEE ON STANDARDS.

To the President, American Society of Mechanical Engineers, New York, N. Y.

Dear Sir:—Your Committee on Standards reports as follows:

A bill to provide for the registration of standards, in accordance with the resolution passed by the Society, was presented to the lower house of Congress early in the last session, and was referred to the Committee on Patents, Butterworth, Chairman. Your committee appeared before that committee, and argued in favor of the bill, and the general sentiment, as unofficially gathered, was favorable. Later correspondence with Mr. Butterworth failed to get any responses, and the introducer of the bill, Mr. Morey, was appealed to, and reported that he had consulted with Mr. Butterworth who stated that he would report in favor of the bill. But he did not report, so far as your committee has been able to ascertain, and it is believed the omission was due to outside duties assumed by Mr. Butterworth. The session expired and the bill expired with it.

Your committee therefore asks for further time in which to try it again at the coming session of Congress.

The Society has before had experience in matters of this kind, and with the same result, it being found extremely difficult to arouse in Congress an interest in a matter which has no politics in it.

Respectfully submitted,

J. W. SEE, *Chairman.*

HAMILTON, OHIO, *November 2d*, 1891.

The Committee on Standard Tests and Methods of Testing presented its Report, upon the recommendation contained in which there followed some little discussion. The Report was as follows:

NEW YORK, *November 17th*, 1891.

Mr. President and Gentlemen.—Your Committee on Standard Tests and Methods of Testing Materials, begs to present the following Report:

Whereas, The Preliminary Report of your above Committee has by many been considered to apply to Routine, or Shop Tests, though a note on p. 587, Vol. XI., Trans. A. S. M. E., distinctly explains that the Report given as paper No. CCCLXXX, p. 604, et seq., applies to Scientific or Standard Tests alone, and that a Report on Routine or Shop Tests would follow as soon as the Committee shall be able to draw it up. The Committee requests that a slip be sent to the Society Members explaining this, which is to be inserted in Vol. XI., p. 604.

Another slip should correct an error on p. 628, changing 32-40° F. to 70-75° F.

The Committee desires to call attention to the desirability of being represented at the next Conference of German, Swiss, Russian, Austrian, and French Engineers, to be held at Vienna, in Sept., 1892. and requests that the Reporter to the Committee, Mr. Gus. C. Henning, be requested to attend that Convention as the representative of this Society. The Committee further advises that the Society communicate with the Secretaries of War and of the Navy, and explain the desirability that the military and naval attachés at the Austrian and German, or other Embassies, be ordered to attend such Convention.

The Committee, at the request of a number of the most prominent engineers and Professors abroad, communicated with the Head of the Dept. of Engineering Congresses of the World's Fair at Chicago, for the purpose of setting a time of holding an International Conference of Engineers, interested in the subjects under discussion by this Committee.

Very respectfully,

HENRY R. TOWNE,
R. H. THURSTON,
T. EGLESTON,
CHAS. H. MORGAN,
GUS. C. HENNING.

Mr. Henry R. Towne.—The members will remember that the Committee has already submitted a quite extensive report which is printed in the past transactions, and has also presented a translation of the minutes of the German Conference on the subject of tests, both of which, it is believed, possess a great deal of value, especially the translation referred to. It is, I believe, the only translation in the English language of the most important conference that has yet been held on this subject. There is a continuing International Committee charged with the consideration of this question, having representatives from all the European countries, and meeting at various points on the continent. The next meeting is to take place in 1892, and Mr. Henning, who has been, I wish to state emphatically, the working member of this Committee, and to whom is due substantially all the credit for what has been accomplished, has expressed a readiness to attend that convention, if it is the desire of the Society that he should do so, without charge for his time and services, but conditioned upon the payment by the Society of his travelling expenses. This is a question which should be considered by this meeting, or else referred to the Council for its consideration and action. The work of the Committee is not concluded, and Mr. Henning is willing to continue to give further time to it. The outcome of his work, with such assistance as the other members of the Committee are able to give him, will, I believe, fully justify the continuance of the Committee.

Prof. Gaetano Lanza.—I would like to ask in what form this is to come up again. Will there be another report at some subsequent meeting?

Mr. Towne.—The hope and expectation is, that the Committee will conclude its labors by drawing up a report embodying suggestions for a standard method of conducting ordinary tests of materials—those which are made in the shop or mill for commercial purposes. The recommendations contained in its report thus far presented relate rather to scientific work and finer investigations than are needed for commercial purposes.

Mr. J. F. Holloway.—In order to bring the matter properly before the Society I would move that the report of the Committee be accepted and their labor continued.

The motion was seconded.

Mr. Towne.—Before the question is put, Mr. President, I think it would be expedient to have some indication of the sense of the meeting in regard to the desirability of our being represented at this International Congress. It is hoped by some of our members, and I think especially by those who have been on the Council in recent years, that the work of the Society is going to broaden in the future,—that means will be provided to extend it, and that from time to time it may be found possible to apply a moderate portion of the Society's funds, and to secure the benefit of the services of some of its members in promoting scientific investigations of subjects which are germane to the work of this Society. The present opportunity is certainly one of this kind, and one of which, in my judgment, it would be desirable to avail if it is deemed by the Council to be within our present means.

The International Conference of Engineers on the subject of uniform methods of test and uniform standards of test pieces, has been doing a very notable and useful work. Its labors have continued now, I think, for eight years. Its membership in other countries includes some of the most brilliant members of the profession and many names of international standing, and it is evident that the work which this international committee is doing will have very great value to the world. At the present time there is being made in every important producing establishment in the iron and steel industry all over the world, and in many other establishments, an infinite number of tests of materials, which, however, are being so conducted as not to make the work comparable in a great majority of cases. If this work can be so standardized as to be comparable in all or a majority of cases, it will contribute immensely to our knowledge of the materials entering into the work of engineering. That is the objective point of the work which is being carried on in Europe, and it is certainly desirable that this country should be represented in it.

No other body of engineers in this country is more fit than this Society to represent American interests there, and the Committee hopes earnestly that the Society may see fit to sanction action which will lead to such representation.

Mr. Gus. C. Henning.—In regard to the results of the work of the Committee, I should like to point out that the Society is now in possession of all of the work which is reported from the Munich and Berlin Laboratories, which are probably the most prominent in the world, in conjunction with the private laboratory of the Paris and Mediterranean Railway. The reports of the latter are not published as a rule, except occasional papers; but we have succeeded in obtaining an interchange of transactions from those two former, and many of the prominent engineers and authorities in Europe have requested that the Society be represented at this Congress or Conference at Vienna in September, 1892, for the distinct purpose of making our deliberations international. It seems that the best investigators in this field carry out their work in a manner so nearly similar to that which obtains here, that with very slight modification we could obtain precisely similar standards, and then all our work would be comparable. But probably an exchange of opinions at this next conference would do more than anything else to establish such similarity of methods and standards, and the next conference will finally settle that matter. It would be the first successful effort to establish international standards of equal value everywhere, so that if anything is published in any other country we can read it with complete intelligence. As it is now, we rarely ever know what the result of a test means, because, if we do not know on what basis the test has been made, we can form no good opinion of what those results mean; for if different standard methods are used the results will differ, and the materials will apparently have different qualities; while if we knew the value of the differences of methods, we could say whether the materials were alike or different. If then we approach the European Engineers and explain our position in the matter, I think there will be very little difficulty in establishing international standards for all countries. Most engineers in France now make their tests almost the same as in Germany. But in England there seems to be a difference of opinion, and many of the engineers prefer to make their tests after their own fashion.

Mr. J. F. Holloway.—Before a vote is taken, I would like to say that I made the motion which I did, with the view and belief that the work which the Committee has in hand is one which commends itself to all who are here. As to the question of sending a representative abroad or spending any further money in regard to it, that is a more serious question. In view of the state of the finances of this Society as exhibited here today, I should think that we are hardly in a condition to warrant any increased expenditures at present. What may be the result of the meeting in regard to trying to increase the resources of the Society, of course is as yet unknown. But I am sure that every one who has listened to what has been said in regard to these tests, realizes the importance of them, not only to this Society, but to engineers generally, and I would be very glad indeed if means were provided by which they could be further extended. So, while it is not in my motion, I think that this question of expenditures is one which will have to be left dependent on the future state of the finances of the Society.

Mr. Towne.—The amount of money involved is not so large as may seem at first glance. Mr. Henning tells me that he estimates it will not exceed $250. This is a sum which should not frighten the Society, under the conditions which, we hope, are hereafter going to obtain in our finances. But if it should be the case that the Council found it inexpedient or impossible to provide even that modest sum, I think it entirely probable —indeed I believe it certain—that the Council would easily secure it from voluntary sources; and I ask therefore if Mr. Holloway will not add to his motion something to this effect: That the matter be referred to the Council with power to take such action as it thinks proper and within the ability of the Society.

Mr. Holloway.—With respect to the ability of the Society, if there is anything more favorable than what has been heretofore stated to-day, I should be glad to know it. However it is a matter that should be brought before the Council, and I would be very glad to accept the amendment.

The Secretary.—It may not be known to the members of the Society—and I am quite sure, from the remarks made by the Chairman of the Committee, that it is not known to him—that after this Committee was originally established, it was found, for reasons entirely satisfactory to itself, that a reorganization was necessary; at that time Mr. Towne was selected as Chairman of the Committee. The principal function of the Committee previous to that time had been the collection of a fund of $155 for the purposes of the work of this Committee. That fund was turned into my hands when the Committee was reorganized and it is still there, less the drafts upon it for the work of the Committee in having tests made in series, two years or more ago. There is nothing whatever, therefore, to prevent this Committee, without going to the Society at all, from spending the sum of $112; and I think, therefore, it would be quite safe for the resolution, as proposed by the Chairman of the Committee, to be passed.

Mr. Towne.—One of the best assets of the Society is the good memory of the Secretary. I am happy to see that that asset is not diminished.

Mr. Hunt.—It is a question whether it is not better for the general policy of the Society to have these funds raised by private contribution instead of being taken out of the Treasury of the Society. I believe that for all ordinary investigations of this kind, a little effort would soon bring the money from private contribution.

Prof. Gaetano Lanza.—I would like to ask whether in the international congress, as far as Europeans are concerned, there are any representatives of societies, or whether it is not rather a congress of individual engineers. I think we ought to be pretty careful about the way in which we adopt definite standards recommended by committees. I know from my own experience that there are a great many reports of committees where things have been proposed on which new light has afterwards been thrown showing that the things proposed by the committee were not good. Now, if the European custom is not to have representatives there of societies, but simply representatives as individuals, it is a question whether it would not be better to have Mr. Henning there representing himself only, giving on his own motion whatever he has to say, while the fact that he has been the Secretary of our Committee, renders it very desirable that he should be there. I have in mind many reports of committees which I have no doubt many of the members of the Society would be ready to criticise adversely. It is a question whether any report is not likely to be open to such an objection.

Mr. Gus. C. Henning.—The conferences are composed as follows: Of engineers and manufacturers, Government employees on railroads, and of what are called "baumeister," that is engineers in charge of construction in different departments, attending as individuals. But these conferences have been of such importance that several Governments order their official representatives to attend. But official representatives of Governments in Europe have no right to express opinions or discuss matters. The important societies all have very many members present at these conferences. Their secretaries invariably attend. But the societies there do not any more than does our society here, ever bind themselves to stand by the report of the committee. Our Society does not adopt the report of a committee, but it is merely published in the Transactions. The Society is not bound to follow any of the resolutions or recommendations of the committee. The committee does nothing but recommend certain standards as the best. But the Society in no way binds itself. The first resolution of these European conferences is, that none of the resolutions of such conference shall be binding, because they may require modification, from time to time, due to more recent or more exact investigation. Hence they can only be of a temporary character. But the way better results can be obtained is by doing the work in the same manner everywhere. One principal reason why this Society should be represented there is, that the Committee has been so requested by leading European engineers. The second reason is, that we shall have to wait from six to nine months, or a year, before we can obtain the reports of the deliberations of these conferences, if we are not represented by a delegate. Thus we have always been a year behind the actual state of affairs in the latest developments. If a representative were present, his report could be issued at the next annual convention in November, and would be fully up to date. The presence of a member of this Committee there would of course in no way bind the Society, but the expense of going there would be considerable, and in addition there is the time that would have to be devoted to it, being about one month—certainly not less than twenty-eight days; so that the Committee could hardly have a representative there, unless he were sent by the Society, and if he would go accredited as a delegate of this Society he would be received with much greater consideration, as you may understand, than if he went as an individual. So far, invitations have not been sent to this country, because Europeans were not aware of what was being done here; but now, due to the correspondence of the Committee, they have been made aware of what we are doing and our presence is desired at these conferences. American engineers going to Europe for other reasons, if they happened to be in Vienna at that time, would perhaps attend; but not otherwise, the distance and expense being so great.

The President.—What was your motion, Mr. Holloway?

Mr. Holloway.—My motion was, that the report of the Committee be accepted, and that they be granted further time for investigation, and that if, in the judgment of the Council, the means at their command will permit, a member be sent to Europe to represent the Society.

Prof. Lanza.—While I imagine that I am in a small minority, I would say that I should prefer that instead of saying that a member be sent to Europe by the Society, that a vote be made that the travelling expenses of the member attending that conference be paid. It is merely a matter of general principle, and I will not offer an amendment unless Mr. Holloway thinks well enough of it to adopt it.

Mr. Towne.—I think Prof. Lanza, does not, perhaps, understand that the Society has placed itself on record that it will not endorse the opinions of any committee. The subject has come up at different times, the sentiment of the Society in the matter is fixed beyond question, and all of us understand that the reports of committees do not bind the Society in any sense. They simply represent information which is gathered by a committee of our members for the use of all the members and of the engineering public. I can see no possible objection to this Society sending an authorized delegate to the Congress at Vienna, for the purpose of obtaining information concerning what takes place there and reporting back to us. He goes there as our Commissioner, to obtain and bring back to us a knowledge of the transactions and discussions which take place there. Surely there can be no objection to that. It commits the Society to nothing, nor will the report of the Committee, when filed, commit the Society to anything, any more than any paper printed in our Transactions commits the Society.

The Secretary read the resolution proposed by Mr. Holloway, and it was adopted.

The Committee on Standard Flanges reported, through its Chairman, Carleton W. Nason, the inauguration of their work, and their hope that something practical might be evolved from the steps which had been taken.

The report was as follows:

Mr. Carleton W. Nason (Chairman).—The Committee at this time report progress only. While a Committee has been listed under this title for some time, it is practically a new one, and has been at work only since the Providence Meeting, last June. The scheme of the Committee for circulating inquiries among manufacturers and others who may furnish the Committee with the information which they seek is one which takes time; but no doubt at the next meeting we shall be able to report not only as to the results of inquiries, but also with recommendations as to standard proportions.

In support of the report the following remarks were made:

Mr. Arthur C. Walworth.—I would like to state for the benefit of those who are here, that at the Spring Meeting of the American Association of Master Steam Fitters, a Committee was appointed on this same subject of standardizing the pipe flanges throughout the United States. Mr. E. P. Bates, of Syracuse, is one member of that Committee and I happen to be another, and Mr. Bates and myself are waiting to see what result will be produced by your Committee in order that we may compare notes and work together. It is evident that if the two Committees can unite on one report, that will carry with it the accelerated force of two associations and have a much better chance of being adopted in the United States. So that I wish to assure members of this Association that the Steam Fitters see the importance of this subject just as they do, that we are working for the same end, and that we propose to work together, and not present two reports which will be at variance.

The Report of the Tellers appointed to count the ballots for Officers at the preceding meeting was then presented as follows:

Total number of votes cast	559
Informal votes	21
Regular	538

Of these regular ballots:

For President	Mr. Charles H. Loring	received	534	—scattering 2
" Vice-Presidents,	" Geo. T. Alden	"	530	
" " "	" E. F. C. Davis	"	528	
" " "	" Irving M. Scott	"	534	
" Managers	" Jas. M. Dodge	"	535	
" "	" Robert Forsyth	"	531	
" "	" Jesse M. Smith	"	532	
" Treasurer,	" William H. Wiley	"	538	

Respectfully submitted,

Henry A. Bang, } *Tellers.*
Geo. Dinkel, Jr., }

Under the head of Special Orders, the proposed amendments to the Rules offered, under the provision in Article 45, at the previous meeting of the Society, were then taken up. The Council, in considering the subjects presented under these proposed amendments, had directed that a circular of explanation, giving the arguments in favor of the judgment of the Council upon them should be sent to all the membership, so that the questions involved might be formally and adequately considered before the members came to the meeting at which the decision upon them was to be had under Article 45 of the Rules. The text of the circular which had been issued was as follows :

CIRCULAR CONCERNING PROPOSED AMENDMENTS TO THE RULES.

There are certain amendments to the Rules on which action is to be taken by the Society at its Annual Meeting in November next. The first three of the proposed Amendments relate to an increase in dues, and notice of them under the Rules was given by a member of the Finance Committee of the Society. The fourth Amendment relates to a suggested change in the method of nominating officers and is proposed by a member not holding office. Inasmuch as the Council may be supposed, from its position as managing the business of the Society, to be better informed as to present wants and future possibilities than the membership at large, it has thought that it would be advisable to give full information as to the Amendments and their bearing upon the Society's work in advance of the meeting, so that those less familiar with the details of the matter can be posted more fully than can often be possible in a crowded and busy session.

The several financial amendments offered for consideration are as follows :

I. To raise the initiation for new Members and Associates elected hereafter from $15 to $25, and for new Juniors from $10 to $15.

II. To raise the dues of Members and Associates from $10 to $15, and of Juniors from $5 to $10.

III. To increase the Life-membership fee (which corresponds to about 15 years' dues) from $150 to $200.

While these Amendments must, under the Rules, be discussed and finally acted on by those actually in attendance at the Annual Meeting, yet it will be useful to have them considered in advance by the entire membership, together with the reasons which have induced the Council to think favorably of them. If unable to be present at the meeting and to take part in discussion, you are therefore urged to forward an expression of your views by enclosed card, so that those who attend the meeting may be informed as fully as possible, concerning the general sentiment of the membership, and may be thus aided in reaching a decision which will commend itself to the judgment of the majority.

Among the reasons which have induced the Council, after careful examination, to give their approval to the proposed financial Amendments are the following :

I. As to the initiation fee : The greater the prestige of the Society, and the more numerous the benefits of membership, the more can be properly asked of those who seek to gain these benefits. The proposed increase affects none now in the Society, but only those who may join hereafter. The value and prestige of membership have grown with the developments made during the time that the present members have been paying dues, and are greater and more valuable to-day than ever before. It is proper that this fact should be recognized.

II. As to the increase in dues: The reasons for favoring this step are many.

1st: The return to members for their dues may be divided into five principal items. 1st—The value of elected membership in a professional society of such standing in the community. 2d—The privileges and opportunities of the Conventions in the matters of intercourse, excursions and visits to works. 3d—The annual volume of Transactions viewed as technical and professional literature. 4th—The use of the House of the Society, with its rooms and office privileges for non-resident members, and its Library for all who can avail themselves of it. 5th—The service of the officers of the Society and its staff of employees whereby its existence is maintained, and its value and usefulness to members enhanced in many ways.

When the present low dues were fixed by the founders of the Society more than ten years ago, they were fairly adjusted to the proposed return. As is well-known to most of the older members, who can compare the earlier standards with those of to-day, this is no longer the case in any department, and especially as regards the annual volume of Transactions. This volume now contains from 1,000 to 2,000 pages of matter which, by common consent, is regarded as most valuable professional records. A book of this size cannot be issued, electrotyped, bound and distributed, with illustrations and tabular matter of the grade consistent with our present standing, for a cost less than $7.50 per volume. The earlier volumes, of half the present size, cost about two-thirds as much, and on this ground, therefore, it would appear that the original dues should be increased fifty per cent. and the junior dues raised to a figure covering the actual cost of the annual volume, so that juniors should be no longer the beneficiaries of the rest of the Society.

2d: If the volumes cost $7.50 each to produce and distribute, the dues of $10 leave only $2.50 from each member to meet all other outgo not chargeable to the volumes, such as general expenses, salaries, postage, rent, etc. This fact has made it necessary, heretofore, to use not only the initiation fees as current income, instead of funding them as capital, but also to expend the fees received for Life-membership, which are supposed to be invested and return interest to the Society. This plan is conceded by all to be wrong in principle and is certainly one which should not be perpetuated. It is the considerable sum received from initiation fees each year, due to what may be considered a phenomenal growth of the Society during the last five or six years, which has kept its head above water; and in the nature of the case, as the Society grows in size the number belonging to the grade eligible and desirable must be expected to diminish.

3d: The present dues amount to less than 3 cents a day, and the proposed increase will add less than 1¼ cents more. Except some European societies, having a different social membership from our own, no engineering society has lower dues. The English professional societies have dues of $15 and $25, and give a return no greater, or in some cases less. It is not conceivable that professional men of the grade of outward success, pre-supposed by our conditions of membership which, for full members, imply experience in the sole and responsible charge of engineering work, would find the amount of the new dues prohibitory or even onerous. If any one should find them so, a procedure may be established by which he can retain membership, but be listed in the junior grade, and thus receiving the volume and enjoying all the privileges of that grade, yet pay no more than he does now.

4th : Objections have been made to the present scheme of a multiplicity of accounts, collecting from a willing and interested few a confusing series of voluntary subscriptions to different funds instead of frankly raising the dues for all. Conceding this objection to be sound, it is proposed if the proposed motion is carried to discontinue all petty charges, and to cancel all subscriptions except such as attach to the Library and House funds. The volume of Transactions will then be sent to all *in library binding,* without extra charge, and pamphlet copies of papers, in small lots, will be distributed free. To do this is again to increase further the return for dues paid. For many members, therefore, the new dues will be no greater than the amount now paid by them voluntarily.

5th: Any surplus of income after obligations are met will be devoted to increasing the return to members, either by conducting and publishing professional investigations by Committees, by the compilation of indexes of professional literature for the members, or by such other methods as may be found most useful. The increased dues will thus be followed by a much more largely increased return, which in turn enhances the value of membership and the prestige it confers.

6th: The increasing number in attendance at the Society's Conventions will very soon bring up the question of an obligation upon the Society to bear a part of the expenses entailed by a Convention upon members resident in the cities where such Conventions are held. The present income does not permit of assuming this outlay in addition to the other demands upon the Treasury. We are now shut out from some cities of interest by reason of the small number resident at such centre, on whom the burden of a meeting could not with any fairness be held.

III. The Amendment relating to the Life-membership fee affects a relatively small number and should be passed if the foregoing is approved, to maintain the usual interest standard on the investment of such fees.

IV. The last Amendment proposes to direct and compel the Committee who present nominations for the offices falling vacant each year, to submit *two* names for each office, instead of one as now. If there is a wide-spread discontent with the present plan (which, however, has seemed to work reasonably well since its adoption, seven years ago), it would seem wise to direct the Council to appoint a Committee to study the practice of other societies in this matter and report. The Council foresees certain objections to the plan proposed in the Amendment, and suggests the expediency of taking further time and seeking fuller information before taking definite action.

The Council having given most careful consideration to these matters, and believing the course herein suggested to be the best for the interests of the Society (in which view the ex-Presidents have concurred), desires, in the interests of unanimity and to save time in discussion, to secure the benefit of your judgment so that it may have due weight at the meeting. To this end it is requested that you will kindly take the trouble to reply by enclosed card at your early convenience, certainly before November 16th.

For the Council,

F. R. HUTTON, *Secretary.*

FORM OF POSTAL CARD REPLY

THIS IS AN EXPRESSION OF OPINION, AND NOT A VOTE.

[Erase the opinions which you do NOT hold; SIGN the card and mail.]

I. I would advise the adoption of the proposed financial amendments and I am heartily in accord with the movement to enhance the value of my membership.

OR,

II. I would advise the raising of the initiation fees; and as to the dues, I will support the action of the voting members at the meeting.

OR,

III. I would advise the raising of initiation fees, but prefer to retain the present rate of dues, and the curtailing of the outlay to be met from dues and a return to the earlier and lower standards of publication and management.

Name, ...

Address, ...

The replies expressive of the opinions of the writers which had been received up to the date of the meeting amounted to 708, of which 651, or a majority of over ten to one, had expressed themselves in favor of the Amendment. In bringing this subject before the Society, the President spoke as follows:

The President.—We now come to the proposed amendments to the rules. The first is the motion made by Mr. Wheeler at the Providence meeting, giving notice that at this meeting he would propose certain amendments to Article 18 with which you are all familiar. It is right and proper that I should announce, as representing the Council, that this matter has received most serious consideration, particularly in view of our knowledge of the Society's financial condition. It was also thought wise to ascertain as nearly as possible the sentiment of the membership in regard to these changes, and not leave them to be alone discussed at this meeting, at which, of course, there is comparatively a small representation of the total membership, and in that view you received the circulars and postal cards with which you are all familiar, embodying the request that you should return to the Council your ideas on this subject. I will not give you the exact figures, and perhaps it would not be fair to do so; but you can draw your own inference when I tell you that this large pile represents the affirmative opinion as to the advisability of these amendments, and this small pile represents the negative.* I am also authorized to state to you from the Council

* The pile of negative replies, about one-half an inch high, stood on the President's table in view of the meeting, at the side of the affirmative replies, making a pile over eight inches high.

that at our last meeting which was held yesterday, and as representing our final consideration of this very important question, the following resolution was passed, and I, as your president, was authorized to lay it before you :

Resolved, That the movement to raise the dues and initiation fee as set forth in the circular sent out to the membership has the unanimous support and recommendation of the Council.

With that statement the matter is before you. But before debate is opened upon this subject allow me to call your attention to one fact. The constitution—the provision of which has been complied with by the proposer of these amendments to it, in giving his notice at our last meeting—does not permit us through it to throw open the door for a general amendment to the constitution. In other words, the vote must be yes or no on these particular propositions. We must either vote to increase the dues to the amount proposed or vote not to do so. We cannot consider amendments to any of the propositions. The constitution provides that a notice must be given of a proposed amendment to the constitution at the meeting preceding the one when action upon it shall be taken, so that you can easily appreciate that it must be a specific amendment. Were it otherwise, a member might say, "I propose at the next meeting to amend the constitution," and when the next meeting came he might move to change the name of the Society to Electrical Engineers, or some other unexpected change.

The matter is now before you for discussion, and I think it is fair that we should hear from some of the members who are not present, particularly those who have taken the pains to write letters with the request that they be read. There are letters on both sides of the question.

Mr. Nason.—If you will read the motion I will be glad to second it.

The President.—It is divided into three sections. I will read all three. But we must discuss them in detail, and the action will be taken on No. 1 first. The motion was to raise the initiation fee for new members and associates elected hereafter from \$15 to \$25, and for new juniors from \$10 to \$15. No. 2 was to raise the dues of members and associates from \$10 to \$15, and of juniors from \$5 to \$10. No. 3 was to increase the life-membership fee, which corresponds to about fifteen years' dues, from $150 to $200. All three are before you, but action must be taken in detail.

Mr. Nason.—I second Mr. Wheeler's motion, Mr. President.

The President.—The motion of Mr. Wheeler is now before you. The Secretary will now please read the letters.

The Secretary read letters from C. Seymour Dutton, Col. J. A. Price, Francis Stryker, Charles M. Morse, Mr. Ryder, and George E. Dixon, warmly favoring the proposed increase in dues and initiation fee, and one in opposition from Gram. Curtis. (The Secretary before reading Mr. Curtis's letter stated that Mr. Curtis evidently misunderstood the procedure by which changes in the rules could be made.)

The President.—As the Secretary states Mr. Curtis certainly misapprehends the condition of things, as our constitution does not provide for a letter ballot, and we are forced to live under the laws which we have adopted. The Council thought it carried out Mr. Curtis's idea in an informal manner by sending to the total membership the request for their opinion, and perhaps if Mr. Curtis saw these piles of postal cards he would think the membership had voted.

Before putting the question I wish to refer to the constitution so that we may be certain that we are absolutely right. Article 45 says, "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."

I will put the question in the first place, and if the vote is in doubt we will have to take a poll.

The question was put, and the motion was unanimously carried.

The President.—I take the risk of declaring that it has been properly carried.

The next question, gentlemen, is on the second clause of the same amendment, which is to raise the dues of members and associates from $10 to $15, and of juniors from $5 to $10. That is covered by the same resolution and seconding.

The motion was unanimously carried.

The President.—No. 3 is to raise the life-membership fee which corresponds to about fifteen years' dues from $150 to $200.

Mr. Gus. C. Henning.—Some remarks have been made adverse to this proposition on the ground that we have life members now who have paid only $150, while the new ones who become such hereafter will have to pay $200. The question was, "whether it was proper to leave the old members on the same basis as the new members, or whether that extra $50 should be paid by the old members as well." Inasmuch as I have been recently spoken to by several out-of-town members, I thought I would mention the fact now.

Mr. W. S. Rogers.—I have had some experience in other societies in this matter of raising the life-membership fee, and it induces me to think that as quick as we put up the amount of the fee we shall cut down revenue from that source. In some cases the rise had to be revoked for this reason, and while I will be glad to have the increase tried, I think we may want hereafter to go back to the lower figure.

Mr. John Humphrey.—The objection may apply as well to all membership fees. I think those who came in first justly entitled to the place at the original rates, and $200 is certainly no more than proportionate to the advance in other fees. In a pecuniary sense, life-membership becomes simply a question of interest upon a single advance payment, and the interest on $200 is less than the $15 annual dues. In fact, proportionately less than at former rates. Therefore, members will be quite as likely to pay $200 in the future as the $150 before the dues were increased.

Mr. Hawkins.—It seems to me that all the objection so far raised to increasing life-membership fees applies to what we have already done in the case of members, etc. If there is any injustice in it there is as much injustice to the ordinary members, juniors and associates, and that we would have just as good right to call on the old members, juniors and associates for the additional fee that we are now going to raise it to, as to call upon life members for it.

Mr. Woodbury.—It appears to me that this motion to increase the membership fees for life members is entirely consistent with the action to increase the ordinary membership fees. I do not perceive how there is any inconsistency in two sets of life members who have paid different fees any more than would apply to the whole membership, and no one would ever propose the idea of charging additional on the back dues of all the other members, going back to the first organization of the Society. The whole spirit of business policy, whether applied to commercial practices or to public affairs, is entirely opposed to retroactive payments, and I do not believe that anything of the kind will be contemplated in connection with the proposed action before the Society.

The motion to raise the life-membership fee being formally put by the President was then carried unanimously.

The fourth amendment was presented by D. K. Nicholson at the last meeting. It proposed to amend the Article No. 31 so as to have it read as follows:

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 at least two nominees for each and every office 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.

In presenting this amendment the following discussion was held:

The President.—As yet we are without a general expression from the membership on this. But I am authorized by the Council to present to you their views as embraced in this resolution, which was adopted yesterday:

Resolved, That the proposition to have two candidates named to each vacancy in the Society's roll of officers each year, does not commend itself to the judgment of the Council, and they think if adopted that it will not conduce to the harmony and success of the Society.

You will appreciate, gentlemen, that the Council, in submitting this and the preceding resolution, do it simply because they feel it to be their duty, having been elected by you to conduct the affairs of the Society, and we think we do not presume too much in believing that we still have your confidence—that nothing has happened to destroy that. This matter has been carefully discussed by us in all its lights and bearings, and the resolutions reflect our judgment; but of course the decision rests with you. Does anybody second Mr. Nicholson's motion?

The Secretary.—I have, Mr. President, two or three letters.

The President.—The motion is not seconded yet.

Mr. Spangler.—I second it, sir.

The President.—Mr. Spangler seconds it. It is now before you, and the Secretary has two or three letters which he will read.

The Secretary read letters from Mr. Randolph, Mr. Charles A. Moore, Mr. J. Leon Gobeille, and Mr. Jesse M. Smith, the two latter emphatically urging valid reasons against the proposed plan.

Mr. Arthur C. Walworth.—I do not see the slightest necessity for this amendment. It seems to me that it would soon defeat itself. I do not see what we should gain by it. If you look at this amendment carefully you can see what could be done by the Nominating Committee. That Nominating Committee is an odd number, and three of that Committee carry the report of the Committee with it. Suppose that three of that Committee had a man that they wished to nominate. They can report that name; then they can report the name of a man who is unpopular, unknown—perhaps the last man elected in the Society. They could practically make one nomination if they wished. I do not say they would do it; but we should look at all contingencies. It seems that this Article 31 is very wisely drafted. If there are five discontented men in the fourteen or fifteen hundred that constitute our membership, they can send in a nomination, and that nomination will appear on the ticket without any distinguishing mark. It is the Australian system brought to perfection; and I believe in leaving well enough alone, and not trying to improve a system which seems to me to be perfect already.

Mr. C. J. H. Woodbury.—The growth of this Society has been wonderful, its transactions valuable, its meetings interesting conventions for the interchange of opinion and experience in their application to the latest methods of engineering practice ; and, in short, it has always performed in the highest degree all of the functions to be expected from a Society of American Engineers. And these results have been obtained as a logical result of the universal harmony which has always prevailed in the Society. The officers have always been closely in touch with the membership as a whole, and I want to ask you, in the name of common sense, considering all the results which have been achieved, is any one going to be wise enough to better that matter? (Applause.) As the President has refreshed our memories with the technical matter of our rules, it would seem that each and every contingency has been fully provided for. The Nominating Committee is freed from local bias by its selection from various portions of the country, and its duties are limited to nomination of new men, thus causing the most rapidly possible rotation in office. How are we going to improve the matter by a double nomination? If, however, there should be any person nominated, under any possible contingency, who should not be acceptable to the membership, then a very small proportion of the membership has the full right and privilege under our present constitution of making a double nomination; and I believe, as one of the members who does not have an opportunity to attend the frequent meetings of this Society, that we are entirely satisfied with the result of the administration of the past officers of this Society. I do not see why we should not have the same satisfaction with respect to any future administration under the same methods that we have so successfully followed in the past. Suppose that there should be a double nomination. I do not care what may be the individual feeling of the nominee: no person could restrain his friends from running to extreme measures, and many of them might be injudicious, and, in fact, unwarrantable. I am a member of two or three societies which have had double nominations, or bolting nominations, and in every case the results have been deplorable, and I think that the American Society of Mechanical Engineers is wise enough to profit by their example. (Applause.)

The President.—By an affirmative vote, you understand that you amend the constitution. A negative vote leaves the constitution as it is.

A vote was taken, and the amendment was declared defeated.

The President.—This is the time for any new business. If any member has any new business to offer, we should be pleased to receive it.

I am authorized by the Council to read this resolution:

Resolved, That the invitations from the Pacific Slope, to hold the Spring Meeting in 1892 in the city of San Francisco, be accepted.

This was the official action taken yesterday. (Applause.)

After some announcements by the Secretary, the morning having been exhausted, the session adjourned, to meet at half-past eight in the evening.

No definite assignment was made for Tuesday afternoon, but the half-day was put at the service of the members in attendance at the meeting either to utilize it in social reunions at the rooms, or for the purpose of visits to points of professional interest in and around New York. Arrangements were made by the Local Committee to give opportunity to visit the following places:

I. Morgan and Quintard Iron Works.
II. North River Iron Works.
III. Pond Machine Tool Works.
IV. Stevens Institute of Technology.
V. Columbia College, Schools of Arts, Law, Mines.
VI. " " School of Medicine (College of Phys. and Surg.).
VII. Pratt Institute.
VIII. Washington Bridge.
IX. Ship-yards and Works Samuel L. Moore & Sons' Co.
X. World Building—Equitable Building.
XI. Metropolitan Museum of Art.
XII. Museum of Natural History.
XIII. New York Oxygen Co.
XIV. Ball & Wood Engine Co.
XV. Harper Bros.
XVI. Power House, 125th St. Cable Road.

The number appended to each place corresponded to a number upon a card which gave directions as to the routes to be followed to reach the object desired and the persons for whom to inquire upon arrival. This plan appeared to work very well and was generally approved.

THIRD SESSION. TUESDAY EVENING, NOVEMBER 17TH.

The professional papers of the session were taken up as follows:

Three papers by A. F. Nagle, of Chicago, entitled "Experiments to Determine the Rate of Rise or Fall of a Mercurial Thermometer under Different Conditions," "Test of Indiana Block Coal at the Chicago West Side Pumping Station," and, "Limitations of Steam-engine Economy," were discussed, respectively, by A. C. Walworth, William Kent, C. E. Emery, Professor Lanza, and O. C. Woolson.

Charles T. Main read a paper on "The Value of a Water Power," which was discussed by Messrs. Emery and Cartwright;

and a paper by Samuel McElroy, on "Brooklyn Pumping Engines of 1860," was discussed by Messrs. Emery and Kent.

Interspersed among the papers, to give life and variety to the sessions, were certain Topical Queries on "Method of Standardizing Sizes for Keys," on "Portable Apparatus for Light for Out-door Work," on "The Best Design for Stopping and Starting Shafting on Any Floor in a Mill."

Wednesday was devoted to excursions and festivities. The steamer *Laura M. Starin* left the foot of West Twenty-third Street at 10 A.M., and carried a party of 300 ladies and gentlemen to the Navy Yard, where the cruisers and battle-ship were inspected by the party, with a few of the shops. Returning to the boat, luncheon was served in transit to the works of the De La Vergne Refrigerating Company. at the foot of East 138th Street. After inspection at this point, the boat brought the returning party to the foot of East Thirty-first Street, where it was dispersed.

In the evening, a theatre party was given by the Local Committee to their visiting friends and their ladies, at Palmer's Theatre, Broadway and Thirtieth Street. The party numbered 354, and occupied nearly the entire ground floor of the house. A reproduction of the Society's pin badge in gas-jets decorated the exterior of the building, and souvenir programmes were distributed to each one as mementoes of the evening.

FOURTH SESSION. THURSDAY, NOVEMBER 19TH.

Professional session for presentation and discussion of papers. A paper by W. E. Crane, entitled "Idiosyncrasies of Chimney Draught," was discussed by Messrs. Jacobus, Le Van, Woolson, Scheffler, and Kent.

The second paper was by William Kent, on the "Influence of the Steam-jackets of the Pawtucket Pumping Engines," which was discussed by Messrs. Jacobus, See, Wheeler, and Dean.

By vote of the meeting, the papers set down for the session of Friday morning were to be taken up at this, Thursday's session, in order that after the close of the festivities on Thursday evening the members might disperse without sacrificing any session. This policy was made possible by the fact that the list of papers at this meeting was unusually short, and had been made so by virtue of the policy in consideration before the Finance Committee, that it was not right that the Publication

Committee should assume a heavy burden for the presentation of papers unless the dues were to be proportionately raised so as to relieve this stricture.

The Secretary then read to the meeting the following telegram:

SAN FRANCISCO, CAL., *Nov.* 19, 1891.

TO THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, 12 W. 31ST STREET, NEW YORK:

The Mechanical Engineers, the Technical Society of the Pacific Coast, the Governor, Mayor, and prominent citizens desire to express satisfaction on the vote of the Society to come to this State. A committee has been formed to make necessary arrangements, and a warm welcome is extended.

W. R. ECKART,
President.

The reading of this telegram was greeted with applause.

The papers of Messrs. Spaulding and Wood were then taken up, having been moved forward to this session; that by Mr. Spaulding being discussed by Messrs. Oberlin Smith and Macfarren, and that of Mr. Wood being discussed by Mr. Emery.

The paper by J. Wendell Cole, on "A Combination Iron and Oak Pavement," was discussed by Messrs. Fowler, Woolson, Sweet, Parsons, Oberlin Smith, Le Van, Ashworth, and Crane.

Topical questions were then taken up in between the papers, the topics being "The Purification of Bad Feed Water," "The Arrangement of Chimney Stacks for a Battery of Boilers," "The Question of Twelve Threads on a Half-inch Bolt instead of Thirteen."

The afternoon was left without assignment, as was done on Tuesday, parties being made up for members to visit different points of interest. In the evening, at Lenox Lyceum Hall, Madison Avenue and Fifty-ninth Street, a full-dress reception and ball were held, beginning at 9 P.M., the orchestra occupying the stage, leaving the floor available for dancing, the boxes being filled by invited guests to whom the Society desired to show honor. At half-past ten a collation was served in the room below, and dancing followed until an early hour.

With this reception the session was closed.

CCCCLXVI.*

PRESIDENT'S ANNUAL ADDRESS.

THE EVOLUTION OF AMERICAN ROLLING MILLS.

BY ROBERT W. HUNT, CHICAGO, ILL.

(President, 1890-1891.)

As annual anniversaries occur and the past year is reviewed, it is not always vouchsafed that the retrospection is as pleasant and satisfactory as the one in which we, as members of the American Society of Mechanical Engineers, can indulge to-night. Our past year as a Society has been blessed with success and prosperity in all directions. The Providence Meeting was the largest in attendance of any ever held by us, and, perhaps, by any scientific society in this country. The papers presented were fully up to the high standard which has prevailed since our organization, and the hospitality extended to us by the citizens of Providence will be warmly cherished in the memory of all who were fortunate enough to receive it. Not satisfied with what they did for the Society during the meeting, the Providence Local Committee presented the surplus from their entertainment contributions to our Library Fund. Thus while the "good things" of their hospitality can be with us but as memories, we will ever have in books the living thoughts of great minds to mark the interest felt in our Society by our Providence members.

This, our last meeting of the year, has by this evening's large attendance, and the honoring presence of so many fair ladies, stamped itself with success.

Our Society has 18 Honorary Members, 13 Life Members, 1,190 Members, 56 Associates, and 166 Juniors, giving us to-night a total membership of 1,443.

Matters of vital importance to the Society have been presented to the membership for consideration, and, while some of them

* Presented at the New York Meeting of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

now who have paid only $150, while the new ones who become such hereafter will have to pay $200. The question was, "whether it was proper to leave the old members on the same basis as the new members, or whether that extra $50 should be paid by the old members as well." Inasmuch as I have been recently spoken to by several out-of-town members, I thought I would mention the fact now.

Mr. W. S. Rogers.—I have had some experience in other societies in this matter of raising the life-membership fee, and it induces me to think that as quick as we put up the amount of the fee we shall cut down revenue from that source. In some cases the rise had to be revoked for this reason, and while I will be glad to have the increase tried, I think we may want hereafter to go back to the lower figure.

Mr. John Humphrey.—The objection may apply as well to all membership fees. I think those who came in first justly entitled to the place at the original rates, and $200 is certainly no more than proportionate to the advance in other fees. In a pecuniary sense, life-membership becomes simply a question of interest upon a single advance payment, and the interest on $200 is less than the $15 annual dues. In fact, proportionately less than at former rates. Therefore, members will be quite as likely to pay $200 in the future as the $150 before the dues were increased.

Mr. Hawkins.—It seems to me that all the objection so far raised to increasing life-membership fees applies to what we have already done in the case of members, etc. If there is any injustice in it there is as much injustice to the ordinary members, juniors and associates, and that we would have just as good right to call on the old members, juniors and associates for the additional fee that we are now going to raise it to, as to call upon life members for it.

Mr. Woodbury.—It appears to me that this motion to increase the membership fees for life members is entirely consistent with the action to increase the ordinary membership fees. I do not perceive how there is any inconsistency in two sets of life members who have paid different fees any more than would apply to the whole membership, and no one would ever propose the idea of charging additional on the back dues of all the other members, going back to the first organization of the Society. The whole spirit of business policy, whether applied to commercial practices or to public affairs, is entirely opposed to retroactive payments, and I do not believe that anything of the kind will be contemplated in connection with the proposed action before the Society.

The motion to raise the life-membership fee being formally put by the President was then carried unanimously.

The fourth amendment was presented by D. K. Nicholson at the last meeting. It proposed to amend the Article No. 31 so as to have it read as follows:

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 at least two nominees for each and every office 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.

In presenting this amendment the following discussion was held:

The President.—As yet we are without a general expression from the membership on this. But I am authorized by the Council to present to you their views as embraced in this resolution, which was adopted yesterday:

Resolved, That the proposition to have two candidates named to each vacancy in the Society's roll of officers each year, does not commend itself to the judgment of the Council, and they think if adopted that it will not conduce to the harmony and success of the Society.

You will appreciate, gentlemen, that the Council, in submitting this and the preceding resolution, do it simply because they feel it to be their duty, having been elected by you to conduct the affairs of the Society, and we think we do not presume too much in believing that we still have your confidence—that nothing has happened to destroy that. This matter has been carefully discussed by us in all its lights and bearings, and the resolutions reflect our judgment; but of course the decision rests with you. Does anybody second Mr. Nicholson's motion?

The Secretary.—I have, Mr. President, two or three letters.

The President.—The motion is not seconded yet.

Mr. Spangler.—I second it, sir.

The President.—Mr. Spangler seconds it. It is now before you, and the Secretary has two or three letters which he will read.

The Secretary read letters from Mr. Randolph, Mr. Charles A. Moore, Mr. J. Leon Gobeille, and Mr. Jesse M. Smith, the two latter emphatically urging valid reasons against the proposed plan.

Mr. Arthur C. Walworth.—I do not see the slightest necessity for this amendment. It seems to me that it would soon defeat itself. I do not see what we should gain by it. If you look at this amendment carefully you can see what could be done by the Nominating Committee. That Nominating Committee is an odd number, and three of that Committee carry the report of the Committee with it. Suppose that three of that Committee had a man that they wished to nominate. They can report that name; then they can report the name of a man who is unpopular, unknown—perhaps the last man elected in the Society. They could practically make one nomination if they wished. I do not say they would do it; but we should look at all contingencies. It seems that this Article 31 is very wisely drafted. If there are five discontented men in the fourteen or fifteen hundred that constitute our membership, they can send in a nomination, and that nomination will appear on the ticket without any distinguishing mark. It is the Australian system brought to perfection; and I believe in leaving well enough alone, and not trying to improve a system which seems to me to be perfect already.

Mr. C. J. H. Woodbury.—The growth of this Society has been wonderful, its transactions valuable, its meetings interesting conventions for the interchange of opinion and experience in their application to the latest methods of engineering practice; and, in short, it has always performed in the highest degree all of the functions to be expected from a Society of American Engineers. And these results have been obtained as a logical result of the universal harmony which has always prevailed in the Society. The officers have always been closely in touch with the membership as a whole, and I want to ask you, in the name of common sense, considering all the results which have been achieved, is any one going to be wise enough to better that matter? (Applause.) As the President has refreshed our memories with the technical matter of our rules, it would seem that each and every contingency has been fully provided for. The Nominating Committee is freed from local bias by its selection from various portions of the country, and its duties are limited to nomination of new men, thus causing the most rapidly possible rotation in office. How are we going to improve the matter by a double nomination? If, however, there should be any person nominated, under any possible contingency, who should not be acceptable to the membership, then a very small proportion of the membership has the full right and privilege under our present constitution of making a double nomination; and I believe, as one of the members who does not have an opportunity to attend the frequent meetings of this Society, that we are entirely satisfied with the result of the administration of the past officers of this Society. I do not see why we should not have the same satisfaction with respect to any future administration under the same methods that we have so successfully followed in the past. Suppose that there should be a double nomination. I do not care what may be the individual feeling of the nominee: no person could restrain his friends from running to extreme measures, and many of them might be injudicious, and, in fact, unwarrantable. I am a member of two or three societies which have had double nominations, or bolting nominations, and in every case the results have been deplorable, and I think that the American Society of Mechanical Engineers is wise enough to profit by their example. (Applause.)

The President.—By an affirmative vote, you understand that you amend the constitution. A negative vote leaves the constitution as it is.

A vote was taken, and the amendment was declared defeated.

The President.—This is the time for any new business. If any member has any new business to offer, we should be pleased to receive it.

I am authorized by the Council to read this resolution:

Resolved, That the invitations from the Pacific Slope, to hold the Spring Meeting in 1892 in the city of San Francisco, be accepted.

This was the official action taken yesterday. (Applause.)

After some announcements by the Secretary, the morning having been exhausted, the session adjourned, to meet at half-past eight in the evening.

No definite assignment was made for Tuesday afternoon, but the half-day was put at the service of the members in attendance at the meeting either to utilize it in social reunions at the rooms, or for the purpose of visits to points of professional interest in and around New York. Arrangements were made by the Local Committee to give opportunity to visit the following places:

I. Morgan and Quintard Iron Works.
II. North River Iron Works.
III. Pond Machine Tool Works.
IV. Stevens Institute of Technology.
V. Columbia College, Schools of Arts, Law, Mines.
VI. " " School of Medicine (College of Phys. and Surg.).
VII. Pratt Institute.
VIII. Washington Bridge.
IX. Ship-yards and Works Samuel L. Moore & Sons' Co.
X. World Building—Equitable Building.
XI. Metropolitan Museum of Art.
XII. Museum of Natural History.
XIII. New York Oxygen Co.
XIV. Ball & Wood Engine Co.
XV. Harper Bros.
XVI. Power House, 125th St. Cable Road.

The number appended to each place corresponded to a number upon a card which gave directions as to the routes to be followed to reach the object desired and the persons for whom to inquire upon arrival. This plan appeared to work very well and was generally approved.

THIRD SESSION. TUESDAY EVENING, NOVEMBER 17TH.

The professional papers of the session were taken up as follows:

Three papers by A. F. Nagle, of Chicago, entitled "Experiments to Determine the Rate of Rise or Fall of a Mercurial Thermometer under Different Conditions," "Test of Indiana Block Coal at the Chicago West Side Pumping Station," and, "Limitations of Steam-engine Economy," were discussed, respectively, by A. C. Walworth, William Kent, C. E. Emery, Professor Lanza, and O. C. Woolson.

Charles T. Main read a paper on "The Value of a Water Power," which was discussed by Messrs. Emery and Cartwright;

and a paper by Samuel McElroy, on "Brooklyn Pumping Engines of 1860," was discussed by Messrs. Emery and Kent.

Interspersed among the papers, to give life and variety to the sessions, were certain Topical Queries on "Method of Standardizing Sizes for Keys," on "Portable Apparatus for Light for Out-door Work," on "The Best Design for Stopping and Starting Shafting on Any Floor in a Mill."

Wednesday was devoted to excursions and festivities. The steamer *Laura M. Starin* left the foot of West Twenty-third Street at 10 A.M., and carried a party of 300 ladies and gentlemen to the Navy Yard, where the cruisers and battle-ship were inspected by the party, with a few of the shops. Returning to the boat, luncheon was served in transit to the works of the De La Vergne Refrigerating Company, at the foot of East 138th Street. After inspection at this point, the boat brought the returning party to the foot of East Thirty-first Street, where it was dispersed.

In the evening, a theatre party was given by the Local Committee to their visiting friends and their ladies, at Palmer's Theatre, Broadway and Thirtieth Street. The party numbered 354, and occupied nearly the entire ground floor of the house. A reproduction of the Society's pin badge in gas-jets decorated the exterior of the building, and souvenir programmes were distributed to each one as mementoes of the evening.

Fourth Session. Thursday, November 19th.

Professional session for presentation and discussion of papers. A paper by W. E. Crane, entitled "Idiosyncrasies of Chimney Draught," was discussed by Messrs. Jacobus, Le Van, Woolson, Scheffler, and Kent.

The second paper was by William Kent, on the "Influence of the Steam-jackets of the Pawtucket Pumping Engines," which was discussed by Messrs. Jacobus, See, Wheeler, and Dean.

By vote of the meeting, the papers set down for the session of Friday morning were to be taken up at this, Thursday's session, in order that after the close of the festivities on Thursday evening the members might disperse without sacrificing any session. This policy was made possible by the fact that the list of papers at this meeting was unusually short, and had been made so by virtue of the policy in consideration before the Finance Committee, that it was not right that the Publication

Committee should assume a heavy burden for the presentation of papers unless the dues were to be proportionately raised so as to relieve this stricture.

The Secretary then read to the meeting the following telegram:

SAN FRANCISCO, CAL., *Nov.* 19, 1891.

TO THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, 12 W. 31ST STREET, NEW YORK:

The Mechanical Engineers, the Technical Society of the Pacific Coast, the Governor, Mayor, and prominent citizens desire to express satisfaction on the vote of the Society to come to this State. A committee has been formed to make necessary arrangements, and a warm welcome is extended.

W. R. ECKART,
President.

The reading of this telegram was greeted with applause.

The papers of Messrs. Spaulding and Wood were then taken up, having been moved forward to this session; that by Mr. Spaulding being discussed by Messrs. Oberlin Smith and Macfarren, and that of Mr. Wood being discussed by Mr. Emery.

The paper by J. Wendell Cole, on "A Combination Iron and Oak Pavement," was discussed by Messrs. Fowler, Woolson, Sweet, Parsons, Oberlin Smith, Le Van, Ashworth, and Crane.

Topical questions were then taken up in between the papers, the topics being "The Purification of Bad Feed Water," "The Arrangement of Chimney Stacks for a Battery of Boilers," "The Question of Twelve Threads on a Half-inch Bolt instead of Thirteen."

The afternoon was left without assignment, as was done on Tuesday, parties being made up for members to visit different points of interest. In the evening, at Lenox Lyceum Hall, Madison Avenue and Fifty-ninth Street, a full-dress reception and ball were held, beginning at 9 P.M., the orchestra occupying the stage, leaving the floor available for dancing, the boxes being filled by invited guests to whom the Society desired to show honor. At half-past ten a collation was served in the room below, and dancing followed until an early hour.

With this reception the session was closed.

CCCCLXVI.*

PRESIDENT'S ANNUAL ADDRESS.

THE EVOLUTION OF AMERICAN ROLLING MILLS.

BY ROBERT W. HUNT, CHICAGO, ILL.
(President, 1890-1891.)

As annual anniversaries occur and the past year is reviewed, it is not always vouchsafed that the retrospection is as pleasant and satisfactory as the one in which we, as members of the American Society of Mechanical Engineers, can indulge to-night. Our past year as a Society has been blessed with success and prosperity in all directions. The Providence Meeting was the largest in attendance of any ever held by us, and, perhaps, by any scientific society in this country. The papers presented were fully up to the high standard which has prevailed since our organization, and the hospitality extended to us by the citizens of Providence will be warmly cherished in the memory of all who were fortunate enough to receive it. Not satisfied with what they did for the Society during the meeting, the Providence Local Committee presented the surplus from their entertainment contributions to our Library Fund. Thus while the "good things" of their hospitality can be with us but as memories, we will ever have in books the living thoughts of great minds to mark the interest felt in our Society by our Providence members.

This, our last meeting of the year, has by this evening's large attendance, and the honoring presence of so many fair ladies, stamped itself with success.

Our Society has 18 Honorary Members, 13 Life Members, 1,190 Members, 56 Associates, and 166 Juniors, giving us to-night a total membership of 1,443.

Matters of vital importance to the Society have been presented to the membership for consideration, and, while some of them

* Presented at the New York Meeting of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

have not received final action, the manner in which their discussion has been conducted proves that harmony of feeling which assures continued and increasing prosperity.

During the year a series of most delightful social entertainments has been given in our beautiful home, which were not only occasions of enjoyment to those present, but have drawn many members closer together, and intensified their affection for and interest in our Society.

We have received some valuable historical gifts, the commitment of which to our care illustrates the confidence felt in our future.

As evidences of appreciation, we have been given portraits of distinguished living men, and, in recognition of our affection for their memory, those of some of our distinguished dead.

Our walls are honored by them.

With this past, and such a future, as much assured as any human thing can be, may not I, as your President, feel happy?

You have seen fit to honor some of us by election to office; others are to be similarly recognized. I know that in the past all so marked by your favor sought to faithfully fulfil the trust; and I also know that those to come will profit by both the successes and mistakes of the past, and so give you a greater future. But remember that success has not come, and cannot come, from the efforts of your officers alone, but is of yourselves.

As my presidential address, perhaps I cannot do better than by recalling to you some of the more prominent facts in the history of the "Evolution of American Rolling Mills."

Ancient narrative is apt to be interesting to but one person, but when current and live events can only be fully appreciated by an understanding of preceding ones, the happening of which made them possible, the danger of garrulity must be risked.

As the production of railway bars or rails represents the largest volume of all the many manufactured forms of iron and steel, it is natural that we should first consider mills in which these are made. As most of you know, iron rails were rolled from a pile composed of a number of bars of wrought iron placed one upon the other, brought to a welding heat in a furnace, and then passed between the grooves of rolls, which welded them together, and gradually elongated and formed the mass into a finished rail. In the early days railway engineers were not only satisfied to accept these rails cut in much shorter lengths than now prevail, but, in fact, up to about 1859 would not receive them over 21 feet long. James M. Swank, in his *Iron in All Ages*, states that the first rails 30 feet in length were rolled by the Cambria Iron Company in 1855, but, there being no sale for them, the rails were placed in the mill yard tracks. He also says the first 30-foot rails rolled to fill an order were made by the Montour Company in January, 1859, for the Sunbury & Erie Railroad Company. I am also indebted to Mr. Swank's work for most of the following dates in relation to the earlier rail mills.

The first American rail mill—that is, one built to produce other than strap-iron rails—was the Mount Savage Works, situated in Allegheny County, Maryland, erected in 1843. Rolling began in 1844. Mr. Swank says that in honor of their first rail, which was of the U pattern, the Franklin Institute awarded a silver medal in October, 1844. The rail weighed 42 pounds per yard. Next came the Montour Works, at Danville, Pa. In October, 1845, in that mill was produced the first T-rail made in America. In May, 1846, the Boston Iron Works, of Boston, Mass., began rolling rails. June 18, 1846, Cooper & Hewitt rolled their first T-rail in their Trenton, N. J., mill. In September, 1846, the New England Iron Company, of Providence, R. I., started rail making. In November of the same year the Phœnix Iron Company, of Phœnixville, Pa., rolled rails. In the fall of 1846 rails were made at Brady's Bend, Pa., by the Great Eastern Iron Company, in works built expressly for their manufacture. These works were very extensive, taking the ore and coal from adjacent property owned by the company, and by means of their own blast furnaces, etc., producing the finished product. About the same time the Lackawanna Iron and Coal Company, of Scranton, Pa., went into operation producing rails from their own raw materials. Other mills were built or remodelled to roll rails, until in 1850 there were some fifteen rail mills in the country, but the commercial conditions were such that the spring of that year saw but two of them in operation.

I have named these early rail mills as matter of history. Some of them are now makers of steel rails in new plants, others are producers of other finished forms of iron and steel, while still others have gone out of existence. In some cases scarcely a vestige remains of a once great establishment. The mines are abandoned, the blast furnaces have but a few stones to mark their sites, and the rolling mills are so completely wiped out that not a trace of them remains. The once populous village of busy workers is now monumented by crumbling ruins of their homes. But not so the industry. The United States to-day leads the world in her rail production.

Our subject leads to the history of one of the survivors. The Cambria Iron Works were built in Johnstown, Pa., in 1853, that location being selected because of an abundant iron ore and coal supply; these two minerals being deposited in the same hills, and within a few yards of the selected blast-furnace and rolling-mill sites. The works were designed to produce iron rails alone. This particular locality had been somewhat exploited since 1809, with more failures than successes.

In view of recent history, I find as a somewhat remarkable coincidence that the first iron works located at Johnstown was a forge, built in 1809, and located on the banks of the Stony Creek, the waters of which were dammed to furnish power for its operations. A flood carried away this dam, and led to the removal of the forge over to the banks of the Little Conemaugh; these two streams uniting at Johnstown, and becoming the Conemaugh. When Mr. Swank made the chronicle, little did he or his readers dream of the greater flood which was to pour down the valley, to which this old forge had been moved for safety, and not only wipe out a large rolling-mill plant, but in addition cause the greatest disaster to life and property known to civilization.

The iron ore which had been used by the earlier establishments, and which led to the building of the Cambria Iron Works, was the outcropping of silicious carbonate. The operations of the smaller plants had been more or less satisfactory, but the consumption of the greater works soon exhausted the outcrop and compelled the use of the leaner ore, which produced iron of an inferior quality. When puddled it was both red and cold short. While the hardness incident to the latter gave good wearing rail heads, the red shortness rendered it difficult to obtain finished rails of which the flanges were not so badly cracked that they had to be thrown on the scrap heap. This happened even after passing through the elaborate system of patching and puttying up cracks which then prevailed, and was favored by the entire absence of inspection bureaus. Up to July, 1857, all rails at Cambria and other mills were rolled on non-reversing two-high trains of rolls. That is, there were but two rolls in a set, and as they were driven constantly in one direction, the metal which was being drawn into shape by their grooves could be rolled only in that direction. After each passage between the rolls, the pile, or bar, had to be passed back over the top roll, its revolution assisting in this. Of course, quite as much time was consumed in this passing back as in the opposite rolling or reductions. Not only was time consumed and the amount of production limited, but the metal under treatment lost heat, and thus augmented the difficulties in obtaining satisfactory welds of the several slabs of iron composing the rail piles and freedom from "red short" cracks in the finished rail. The difficulties at Cambria continued to increase, and, with no other available ore supply, were at last so serious that the prospects of the company were gloomy indeed. But, as has so often and so fortunately happened, the difficulties and seeming hopelessness of the situation forced a solution of the particular case, and, more than that, led to an invention which was destined to revolutionize the rail industry of the country.

Our esteemed fellow-member, John Fritz, was then the chief engineer of the Cambria Company. His keen mechanical perception and good judgment saw the solution of the problem. It was to save the time and heat lost in passing the bar back in idleness over the top roll; this could be done by adding another or third roll, and so making that which had been the top a middle one. This top roll revolving in the opposite direction to the middle one permitted grooves to be added, in which the metal could be reduced as it was brought back to the front of the train of rolls. To accomplish this successfully he invented the Fritz Yielding Hanging Guides and Driven Feed Rollers. This solution seems now such a simple one. But we must remember that at the time of its conception rolling mills were considered old institutions, and their designers and managers thought themselves, and were thought by others, to be very smart men. Moreover, there were difficulties in the construction and operation of this proposed mill which would appear only to those possessed of some practical knowledge. Indeed, some of those high in the councils of the Cambria Company entered solemn and official protests against that crazy man, Fritz, being permitted to waste the company's money. Its affairs were badly enough off, as it was, without adding this foolishness.

In addition, some of Mr. Fritz's brother engineers and intimate friends compelled themselves, as a matter of fraternal duty, to labor with him against his folly, and thus prevent his scattering to the winds his most excellent and hard-earned reputation. In spite of all, he had the courage of his convictions, and the new mill was built. Let me tell of its start in his own words: "The three-high mill was started on Wednesday, July 29, 1857. We charged and heated six rail piles. We rolled three of them, making perfect rails, when. the eccentric of the rail mill engine became hot, and bent the rod badly. Having tried the mill, and all gone perfectly, we stopped, resuming work on Friday morning, and continuing regularly until the usual quitting time on Saturday afternoon. Alexander Hamilton, then and now the superintendent of the mill, and I left the works about half-past five in the evening, congratulating ourselves that our troubles, so far as the rail mill was concerned, were practically ended. About seven o'clock that night I heard the mill whistle blow for fire, and at once started for the works, to find the mill in flames from one end to the other. In less than an hour's time the whole structure was consumed. I will leave you to imagine how I felt while seeing in one short hour our best efforts and the labor of a whole year destroyed. But we went at it again, and in about one month's time were again making rails. As almost every person who was supposed to know anything about rolling mills had predicted a failure, the story got out that we had tried the mill, and, finding it a failure, had burned the whole thing down to hide the blunder."

I doubt if ever during Mr. Fritz's subsequent eventful life he has had to carry quite as heavy a mental load. Since then mighty works have grown from his designs, and under his charge. Millions of dollars have been invested on his judgment. The monster steam hammer of the world is his creation; but I venture to say that, while waiting for the shock of its first 125-ton blow, his anxiety was but as that of a child compared to that felt while the first rail pile was passing between the rolls of his 1857 mill. In designing and perfecting this mill, Mr. Fritz was assisted by his brother George, who, upon John's resignation from the Cambria Company in 1860, to organize the Bethlehem Iron Company and design and build their works at Bethlehem, Pa., succeeded to his position as chief engineer of the Cambria Company.

The Fritz mill was rapidly adopted by the rail mills of the country. Mr. Fritz protected himself by patents, which were soon acquired by a combination representing the larger rail mill organizations, under the title of John M. Kennedy & Co., of Philadelphia. In 1864 and 1866 I made in their interest a tour of inspection of all the rail mills of the United States, and found the Fritz mill in universal use. Some had not secured proper licenses, but I believe all ultimately settled. The iron rail industry of America grew rapidly, but as the traffic, weight of equipment, and speed of trains of the railroads increased, the demand for more enduring rails became imperative. This resulted after a long and busy period of experiments in the invention and adoption of Bessemer steel rails.

I have already placed upon record, in a lecture before the Franklin Institute, Philadelphia, January 21, 1889, that the first commercial rolling of steel rails was at the Cambria Works in August, 1867, on an order from the Pennsylvania Railroad Company, from steel made by the Pennsylvania Steel Company at their Steelton plant. These rails were rolled on a three-high 21-inch train, on which the heavier sections of iron rails had been rolled. At first the steel ingots were drawn into blooms under steam hammers. George Fritz concluded, as stated by me in the "History of the Bessemer Manufacture in America" (*Transactions American Institute of Mining Engineers*, Vol. V.), that this was not the proper manner of treating the material. He had blooming rolls prepared, and placed in one set of the 21-inch rail train housings. A. L. Holley was then in charge of the Pennsylvania Steel Works, and, sustaining Mr. Fritz in his experiments, had ingots 8½ inches square cast and sent to him. These were bloomed by the rolls to 6½ inches square, recharged in the heating furnaces, wash-heated, and then rolled into rails. This practice was successful, and I believe this was the first cogging, or blooming, mill.

In 1868 Mr. Holley relinquished the management of the Pennsylvania Works, and later in the year again took charge of the Troy Bessemer Works, which he had originally built. In January, 1871, he started a 30-inch three-high blooming train, provided, front and back, with lifting tables, containing loose rollers and raised by hydraulic power. The rolls were turned to receive 12-inch square ingots, which were cast heavy enough to make two rail blooms. These ingots, after being placed on the loose rollers of the front table, were pushed into the rolls, both on the front and back sides, by hand, it requiring eight men to operate the mill. This train had the top and bottom rolls stationary. The middle roll was moved up and down by four screws running through the bolsters carrying its necks, these screws being rotated by a friction clutch, which was driven by a belt off the main shaft of the engine driving the train, and controlled by a hand lever at the end of the rolls. There were four grooves in each roll, and by the travel of the middle roll the possible reductions were increased. The mill worked well and was a great advance in the art.

On July 10, 1871, the Bessemer works of the Cambria Iron Company made their first blow, the steel from which was rolled in a blooming train designed and built by George Fritz. As I have previously recorded, he and Holley were close personal and professional friends, and their interchange of ideas and mutual assistance was frank and full. Fritz was an ardent admirer of Holley, but he could not simply copy any man; hence, while cheerfully giving Holley credit for everything taken from him, he introduced many new ideas in his arrangement of the Bessemer plant and blooming train. In the latter he made the middle roll stationary and moved the top and bottom ones, thus saving time in setting the passes; and as distinctively new and important features he arranged to drive the rollers in the tables by means of gears, controlled by friction clutches, deriving their power from the train engine. He also invented a hydraulic pusher, working between the rollers, for turning over and moving the ingots on the tables. These last two features constituted the Fritz blooming mill patent. By these improvements the mill force required was reduced to four men. In 1876 I wrote: "The merits of rolling as compared with hammering had been fully discussed between Mr. Fritz and Mr. Holley, and they had at various times gone over the numerous details of a blooming mill, and Mr. Holley, as already stated, had built one at the Troy Works. Mr. Fritz had availed himself of the benefit of the extensive knowledge and sound judgment of his brother, John Fritz, of Bethlehem, Pa., and the result of all was the Johnstown Blooming Mill, which marked a new era in the Bessemer manufacture While living to see many difficulties overcome and great progress made, George Fritz died too soon, his country losing one of her noblest and ablest sons. He died August 5, 1873."

In October, 1873, the works of the Bethlehem Iron Company, of Bethlehem, Pa., began operations, having been designed and built by John Fritz, Mr. Holley being connected with him as consulting engineer. Mr. Fritz made an important improvement in his blooming mill in dispensing with the friction arrangement for driving the table rollers by putting in a pair of small reversing engines, the power being transmitted through a belt. Holley soon improved on this by using an intermediate gear in place of the belt. Of course, many improvements have been made in the later blooming mills; in fact, Mr. Fritz himself soon built another one in which he made the rolls heavier and longer, thus being enabled to add another groove, which allowed him to make all three rolls stationary.

It must not be supposed that the innovation of rolling ingots into blooms was received without dissent. In fact, the *Transactions* of the American Institute of Mining Engineers, as well as the files of some of the technical newspapers, record animated discussions in which the advocates of hammering sought to prove that good and uniform results could not be obtained by rolling. But the tide could not be turned back, and about all the rail steel of the world has been bloomed in rolling mills for many years. It has always been a little dangerous to say "can't" in connection with the Bessemer manufacture.

As an instance of the prejudice against rails rolled from blooms other than those which had been produced by hammering, and the deep amount of knowledge of the business which then prevailed, I recall that it was thought necessary by the Cambria management, at whose mill hammering had been abolished, to invite J. Edgar Thomson (then the president of the Pennsylvania Railroad, and I believe the first American railroad president to recognize the merits of steel rails) to visit their works, where they were filling a steel rail order for his road, and witness some drop tests of rails, which it was hoped would convince him that even if hammering the ingots ordinarily gave better results than all rolling, the system of bottom casting the ingots then practised at the works gave such superior steel that their resulting rails were equal to any in the manufacture of which hammering had played a part. Mr. Thomson made the visit, had the theory explained to him, witnessed the tests, seemed satisfied—at all events, did not talk back—and every one was happy. While those drop tests were being made, how some of us did tremble from fear that one or more might break; but luck was on our side that day.

In 1868 the Freedom Iron and Steel Company, with works near Lewistown, Pa., started operations in a plant of almost entirely imported English machinery. In fact, the Bessemer blowing engine, which was built by I. P. Morris & Towne, of Philadelphia, was the only item of any importance of American construction. The company's original intention was the manufacture of Bessemer steel plates and tires. They put in a reversing plate mill, driven by a reversing Ramsbottom engine. This was soon changed to a rail mill, and was the first reversing rail mill in America. The works were not successful, continuing in operation but about one year.

The Bessemer machinery was sold later to the Joliet Iron and Steel Company, of Joliet, Ill. Later still the Cambria Iron Company purchased the rail mill and engine, and set it up at Johnstown as a blooming mill, in place of the original Fritz mill, which was too light to handle the 18-inch square ingots they desired to cast. The Freedom mill is still in existence, but the Cambria Company have built another, also of the two-high reversing type. The English tire rolling mill at Freedom was a fine tool, and became the property of the Standard Steel Company, who operated it in the original location, rolling open-hearth steel made by the Otis Company, of Cleveland, Ohio.

In August, 1881, the Pittsburg Bessemer Works, at Homestead, Pa., started. In their construction it was sought to keep clear of all patents controlled by the Bessemer Association. I presume this, together with their desire to produce billets and slabs of various sizes, led to the adoption of a two-high reversing blooming mill, designed and constructed by Mackintosh, Hemphill & Co., Limited, of Pittsburg, Pa. This was the first complete mill of that type of American construction. But in 1879 Messrs. Shoenberger & Co., of Pittsburg, had experimented with a reversing blooming mill; and experience gained from this mill guided the firm named in designing the Homestead Mill. These works are now part of Carnegie, Phipps & Co.'s Homestead plant. The South Chicago Bessemer plant and rail mill of the North Chicago Rolling Mill Company went into operation in June, 1882. This mill was designed by Henry C. Kriete (since deceased), the mechanical engineer of the company, and most of the machinery was built in their own shops. Mr. Kriete adopted a Fritz blooming mill, but a reversing two-high rail mill, this being the first reversing rail mill built in the country.

In May, 1883, the Scranton Steel Company, of Scranton, Pa., started. Their works were designed under the personal direction of W. W. Scranton, president of the company. He imported from England both reversing blooming and rail mills and their engines. It is needless to say that during the time covered many changes had been made in the various mills of the country, and none were exact copies of any other; each engineer seeking to improve on previous construction, and generally succeeding. But while there were many changes in the details, the general plans of the three-high blooming and rail mills remained the same.

In the early days of iron-rail making the only means of cold straightening the product was by the blows of a heavy sledge. In fact, I think it was not until about 1856 or 1857 that a straightening press was employed. It is needless to say that the sledge process was slow, and would have been inefficient on heavier sectioned rails of either iron or steel. After the introduction of the cold straightening press the hot bed remained as a weak spot. The rails had to be hot curved against their heads, and to do this by hand, and then drag them down the hot beds by the same power, was both slow and exhausting work. The several mill managements devised various plans to dispense with as much labor as possible in handling the rails from the time they left the finishing pass in the rolls until delivered on the cooling beds. In many mills this was successfully accomplished up to the hot curving. At that point there was a long halt.

In 1875 A. J. Gustin, then the superintendent of the St. Albans, Vt., Rail Mill, invented a rail-curving machine, which, together with power appliances for dragging the curved rails down the hot bed, he patented. These devices were soon adopted by many mills. Later William Clark, of Pittsburg, invented a rail curver which many prefer to the Gustin. Some mills have adhered to arrangements of their own, but the Gustin and Clark devices may be accepted as the American practice. Without some such arrangements the increase of product which has been so great would have been an impossibility.

Even before the introduction of steel rails a number of inventors had sought to design rail mills which would be practically automatic in their action, but I believe down to the time to which we have traced the art none were actually built. Holley used to say, in that spirit of prophetic jest so constant with him, "that the day would come when we would start a rail mill on Monday morning and then go home after locking the doors, only returning each morning to count the rails that had been made during the preceding twenty-four hours, no other manual labor being necessary." We have not yet reached that point, but how some of us have mourned that he could not have been spared to see what we have accomplished and glory with us in it. He knew it could be done, but for some incomprehensible reason the way did not open to his mind.

In March, 1884, I introduced driven tables in front of the finishing rolls of the rail train of the Albany and Rensselaer Iron and Steel Company, Troy, N. Y. They worked so well that I put an automatic arrangement in front of the roughing rolls. This was more particularly designed by Max M. Suppes, member of this Society, then the master mechanic of that department, and who rendered me valuable assistance during all of my experiments. This last table was also successful. Of course, we protected ourselves by letters patent. Later I placed tables on the catchers' side of the train. Captain Wm. R. Jones, then the general superintendent of the Edgar Thomson Works of Carnegie Brothers & Co., at Bessemer, Pa., at once advised his firm to secure authority from me to use my patents. As he proposed a different arrangement for the roughing rolls, he did not care for the Hunt-Suppes claim. The arrangement having been consummated, he constructed an elaborate system of tables, both front and back, all of which he subsequently patented, and we joined interests in the same. As an instance among many of his great heart, he wanted to give me the whole thing, saying: "You were the first to put this matter in a practical shape, and deserve it. I have only gone further with your ideas." Of course, I did not accept his proposition, and I know it was to his regret.

Fate sometimes seems too hard to be just. That my friend should have been cut off in the middle of his great life work, and by such a death, was one of those catastrophes the justice of which is beyond human comprehension. Yet I believe he died as he would have chosen—foremost in danger. Often had he risked his life amid shot and shell, and where his example was the incentive to others for higher daring. He escaped hurt or death on the battle-field, but yet to die at what he thought the post of duty. An ardent admirer of Captain Jones, and whose own soldier life had given him a closer sympathy, said, when he heard the details of the accident: "Another general shot on the picket line." This was true, but those who know the history of that fateful afternoon appreciate why he could not be content save at the very front. I know he said to himself: "This company has trusted me; has to-day shown that they appreciate and love me. Their losses are mine, and until I know that furnace is safe I cannot go home." It meant his death; but to him better that than the suspicion of want of devotion. And was he not right? It must come. Better let it find duty overdone than ever so slightly shirked. Please excuse my digression while paying this poor tribute to Captain Jones's memory. It deserves more than I have said. Such men have made, have saved, and will continue to be our nation.

Before the introduction of automatic appliances from 15 to 17 men were required to operate a three-high rail train. The tables of which I have spoken reduced this number to five, including the roller in charge of the train. The next mill to put in automatic tables was the Pennsylvania Steel Company, at Steelton, Pa., in 1884, under licenses from Mr. Suppes and myself. The Joliet Steel Company, at Joliet, Ill., had been experimenting for some time with a large-sized working model of an automatic rail mill, from designs of H. S. Smith, general manager; Charles Pettigrew, chief engineer; and F. H. Treat, the latter gentleman being especially engaged upon them. In July, 1885, they altered their blooming and rail mill to conform to the plan, which proved to work as well in actual practice as in experiment. They kept clear of the Jones, Suppes, and Hunt patents, except in one particular, and for this they have since taken a license. For some time this mill has been rolling more billets than rails, the practice being to reduce a 15-inch ingot to 4-inch billets at one heat. The product has been brought up to 500 gross tons of such billets per turn, or 1,000 tons per day.

The Union Steel Company were reorganized in 1885, after their failure of some years before, and the Bessemer works, blooming and rail mills, completely rebuilt by our fellow member, Robert Forsyth. While his company secured licenses from our party, he made radical changes in the arrangement of his tables, and, though hampered for want of room, with great success. The new mill started in 1886. He arranged his mill to avoid reheating the rail blooms, although they had to be carried a long distance on driven rollers, from the blooming mill to the rail mill tables, and make two right-angle turns in the journey. The Union Mill's best record on standard sectioned rails is, for 24 hours, 1,312 gross tons; for month, 28,490 gross tons of rails.

In the same year Cambria remodelled the mill on which the first commercial American steel rails were rolled, and put in automatic tables, designed by Joseph Morgan, Jr., a member of our Society. This company have since taken licenses from Jones, Suppes, and Hunt. In 1887 the Worcester Steel Company, of Worcester, Mass., made terms with us, and added tables to their rail train, which were designed by C. M. Ryder.

The Edgar Thomson Mill had been doing great work, but neither Captain Jones nor his superior officers were satisfied with being so closely pressed in the amount of rails produced by the Union Works, the South Chicago Works, and the Scranton Steel Company. Instructions were given him to build the best rail mill he knew how, regardless of cost. This order was obeyed in every particular, and resulted in the present Edgar Thomson Blooming and Rail Mills. The first rails were rolled in the new mill in 1888. Since then the old mill has been idle.

In designing his new mill, Captain Jones made some radical departures. He adhered to the reheating of the blooms after leaving the blooming mill, but caused them to pass through heating furnaces on their road to the rail train. This train he divided into three sets, the first two with three-high rolls, the last with but two, all of 24-inch pitch, each set being driven by its own engine, and provided with automatic tables. In the first or roughing rolls five passes are made. The bloom is then carried by driven rollers to the second or intermediate train, in which it receives five more passes, and is then carried to and through the finishing pass in the two-high set. These trains are placed in echelon and far enough apart to permit three 30-foot rails to be rolled. The mill is a very simple one, and has many mechanical arrangements which make roll changing the work of but a few minutes, while every part of each set of rolls is easy of access. After the rails leave the cambering machine they are carried down the hot bed by power and automatically distributed to the cold straightening presses. This arrangement is simple, substantial, and inexpensive of operation.

Since Captain Jones's death some changes have been made, but none of a radical character, and some of them had been foreseen by him. The best product of this mill has been 941 gross tons of rails in 12 hours, 1,933 gross tons in 24 hours, and 33,181 gross tons in one month. In 1886 the Indianapolis Rolling Mill Company, of Indianapolis, Ind., added an open-hearth steel plant to their works and spent considerable money in remodelling their rail mill, and among other things adopted automatic devices for handling the material at the rolls. This was done under the charge of and from plans of D. H. Lentz. The mill worked very well, but commercial reasons soon caused rail rolling to be abandoned, and it has never been resumed.

The great progress made in the introduction of street car roads and the increased weight of rolling stock required for the cable and electric systems, as well as the requirements by municipal governments that the intermediate pavement, as well as that outside of the rails, should be of good character, led to the invention of the girder street car rail. The first to secure recognition was the invention of T. L. Johnson, of Cleveland, Ohio. These were for some time rolled by the Cambria Company, but the demand increased so rapidly that Mr. Johnson's company decided to build a mill of their own. This was done under the direction and from the plans of A. J. Moxham, the president of the Johnson Company. The mill was located at Moxham, near Johnstown, Pa., and commenced rolling in 1888. Owing to the extremely difficult character of the proposed sections, Mr. Moxham decided upon an English two-high reversing mill, importing his engines and train. I think he was wise in his selection of type of mill, as in such a one the piece can be entered while the rolls are revolving very slowly, and if such entry is not satisfactory the mill can be reversed at once and the piece backed out. At all events, the works have been very successful. But it must not be inferred that such sections cannot be rolled on three-high trains, because it has been and is being done; this at the "Old Mill" of the Cambria Company, the Tidewater Steel Company's Chester, Pa., plant, the North Chicago mills of the Illinois Steel Company, and notably by the North Branch Steel Company, of Danville, Pa.

In 1886 the Duquesne Steel Company, of Pittsburg, were organized, and located works at Oliver on the Monongahela River, some fourteen miles above the city, and about two miles above and on the opposite side of the river to the Edgar Thomson Works. After building a converting and reversing blooming-mill plant, operations were suspended. Late in 1887, or early in 1888, a new company was formed under the name of the Allegheny Bessemer Company, and the partially built works were completed by the addition of a three-high rail mill. This was built by Mackintosh, Hemphill & Co., Limited, under license secured from the parties controlling the Joliet table patents, and from the designs of William Clark, of Pittsburg, who was interested in the new company. This mill was composed of two trains of rolls, standing one in front of the other, and back of the blooming mill. Each train had two sets of rolls and was driven by its own engine. Good work was accomplished, but neither the converting works nor blooming mill was able to furnish steel enough to test its capacity. The first rails were rolled in 1889. Carnegie, Phipps & Co. have since acquired the works, which are now principally devoted to the making of blooms and billets.

After the consolidation of the Joliet, Union, and North Chicago companies into the Illinois Steel Company, it was decided by the new organization to dismantle their two-high reversing rail mill at the South Chicago Works, and replace it by a three-high mill with automatic tables, again taking license from us. Mr. Forsyth had become the chief engineer of the new company, and built this new mill. In its arrangement and construction he again made changes from anything which had gone before, and his results are speaking loudly for themselves. He divided his rail train into two sections somewhat like the Allegheny Bessemer Works, each having two sets of rolls in three-high housings, and each section driven by its own engine, with automatic tables front and back of the rolls. These trains stand in echelon with the blooming mill, which is a 40-inch three-high train. The rail mill rolls are 27-inch pitch.

In the practice of this mill, as at the Union, the ingots are kept after being cast in a perpendicular position; they are charged upright in gas-fired, soaking-pit furnaces of the Hainsworth type, but which are of Mr. Forsyth's own design. After the ingot is reduced in the blooming mill it is carried by power rollers toward the first rail train and through a shear by which the end which was the top of the ingot is cut off and the long bloom sheared in two, each half making two or three rails, according to the weight of the intended section. The first half at once passes through the rail-roughing rolls, the second one being held for a few seconds, or until the first has made three passes, when it is also sent forward. If from any reason the bloom when sheared should have become too cold to be safely and successfully finished, a power overhead traveller is provided to carry it at a right angle into a wing at the side of the mill, in which heating furnaces are located, with a Wellman charging and drawing crane in front of them. When sufficiently heated the same tool conveys the steel back on to the table rollers.

By this arrangement cold cobbles or other rail blooms can be heated and delivered to the rolls. In the roughing rolls the bloom receives five passes in three-high rolls. It is then passed to the second roughing tables, and is given three passes in three-high rolls. The partially formed section is elevated to the back tables of two-high rolls, and making one pass through them reaches a dummy table in front, from which it slides down on to driven rollers, and is by them carried back to the three-high set of rolls, which are in line with the first roughing rolls and driven by the same engine. In these it receives four passes, making in all thirteen rail mill passes. It is now a finished section, long enough to cut into three 30-foot rails. This is done at one operation by four saws. After passing through the cambering machine the rails are carried by power down the hot beds. When sufficiently cool they are loaded by power on to a spider car, which is handled by a special locomotive. The rails are conveyed to the several cold beds, located conveniently to the cold-straightening presses, and are unloaded on to these beds by an automatic arrangement of arms or levers, receiving their power from steam taken from the locomotive boiler.

Up to date the best record of the South Chicago Works on standard rail sections is: In 12 hours, 845 gross tons of rails; 24 hours, 1,571 gross tons; week, 8,152 gross tons; month, 34,381 gross tons. Owing to the depression in the rail market, the Union Works of the Illinois Steel Company have not been running steadily for some time, but on October 30, 1891, the mill was rolling light-sectioned rails, weighing 35 pounds per yard, and by way of keeping their South Chicago friends from forgetting their existence they took occasion to make a record on such work. The rails were rolled direct from 15-inch ingots, without reheating, and the result was: Day turn, 3,298 rails; night turn, 3.069; making 6,357 rails for the 24 hours, and weighing 989 gross tons. Three hours and 28 minutes out of the 24 were lost from various causes, such as changing passes, dressing rolls, etc. As might be expected, South Chicago soon sought revenge by rolling 3,540 rails of 40-pound section in 12 hours. These were also rolled direct from 15-inch ingots.

The last American rail mill to join the sisterhood was the Sparrows Point plant of the Maryland Steel Company, a very near connection of the Pennsylvania Steel Company. These works are situated on Patapsco River, a few miles from Baltimore, Md. They are not yet fully finished, but are designed to be among the best, if not the best, in the country. F. W. Wood, president and general manager of the company, and also the general manager of the Pennsylvania Steel Company, designed them, and he has spared neither money nor brains. The blooming mill is a two-high reversing one, built by Messrs. Mackintosh, Hemphill & Co., Limited. The rail mill is three-high, and consists of three sets of 27-inch rolls standing in line, with a Porter-Allen engine at each end of the train. Either engine is calculated to have power enough to do all the work, but it is intended to employ both. The rolls have automatic tables, and the work is transferred sideways, as in the earlier mills. It is expected to roll long lengths. The Lackawanna Iron and Steel Company have acquired licenses to use the Jones, Suppes, and Hunt patents, and I believe propose to put automatic tables in their upper works at an early day.

COLD ROLLING.

I come now to speak of what may more appropriately be termed a process, because the accomplishment was reached without a special mill. Of course, the metal of the rolls and the grooves in them had to be of a special character, but the work was done on an ordinary merchant bar mill. I refer to the cold rolling of iron. This was invented by Bernard Lauth in 1859. His patent was dated August 23d of that year, and the process became a distinctly American one. Jones & Laughlins, of Pittsburg, Pa., acquired the sole control in this country, and derived fame and fortune from it. While a great deal of cold rolled iron shafting and other articles is still used, the cheapening of steel has caused that metal largely to replace iron, and its greater stiffness has rendered unnecessary its being cold rolled for most purposes.

AMERICAN PLATE MILLS.

In 1864 Mr. Lauth patented another invention, and that was the three-high plate mill, with the diameter of the middle roll much smaller than the other two. Much of what I have said in favor of the three-high rail mill applies to this type of plate mill, and it soon became the American mill, and was also largely adopted in other countries. Mr. Lauth has been a constant experimenter, and has made many rolling-mill improvements of great originality and value. His name must always rank high among those who made successful the iron and steel industry of his country. American plate mills have developed in many points, as the greater requirements of steel made more powerful trains a necessity.

The plant of the Otis Steel Company, of Cleveland, Ohio, designed and built by our fellow-member, S. T. Wellman, in 1873–74, and started in 1875, afterward added to and improved by him, was for a long time the most complete one in the country. The increasing demand for steel plates, also for armor and other heavy plates, has led to the building of other large mills, notably by the Wellman Steel Company, of Thurlow, Pa., by Park, Brother & Co., Shoenberger & Co., Linden Steel Company, Spang Steel and Iron Company, etc., of Pittsburg, and, particularly, Carnegie, Phipps & Co., of the same city. The latter works possess some powerful mills, which have been lately increased. The universal mill has been largely employed in America, but while the original designs have been added to, I think Wagner, of Austria, deserves credit as the original inventor. In 1853 Charles Hewitt, since deceased, designed and built for his firm, Cooper, Hewitt & Co., of Trenton, N. J., a beam mill on the universal principle which was a radical departure from all previous plans.

WIRE ROD ROLLING.

Previous to 1869 all wire rods were rolled in this country upon ordinary guide mills, the manipulation of the material being entirely by hand. Billets of about 1⅛ inches square and 18 pounds in weight were used, and 6 tons of No. 4 rods was regarded as a good day's or turn's work. In the spring of 1869 the Washburn & Moen Manufacturing Company, of Worcester, Mass., put in a continuous wire rod mill after the design and patents of George Bedson, of Manchester, England.

At first this mill was not a success, the stock used being iron billets, which material would not withstand the strain from pulling between the continuous pairs of rolls. Fortunately, about this time Bessemer steel came into general use, and it was found that with it the continuous system was practicable.

On this mill 1⅛-inch billets were used, but weighing 80 pounds each, and without any manual labor rolled to No. 6 rods. Seven tons of these was considered a satisfactory day's work. Our fellow-member, C. H. Morgan, was in charge of the Washburn & Moen Works, and he soon discovered that the production of the continuous mill was limited by the reel upon which the rods were coiled as they came from the rolls, this reel being operated by hand. He put in a power reel, and was soon enabled to bring the production of the mill to over 20 tons per turn.

A few years later the Roeblings, of Trenton, N. J., following a Belgian practice, built a rod mill composed of two separate trains of rolls, the first, or roughing rolls, being of the largest diameter, and driven direct from the engine shaft, while the second, or rod rolls, were placed some 30 feet back of the roughing, and driven by belt at a much higher speed. This arrangement was much superior to the ordinary mill, but did not dispense with any labor. The Trenton Iron Company, of Trenton, N. J., and Washburn & Moen also put in Belgian trains. The Roeblings and the Trenton Iron Company have both greatly improved their mills. In 1876 the Cambria Iron Company built a rod mill after the designs of Henry B. Corner.

While all the rolls of this train were in a continuous line, they were divided into groups, each succeeding one of which was driven at an increased speed, by a line of shafting placed directly under the train. This mill has since been altered and improved.

Washburn & Moen put in another continuous mill, in which many improvements were made. This was also built under C. H. Morgan's direction, assisted in this, as in the development of the other continuous mill, by F. H. Daniels, also a member of our Society. I believe many of the patents covering these improvements were taken in their joint names.

Since Mr. Morgan severed his connection with Washburn & Moen, the good work has gone on under Mr. Daniels, who succeeded him in the general superintendency.

Rods are now finished on the mill at a speed of 50 feet per second, and reeled with ease and certainty. Mr. Morgan built in 1888 a continuous mill for the American Wire Company, of Cleveland, Ohio, on which over 118 gross tons of No. 5 rods have been rolled in ten hours, through a single groove or pass. They have rolled 500 tons per week of No. 8 rods for three consecutive weeks. On the same mill a production of 35 tons per turn for two weeks of No. 9 rods, 0.148 inch diameter, rolled from billets weighing 210 pounds, has been reached. This was a reduction of area of 99.89%. The finished rods were 3,620 feet long.

In 1882 William Garrett, a member of our Society, then superintendent of the merchant mill department of the Cleveland Rolling Mill Company, Cleveland, Ohio, patented and built a mill which was destined to play a conspicuous part in the American wire industry. The continuous mills were using 1⅛-inch billets, the Belgian mills 2-inch ones. To produce billets of these sizes it was then necessary to roll the steel ingot to blooms, reheat the blooms, and roll to billets. Mr. Garrett's desire was to construct a mill which would take a billet of a large enough section to permit its being rolled direct from the ingot without any reheating. He settled upon 4 inches square as being that size. To accomplish this he went beyond the Belgian mill by putting in three separate trains of rolls, placed in echelon, and driven at progressively increasing speeds. Hence the billet rolls could run at comfortable speed for the workmen to handle the billets without interfering with the speed of the finishing trains. This arrangement not only permitted the use of the larger billet, but made it possible to have several distinct pieces in the rolls at the same time. The present practice is four pieces, and sometimes five rods will be reeled off simultaneously.

Since the Cleveland mill, several other works have put in the Garrett mills, and he has sought to make each last the best. Beaver Falls, Oliver & Roberts, Braddock Wire Company, the Joliet Works of the Illinois Steel Company, American Wire Company, H. P. Nail Company, American Wire Nail Company, and the Newcastle Wire Nail Company have put in the Garrett mills. In all forms of mills, excepting the continuous, advantage has been taken of the device known as the "repeater." This was first patented by John Bevis, of Cleveland, Ohio. A later one was invented by Mr. McCallip of Columbus, Ohio, and the last, and by many thought the best one, was invented by Frank G. Tallman, a member of this Society. By turning the piece from one pass to another the repeater saves a large amount of labor. On one of the Garrett mills 140 gross tons of No. 5 rods have been made in ten hours, 1,300 tons in a week, and nearly 5,500 tons in a month. While the advocates of the continuous system admit that greater product can be obtained on the Garrett, they claim a saving in labor and in loss by oxidation. For the average of three years' work, I am given the loss on weight from billets used to finished product on a continuous mill as 2.08 per cent. If there were no differences of opinion, and professional and commercial rivalries, we would be without progress.

Among the radical manufacturing departures in rolling mills I would mention the Munton Tire Mill, which is in successful operation at the works of the Chicago Tire and Spring Company, at Melrose, near Chicago. The process which this mill makes possible and the mill itself are the invention of James Munton, the superintendent of the works. He entirely dispenses with hammering in making locomotive or other steel tires. This by the following practice : The ingot is cast with a hole cored out large enough to admit a small roll. The ingot is heated and taken to the rolling mill, where its top, with its imperfections, is sheared off by the rolls, and the bloom left of a given weight. At the same heat and by the same operation the bloom is also roughed out by the roughing rolls of the mill and edged down by horizontal rolls. The bloom is reheated and placed in the tire rolling mill, where it is rolled and finished to the exact inside and outside diameter required. Mr. Munton's mill is so constructed that on it a bloom can be "rolled back" to a smaller diameter. This also applies to a finished tire. Another tire mill embracing several new features and of great power is the one designed and built for the Latrobe Steel Company, Latrobe, Pa., by Julian Kennedy, their chief engineer. The other American steel-tire makers are the Midvale Steel Company, of Philadelphia, Pa, and the Standard Company, already mentioned.

During the years of the rolling mill history which I have sought to sketch there have been many improvements made in the construction of all forms of mills, and many sections have been successfully produced on them which were previously thought impossible. I have not attempted to enumerate all of these, but have sought to confine myself to those which have been distinct departures in the roll trains. I have done so because, while many things and men deserve mention, the time required would have gone far beyond the possibilities of this occasion.

I will, therefore, close with a reference to the latest radical departure, which, while not yet quite a commercial success, still has been placed in a sufficiently prominent position to deserve record. I refer to the rolling of liquid steel by Edwin Norton, at the works of Norton Brothers, in Chicago. It has been known for some time that Mr. Norton was experimenting in this direction, and, in fact, had taken out patents in this and other countries. The paper read by Sir Henry Bessemer at the late meeting of the Iron and Steel Institute, on "Rolling Fluid Metal," and which has already been published in several of our technical papers, has called renewed attention to this system of producing steel sheets for tinning.

The commercial changes in the tin-plate industry have been somewhat violent of late, and appearances indicate that the location of a large part of the manufacture may come to this country. At all events, changes which will cheapen the production have become necessary, no matter where the plates are made. The Messrs. Norton are the largest individual tin-plate consumers in America, and as such have naturally kept fully in touch with all phases of the trade. Edwin Norton is the mechanical member of the firm. Some years ago he fully perfected the rolling of soft metals direct from the liquid state into finished sizes. He was fully posted as to what had been attempted in the years past in such rolling. His success with soft metal led him to go further, and strive to do the same thing with liquid steel. He had encouraging results. He protected himself by patents at home and abroad. He felt full reverence for all that Bessemer had given the world, and knew that some thirty odd years ago he had experimented with this very process, and thought he knew wherein Bessemer had not gone quite far enough to insure complete and uniform success. Still, he felt that he deserved recognition. Moreover, Mr. Norton, as a successful American, duly appreciated the advantages which would naturally accrue from having Bessemer indorse and become interested in his process. For these reasons, after protecting himself by patents, he wrote Sir Henry Bessemer; also sending him a piece of sheet which he had rolled direct from liquid metal; explained what improvements he had made, and proposed an honorable commercial relationship. Certainly, there was nothing disrespectful in this, no matter how great the man to whom it was made.

Now, mind you, up to this time Bessemer seems to have attached little importance to his experiments of many years ago. Messrs. McKinley and Norton recalled them to his mind and gave them value. Perhaps this is none of my business, but Sir Henry has aroused my American ire. He says:

"I received, about two years ago, a parcel from America containing a small sample of sheet metal which was being successfully manufactured there. The *person* from whom I received it informed me that it was made by a slight alteration or improvement on my patent of 1857, for rolling continuous sheets and thin bars of iron or steel direct from fluid metal. He offered me one-half of his patent if I would undertake its introduction into this country. I did not accept his offer and there the matter rested. The circumstance stated has, however, afforded me an opportunity of showing you a small sample of a continuous sheet, produced direct from fluid metal at a single operation, and proves, beyond doubt or question, the important fact that fluid metal may be chilled and formed into a continuous sheet between rolls that are kept cold, while it well illustrates the spirit of enterprise of our American cousins, who are so prompt to recognize, to adopt, and to improve upon the inventions brought forward in Europe."

I think he might have treated Mr. Norton a little less cavalierly, without any prejudice to his great fame. This thing had lain dead all these years. Should not some honor be given the man who was able to put life into the corpse, and conquer for it a place among live industries? The greatest have always been under obligations to others for portions of their triumphs, and always will be. The Bessemer process was a failure for all but the higher purposes until Mushet's invention, and while Sir Henry honored himself by settling an annuity upon Mushet, who at that time had made nothing from his invention, still I, for one, would have had my already profound respect increased, on tests of duty, the actual performance reported has been placed on a basis which cannot be challenged, since it was strictly commercial; and when, up to 1800, the Cornish miners had paid Watt and Boulton about $900,000 for coal saving, we may rest assured that the account was not overpaid or left in any doubt.

Situated on the southwesterly promontory of England, the coal of Wales was more convenient than that of Newcastle or Derbyshire, and more valuable under test, ranking in some trials as 9.27, 8.52, and 6.77 lbs. evaporation respectively for ordinary use. Its cost in some estimates was 17 shillings per ton at the mines, free of duty; and at London, in 1855, it was 26 shillings.

The use of spherical boilers in ancient times, for the temples and for power, is on record, and these were the primitive plans of the eighteenth century. If Papin, before 1700, used steam of 612°, or about 1,400 lbs. per square inch,* his knowledge of safe construction and management must have been great. He used, in one case, a balloon-shaped boiler, 8 by 5 ft., with a flue 10 inches square, generating spheroidal steam, and got an evaporation of 8 lbs. per pound coal. He also introduced the balanced safety valve.

Savery, in 1702, used similar spheres, set in brickwork, over the fires.

Newcomen, 1705–20, used convex top spherical boilers, with flat bases, over the fires, the cylinder being over the boiler, for beam engines with single acting lifting pumps.

Blakey, 1766, used a high-pressure boiler, with small water-tubes in zig-zag bends over the fire; but an explosion due to want of strength discouraged the use of much pressure.†

Dr. Hale, 1740, investigated and indorsed the economy of air charges to a boiler, since demonstrated in various cases on the theory to which Ericsson devoted so many years.‡

Allen, 1729, proposed a firebox boiler, with spiral flue, with a furnace air blast, used by Ericsson's "Novelty" in 1829.

Brindley, 1759, proposed to check radiation with stone base and wood chamber boilers, with cast-iron furnaces and flues, and copper smoke tubes. Similar boilers were used at the Phila-

* Sewell. *Steam*, p. 245.
† Sewell, *Steam*, etc., pp. 248–261.
‡ Pole, *Cornish Engine*, p. 7.

CCCCLXVII.

*EXPERIMENTS TO DETERMINE THE RATE OF FALL (OR RISE) OF A MERCURIAL THERMOMETER UNDER DIFFERENT CONDITIONS.**

BY A. F. NAGLE, CHICAGO, ILL.
(Member of the Society.)

AT the Cincinnati meeting of this Society I presented a paper on the subject of the sensitiveness of automatic sprinklers,† and this paper is an elaboration of the experiments therein recorded.

The method of determining the tardiness of mercurial thermometers in responding to rapid changes of temperatures, as announced in that paper, is undoubtedly correct, but the *constant* for correction then found needs further explanation.

It will be remembered that the rate of *fall* of a thermometer from a high to a lower temperature was assumed to hold equally good for its *rise* from a low to a higher temperature. This is not strictly correct in theory; because when a heated thermometer is exposed to a cooler atmosphere, it loses heat both by radiation and convection, while in the inverse process, when a cold thermometer is heated by hot gases it receives heat only by convection and not by radiation. It was therefore determined to investigate farther the effect of *velocity of current* in cooling the thermometer. When the first experiments were made I knew that it made a difference whether the thermometer was held still in the air or the air kept in circulation by fanning; but I did not suspect that the varying velocities of the cooling air would affect the fall of temperature so greatly as I have now found them to do.

Tests have now been made under three conditions, not sufficient to establish a law covering any particular velocity of current, but sufficient to show conclusively that the velocity of current has altogether too great an influence to be neglected.

In the first case the thermometer was heated to 345° Fahr.,

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* Presented at the New York meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

† Trans. A. S. M. E., Vol. XI., p. 699, No. 383.

and allowed to cool in the centre of a room at 74° Fahr. without any artificial circulation of air. In this case radiation and convection together reduced the temperature at a rate which gave a constant of 10.5.

The method of obtaining and applying this constant is explained in the former paper. The mean difference between the temperature of the thermometer and the room at intervals of ten seconds, is divided by its fall during the same time.

In the second test the air was fanned quite violently and gave a constant of 5.90, while with the same thermometer, designated "B" in the former paper, under like conditions, except such dif-

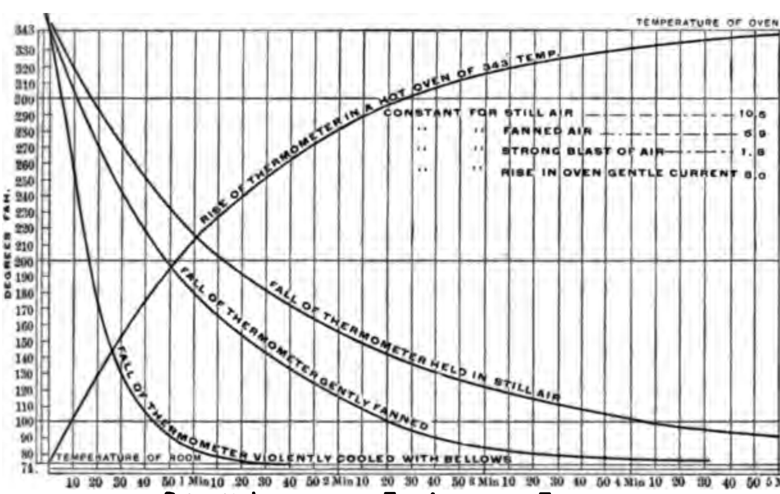


DIAGRAM ILLUSTRATING THE AMOUNT OF TARDINESS OF A MERCURIAL THERMOMETER UNDER DIFFERENT CONDITIONS.

FIG. 1.

ference as would naturally result from such imperfect methods of cooling, the constant was found to be 5.44.

In the third test the thermometer bulb was held in front of a $\frac{3}{4}$ inch tube, from which issued a violent blast of air, 74° Fahr., generated by a strong foot bellows. This gave a constant of 1.80. See table and diagram (Fig. 1).

In each of these cases the radiation must have been the same, and we learn that by far the greater effect was produced by the *velocity with which the cold air impinged upon the heated thermometer.*

In order, however, to eliminate the effect of radiation, but with no facilities for producing currents of any greater velocity than such as was generated by the hot gas itself, the following test was made: The same oven was used as is illustrated in Fig. 110, Vol. XI., page 700, and heated to 343°, and held at that temperature by lifting the cover slightly. Nearly the full force of gas was turned on. This condition is approximately that prevailing during a sprinkler test. The thermometer of 74° was inserted through a small hole in the top, and its rise noted. The result is recorded in the table and illustrated in the diagram. A constant of 8 was found. As this *happens* to be about the same constant (7.8) as was obtained with another thermometer by the fanning process, illustrated in the former paper, Fig. 111, Vol. XI., page 700, holds good for illustrating the difference between apparent and actual temperatures.

All tests were repeated several times to insure accuracy of results.

We learn from these simple experiments that automatic sprinklers, or other bodies to be heated for scientific tests of this character, are so greatly influenced by the *velocity* of the current of either hot or cold air, that unless full cognizance be taken of this fact the results will be of but little scientific value.

There are two methods of preparing ovens for sprinkler tests —large ovens having a fixed temperature, and small ovens with a rising temperature.

In the former case it is exceedingly difficult to maintain a fixed temperature during the operation of tests without *adding* (or losing) heat, and this must cause currents of more or less magnitude *in different parts of the oven.*

In the small oven used, 8″ × 8″, it was found that there was as much as 10° difference in two positions of the thermometer within 6 inches of each other, separated only by a thin and small shield 1½″ × 2″, used to protect the thermometer from flying pieces when a sprinkler opens, and in this oven test all pipes were removed except this little shield. Of course the higher temperature (10° in 343°) was found nearest to the larger vent aperture when the slight current must have been the stronger.

I am inclined to favor a small oven, where *all* the hot gases will pass around the sprinkler, and an adjustable vent above it admits of producing any desired final temperature.

Explanation of table:

Columns 1, 5, 9, and 13 are the initial temperatures of the thermometers every 10 seconds.

Time. Minutes and seconds.	Thermometer stationary. Temperature of room 74°.				Thermometer fanned. Temperature of room 74°.				Thermometer violently blown. Temperature of room 74°.				Therm. inserted in oven at 343° temp.			
	Thermometer.	Rate of decrease.	Difference between therm. and room.	Constant.	Thermometer.	Rate of decrease.	Difference between therm. and room.	Constant.	Thermometer.	Rate of decrease.	Difference between therm. and room.	Constant.	Thermometer.	Rate of increase.	Difference between therm. and oven.	Constants.
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
0	384	......	274	....	346	....	272	....	350	....	276	......	74	....	269	...
10	316	32	242	8.	306	40	232	6.3	250	100	176	2.3	103	29	240	8.8
20	293	23	219	10.	268	38	194	5.6	170	80	96	1.7	130	27	213	8.4
30	271	22	197	9.5	240	28	166	6.4	128	42	54	1.8	154	24	189	8.4
40	251	20	177	9.9	218	22	144	7.0	106	22	32	2.0	176	22	167	8.1
50	232	19	158	8.4	196	22	122	6.0	92	14	18	1.8	196	20	147	7.9
1–0	217	15	143	10.0	180	16	106	7.1	84	8	10	1.7	213	17	130	8.1
10	204	13	130	10.5	166	14	92	7.1	79	5	5	1.5	228	15	115	8.2
20	192	12	118	10.3	155	11	81	7.8	76	3	2	1.2	241	13	102	8.3
30	182	10	108	11.3	144	11	70	6.9	75	1	1	2.5	253	12	90	8.0
40	173	9	99	11.5	134	10	60	6.5	74	1	0		264	11	79	7.7
50	164	9	90	10.5	125	9	51	6.2	......	......	......	......	274	10	69	7.4
2–0	156	8	82	10.7	116	9	42	5.2	......	......	......	......	283	9	60	7.2
10	149	7	75	11.2	108	8	34	4.7	......	......	......	......	291	8	52	7.0
20	142	7	68	10.2	102	6	28	5.2	......	......	......	......	298	7	45	7.0
30	136	6	62	10.8	96	6	22	4.2	......	......	......	......	303	5	40	8.5
40	131	5	57	11.9	92	4	18	5.0	......	......	......	......	308	5	35	7.5
50	126	5	52	10.9	89	3	15	5.5	......	......	......	......	312	4	31	8.2
3–0	122	4	48	12.5	86	3	12	4.5	......	......	......	......	316	4	27	7.3
10	118	4	44	11.5	84	2	10	5.5	......	......	......	......	319	3	24	8.5
20	114	4	40	10 5	82	2	8	4.5	......	......	......	......	322	3	21	7.6
	Average constant....			10.5	Average constant....			5.9	Average constant....			1.8	Average constant....			8.0

Columns 2, 6, 10, and 14 give the amount of decrease (or increase) every 10 seconds.

Columns 3, 7, 11, give the difference between the temperature of the thermometer and the room every 10 seconds, and column 15 the difference between the temperature of the oven and the thermometer.

Columns 4, 8, 12, and 16 give the arithmetical mean of two preceding intervals divided by the rate of decrease (or increase) given in columns 2, 6, 10, and 14.

This latter gives the constant for a given thermometer under the conditions prevailing during the experiment, which is used for correcting the apparent temperature. To illustrate its application : Assume the temperature to be rising at the rate of 12° in 10 seconds, and the indicated temperature at the end of that interval to be 217°, there must be added 12 x constant, which in this case assumes to be 8, giving 96° to be added, or a total of 313° as the actual temperature at that instant.

DISCUSSION.

Mr. D. L. Barnes.—The results of Mr. Nagle's investigations regarding the accuracy of the readings of thermometers in atmospheric air which is rising in temperature, or when the thermometer itself is at a considerably higher temperature than the air, are highly interesting. He shows that the velocity of air in passing the thermometer materially affects the rapidity of fall or rise of the mercury column. This might be expected, as the thermometer is cooled by contact with the molecules of air, and the greater the velocity at which the air moves the greater the number of molecules which will come in contact with the thermometer in a given time. It would be interesting to know whether a thermometer inserted in a rapidly moving current of air would, after reaching a normal reading, register a temperature higher than the true temperature of the air.

One application of these results is found in steam calorimeters when used on locomotives. Under the conditions which there exist, the amount of moisture in the steam varies considerably within a few seconds. Within one minute from the time of starting a locomotive the rate of steam used per minute from the boiler will vary several hundred per cent., and consequently the amount of moisture entrained with the steam varies considerably. To determine the variation in wetness during this short interval, it is necessary to have a quick-acting thermometer with the bulb placed in the wire-drawn steam in the calorimeter. Evidently, the results in Mr. Nagle's paper show that the readings of a thermometer under these conditions would need to be corrected considerably before they would be true.

Another application of the results obtained by Mr. Nagle is found in steam heating. A steam radiator in a room heats the air in two ways : namely, by convection and by radiation. With indirect systems of heating, the air is heated almost solely by coming in contact with the radiator as it passes through the heating chambers. According to Mr. Nagle's results, one might expect that the heat extracted from a radiator in a given time would largely depend upon the velocity of the current, and perhaps the law of the falling temperature in a thermometer, as determined by Mr. Nagle, might apply to the extraction of heat from a steam radiator placed in a current of air; that is, the amount of heat extracted from a radiator might be inversely proportional to the constants which Mr. Nagle has determined. Perhaps there are other places where there are rapid fluctuations in temperature which would be difficult to record unless some correction was made for the inertia of the thermometer. What Mr. Nagle has done is to show us how to correct for this inertia. We may possibly learn from these results that in a room having too little steam heating surface under normal conditions, the use of a small electric fan to put the air in the room in motion would enable that small surface to heat the room more satisfactorily.

Mr. Arthur C. Walworth.—It occurs to me that the experiment is very analogous to the experiments that have been tried with steam radiating surfaces in which the great difference in the cooling of such surfaces by a blast of air has been noticed over that of same surfaces placed in air that is motionless. For instance, the heat radiation of radiators is generally and accurately measured by the condensation of steam therein. Twenty-five or thirty years ago the late Joseph Nason demonstrated that an iron radiator made of one-inch pipe condensed from .23 to .41 of a pound of water per hour (the latter being the condensation in a single line of one-inch pipe), in the atmosphere in its natural state. At that time air was not forced through radiators to any great extent. Afterwards, when we came to blow air through radiators with fans, steam heating engineers were somewhat surprised to see the great increase in condensation. For instance, we find that with a velocity of thirty or forty feet per minute of air driven through a steam radiator, we obtain a condensation equal to the evaporation of the same amount of boiler surface; that is to say, a square foot of radiating surface will condense two pounds of water per hour, which is a fair amount of evaporation per square foot in the ordinary horizontal tubular boiler; these two pounds of water being from six to eight times the amount of condensation found in a radiator not exposed to an artificial current and corresponding to the number of heat units lost by the same surface in each case. If the radiator tubes are placed horizontally, the natural velocity of the air passing through it will be about two or three feet a second, while in the other case the velocity of the artificial blast would vary from twenty-five to forty feet per second, and there is a corresponding—not a proportionate but a natural—difference in the condensation under these two circumstances, just as we see in the curve which is shown in this paper of the thermometer's variation. I thought that the comparison was interesting enough to take a minute or two to state.

CCCCLXVIII.*

LIMITATIONS OF STEAM-ENGINE ECONOMY.

BY A. F. NAGLE, CHICAGO, ILL.
(Member of the Society.)

THE writer has recently worked out a table, and plotted the same, which gives the amount of water per horse-power per hour with different degrees of expansion and under different steam pressure for both condensing and non-condensing engines, and it is submitted for the study of such as have not already the information. It is simply the amount of steam an engine would consume if there were no clearance or waste space, nor any condensation within the cylinder.

The formula by which the results are obtained is very simple, namely:

$$W = \frac{1{,}980{,}000}{V \times 144 \times R \left[\frac{1 + \text{hyp. log. } R}{R} \times P - p\right]}$$

W = Pounds of water per horse-power per hour.

V = Cubic feet per pound of steam of pressure P.

R = Number of expansions.

P = Initial steam pressure, absolute.

p = Back-pressure, taken at 16 lbs. (absolute) for non-condensing and 2 lbs. for condensing engines.

1,980,000 = Foot pounds of work per horse-power per hour.

144 = Square inches per square foot.

To illustrate the formula assume a steam pressure of 150 lbs. and an expansion of 15, condensing engine.

$V = 2.996$.

Hyp. log. 15 = 2.7081.

$$W = \frac{1{,}980{,}000}{2.996 \times 144 \times 15 \left[\frac{1 + 2.7081}{15} \times 150 - 2\right]} = 8.83.$$

* Presented at the New York meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

There is added on the diagram (Fig. 2) the water used by several representative types of engines, and it enables one to study at a glance the lines within which we must make improvements to secure greater economy.

If a triple expansion engine consumes 12½ lbs. of water, and the theoretical quantity with the same steam pressure (150 lbs.) and same degree of expansion (20) be 8.30 lbs., then the engine has an efficiency of

$$\frac{8.30}{12.50} = 66\%$$

A compound engine consuming 14 lbs. of water with 105 lbs. of steam pressure and 15 expansions would require, theoretically, 9.25 lbs. of water, an efficiency of

$$\frac{9\ 25}{14} = 65\%$$

If we could retain the same efficiency with much greater steam pressure and still greater number of expansions, how much could be saved?

If the steam pressure were increased to 250 lbs. and the expansions to 30, a theoretical consumption of 7.22 lbs. would be obtained; or, with an efficiency of 66%, $\frac{7.22}{.66} = 10.90$ lbs. would be used, a saving of only 1.60 lbs. of water on 12.50, or 12%. Is that worth saving at the cost necessary to obtain it?

Let us turn for a moment to the non-condensing engine. An actual consumption of 23.60 lbs. of water with 105 lbs. of steam pressure and 5½ expansions is now obtained. Theoretically, the consumption would be 17.10, or an efficiency of $\frac{17.10}{23.60} = 72\%$, apparently somewhat greater efficiency than the condensing engine gives.

If the steam pressure were increased to 250 lbs. and the expansions to 7, a theoretical consumption of 12.12 lbs. would be obtained. Assuming an efficiency of 72%, the actual quantity required would be $\frac{12.12}{.72} = 16.83$ lbs., a saving of 6.77 lbs., or nearly 30%.

Seven expansions could be obtained with a single cylinder. If two cylinders were used, making possible 12 or 15 expansions, and with 250 lbs. steam pressure, we could obtain the

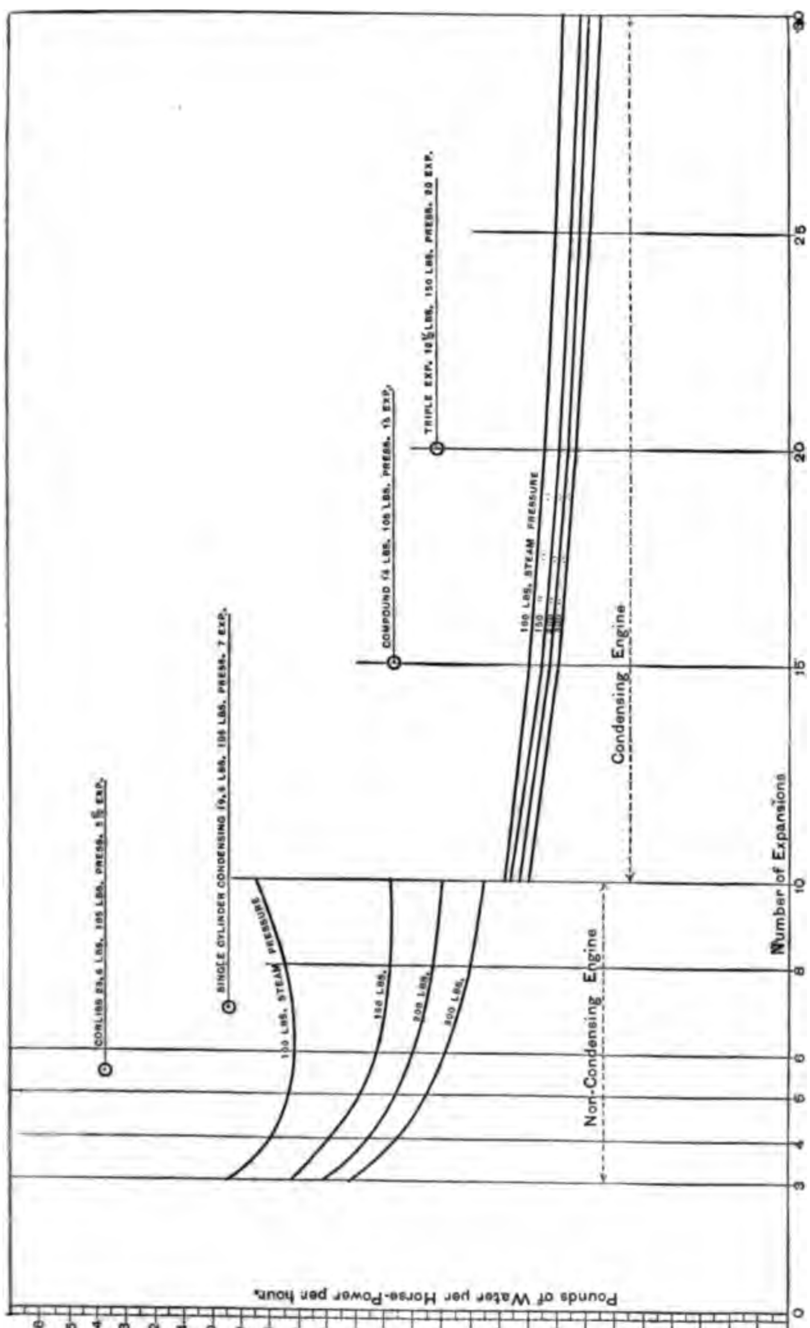

Pounds of Water per Horse-Power per hour.
Number of Expansions
Non-Condensing Engine
Condensing Engine

FIG. 2.

following result: 250 lbs. steam pressure, 12 expansions, and assuming an efficiency of 75%.

W = 14.78 lbs of water per horse-power per hour.

With 300 lbs. steam pressure, 15 expansions, and an efficiency of 75%.

W = 13.63 lbs. of water per horse-power per hour.

These figures are simply indications of what is not an unreasonable possibility of steam-engine economy.

DISCUSSION.

Prof. F. R. Hutton.—In the discussion in the Publication Committee of the Society upon the acceptance of the foregoing paper it was brought out that, while it shows, through a considerable range of pressures, the value of one of the elements which enter into the cost of power in steam-engines, it was manifestly incomplete as an exhaustive discussion of that subject. Mr. Chas. E. Emery, member of the Society, presented at the Scranton meeting a paper on the "Cost of Power in Non-Condensing Steam-Engines," published in Volume X., at page 233, in which the calculated cost of power was worked out, not only to include the quantity of steam required to fill the cylinder to the point of cut-off, as in the paper herewith, but also the quantity of steam necessarily condensed to furnish the heat transmuted into mechanical work.

We would be glad if Mr. Emery would supplement the paper by stating how great addition should be made to take account of that which is neglected in the author's paper.

Mr. Chas. E. Emery.—The quantity of steam necessarily condensed to furnish the heat transmuted into mechanical work amounts by itself to 2½ lbs. and upward per horse-power per hour, as shown in column 5 of the paper referred to in Volume X. In addition to this, 1 to 1½ lbs. of water per horse-power per hour are required under practical conditions when the clearances are considered, so that nearly 4 lbs. should be added to each of the costs stated in Mr. Nagle's table to show the actual calculated cost arising from necessary conditions. The actual costs found in practice in excess of the quantities given in Mr. Nagle's tables, plus about 4 lbs. of water in each case, may be considered as due to causes more or less preventable, such as cylinder condensation, imperfect distribution of steam, and the like.

In this connection it may be noted that curve H of my first table shows for 80 lbs. steam pressure the calculated cost without clearance when both the steam required to fill the cylinder and that condensed for work are included; that curve G shows the relative cost when a clearance of $\frac{1}{10}$ of the displacement is considered; and that curves D and F show costs obtained in actual practice for engines of different sizes. The positions of the latter in relation to the former show, for the pressure stated, the increase of cost due to causes more or less preventable above referred to. But the author deserves credit for having taken a great deal of pains to work out that single element of cost of which the paper aims to treat.

Prof. C. H. Peabody.—It appears that this paper was considered by the publication committee to be open to the criticism that it is a partial discussion of a problem which had already been presented to the Society in a more complete form. A little investigation reveals the graver criticism that the methods employed in the paper are empirical, and that the results are in error. It would take more space than appears proper at this time, to give a full and correct presentation of this subject, with explanations and comparison with engine tests. I shall, therefore, merely call attention to the places where the paper deviates from the ordinary methods of thermo-dynamics, and give a correct solution for one of the examples. Standard works on thermo-dynamics, such as those by Rankine, Clausius, and Zeuner, supply the material for such investigations and calculations, though perhaps not in the form most convenient for the present purpose.

It ought not to be necessary at this day to point out the fact that the expansion curve for steam in a non-conducting cylinder is not even approximately a hyperbola. There appears to be no reason, theoretical or practical, for using the hyperbola in connection with the indicator diagram of a steam engine, unless it be in roughly blocking out the design for a new engine.

The accepted definition of the efficiency of a heat engine of any sort, is the ratio of the heat changed into work to the heat supplied. Thus the actual efficiency found from a test made on the triple expansion engine in the laboratories of the Massachusetts Institute of Technology and reported to this Society,* is

* *Transactions* American Society Mechanical Engineers. Vol. XII., page 747.

0.183 ; the efficiency for a nonconducting engine with the same boiler pressure, terminal pressure and back pressure, is 0.225. The steam consumptions are respectively 13.7 and 11.3 lbs. per horse-power per hour. The ratio of the efficiencies is 0.813 ; the inverse ratio of the steam consumptions is 0.82. It is therefore evident that the steam consumption does not form a basis for the comparison of the efficiencies.

Mr. Nagle finds by his equation that an engine supplied with steam at 150 lbs. pressure absolute, expanding to 10 lbs. absolute, and exhausting against a back pressure of 2 lbs. absolute, will require 8.83 lbs. of steam per horse-power per hour. A solution of this example by the methods of thermo-dynamics gives for the efficiency 0.219, and for the steam consumption per horse-power per hour 10.6 lbs. The discrepancy cannot be considered unimportant, even though the result is merely to be the basis of a guess at the probable steam consumption of our actual engine.

Prof. Lanza.—I supposed, Mr. President, that the time had gone by when it was assumed that a hyperbolic curve gave correctly the operation of the steam in a cylinder with non-conducting walls.

*Mr. A. F. Nagle.**—For the purpose of the paper, namely, to represent at one glance the relation which steam pressures and degrees of expansion bear to steam economy, I thought that the simple hyperbolic curve would be sufficiently accurate, and I still think so. I chose it also because the equation would then be understood by a much larger number of our readers than if the thermo-dynamic equation had been used.

The graver error I consider to be the one pointed out by Mr. Emery, namely, that no account was taken of the heat transmuted into work, which amounts to about 2½ lbs. of water per hour. This amount should be added to that given in the tables aad should be embraced in the plotted curves. in order to represent the better what I aimed to do.

* Author's closure.

CCCCLXIX.*

THE BROOKLYN PUMPING ENGINES OF 1860.

BY SAMUEL M'ELROY, NEW YORK CITY.

(Member of the Society.)

THE following paper is a contribution to the history of pumping engines in the United States and in England up to 1860:

In 1856 the Nassau Water Company executed a contract with Henry S. Welles & Co., for the Brooklyn Water Works, to build the whole work for $4,200,000, which was subsequently assumed by the city for construction. It became my duty, as chief engineer of the company, to prepare the contract, specifications, and plans; and, as all the supply from the Long Island streams had to be pumped from near tide level, into a reservoir about 170 feet above tide, for the main supply of 20,000,000 gallons per day, with an auxiliary reservoir and supply, 26 feet higher, of 2,500,000 gallons, the pumping plant was an important feature, unusual in both quantity and elevation here.

The state of the art of water-works pumping in this country up to this date was confused and unsatisfactory. Philadelphia, in 1822, had abandoned, successively, the engines of 1801, 1812, and 1815, for the more economical water-power of Fairmount Dam, to resume steam-power for Spring Garden section in 1844, and to use it for subsequent city additions.

The engines of 1801 were beam, double-acting; the *Centre Square*, 6-feet stroke by 36 inches, with 18-inch pumps; with wood boilers, beams, shafts, and pumps, lifting about 1,000,000 gallons per day. The *Schuylkill*, of same stroke, 40-inch cylinder and 17½-inch pumps, lifting about 1,500,000 gallons per day about 50 feet.

In 1815 a Boulton and Watt engine of 6 feet by 40 inches, 20-inch pumps. was started with about 102 feet lift; capacity,

*Presented at the New York meeting (1891) of the American Society of Mechanical Engineers and forming part of Volume XIII. of the *Transactions*.

1,750,000 gallons, with 2½ to 4 lbs. steam. A second, Oliver Evans engine, was built of 5-feet stroke, 20-inch cylinder, pump 4 feet by 20 inches; capacity, 3,000,000 gallons, with 194 lbs. steam, using 13 cords of wood, or about 60,000 ft. lbs. per pound wood (115,000 per lb. coal). For such an engine the use of boilers under 220 lbs. steam showed an advanced study of economy, fully demonstrated afterward by the Western steamboat practice.

The Spring Garden engines of 1845 were two-beam, crank and fly-wheel, double acting, 6 feet by 36 inches, pumps 18 inches, with a half-beam engine; in 1849, cylinder vertical, pump horizontal, 6 feet by 36 inches; pump, 4 feet by 21 inches. In 1854 a Cornish engine was built, 10 feet by 60 inches, plunger 30 inches. The duty report for 1856, pump friction included, gives for Nos. 1, 2, and 3, 317,800 ft. lbs. (per pound coal) for year; 364,400 for July; No. 4 (Cornish), 381,000 for year; 512,000 in November; a special test of June, 548,190.

At Buffalo a "Bull" Cornish engine had been in use several years; duty about 370,000 ft. lbs. At Belleville, Jersey City had a Cornish engine; Hartford had a double-acting cam engine; Cambridge, a compound Worthington. The dry dock crank engine of the Brooklyn Navy Yard was for several years under my charge, with 12 feet by 50-inch cylinder and two lifting pumps of 8 feet by 63 inches; duty low, or about 225,000 ft. lbs.

Various patterns of horizontal double-acting engines were in use: at Pittsburgh, of about 171,500 ft. lbs. duty; Detroit, 341,000; Cincinnati, 350,000 to 400,000.

On the other hand, the history of the mining operations of England had become that of the single-acting engine in its full development of the principles of steam and water motion, in the front rank as to size, power, and economy. As a study of interest, we may glance at the conclusions thus taught at the time in the distinct divisions of boilers, engines, and pumps.

Boilers.—It was fortunate for the earlier development of pumping machinery under steam that the cost of coal in Cornwall was a burden strongly felt, and until Parliament reluctantly withdrew the duty on importation, in 1741, this kept back engine use. Forming, as this did, a strong incentive to invention, to be rewarded in money for each of its gains, and a keen safeguard on tests of duty, the actual performance reported has been placed on a basis which cannot be challenged, since it was strictly commercial; and when, up to 1800, the Cornish miners had paid Watt and Boulton about $900,000 for coal saving, we may rest assured that the account was not overpaid or left in any doubt.

Situated on the southwesterly promontory of England, the coal of Wales was more convenient than that of Newcastle or Derbyshire, and more valuable under test, ranking in some trials as 9.27, 8.52, and 6.77 lbs. evaporation respectively for ordinary use. Its cost in some estimates was 17 shillings per ton at the mines, free of duty; and at London, in 1855, it was 26 shillings.

The use of spherical boilers in ancient times, for the temples and for power, is on record, and these were the primitive plans of the eighteenth century. If Papin, before 1700, used steam of 612°, or about 1,400 lbs. per square inch,* his knowledge of safe construction and management must have been great. He used, in one case, a balloon-shaped boiler, 8 by 5 ft., with a flue 10 inches square, generating spheroidal steam, and got an evaporation of 8 lbs. per pound coal. He also introduced the balanced safety valve.

Savery, in 1702, used similar spheres, set in brickwork, over the fires.

Newcomen, 1705–20, used convex top spherical boilers, with flat bases, over the fires, the cylinder being over the boiler, for beam engines with single acting lifting pumps.

Blakey, 1766, used a high-pressure boiler, with small water-tubes in zig-zag bends over the fire; but an explosion due to want of strength discouraged the use of much pressure.†

Dr. Hale, 1740, investigated and indorsed the economy of air charges to a boiler, since demonstrated in various cases on the theory to which Ericsson devoted so many years.‡

Allen, 1729, proposed a firebox boiler, with spiral flue, with a furnace air blast, used by Ericsson's "Novelty" in 1829.

Brindley, 1759, proposed to check radiation with stone base and wood chamber boilers, with cast-iron furnaces and flues, and copper smoke tubes. Similar boilers were used at the Phila-

* Sewell, *Steam*, p. 245.

† Sewell, *Steam*, etc., pp. 248–261.

‡ Pole, *Cornish Engine*, p. 7.

delphia Water Works in 1812, and by Fulton for only one trip on the *Clermont*, in 1807.

Smeaton, 1772, used the Newcomen form, with the cylinder over the boiler, and in some cases the firing improvement of coal feeding tubes. His evaporation reached 7.88 lbs., to 8 lbs. of steam.

Watt, 1762–1800, introduced the wagon top boiler, with concave base over the fire, with side returns, and automatic water feed and damper regulators. His pressure was usually limited to "three inches," in the earlier engines.

The boilers modified by Fulton, and used in the United States steamer *Fulton*, in 1837, were well built of copper, with a large central flat D or "Kidney" flue, and side returns.

Woolf, 1796–1804, used a wagon top boiler, with eight or more transverse water tubes or cylinders in the furnace flue, arranged like a fish weir, for flame contact, and worked under higher pressure; and as it appears from his testimony in 1817, his boilers were of heavy castings 2½ to 3 inches thick.

Trevithick, 1790-1816, used a high-pressure spherical boiler, set in brickwork; and afterward a cast-iron central flue boiler, the furnace within the shell, leading directly into the chimney, as adopted for his locomotives. The Watt and Evans' systems added the return flues.*

The Cornish double return drop flue boiler, which came into common use in England, and was much preferred to the wagon top, has been usually attributed to Trevithick; but Sewell † makes it clear that Evans was the first to use the drop return under the boiler. If so, the favorite Cornish boiler is essentially American in principle.

Oliver Evans, 1784–1804, an American engineer, knowing the economy of high steam, designed a long cylinder, with an internal cylindrical *furnace* flue, and drop return flue; a hot feed coil was also used. In the Philadelphia case these were used under 220 lbs. (gauge), and in western practice, 180 to 200 lbs. was the usual pressure for large flue cylinder boilers. He also introduced feed heating by exhaust steam, an important step in economy.

Stevens, another American, in 1804 introduced the water tube boiler, with a large number of short copper tubes, 81 of 24 by 1 inch, horizontal, in the first small boiler.

* Sewell, pp. 257–291.

† *Ibid.*, p. 282.

As will be observed later, the cylindrical, double return drop flue boilers we adopted were at that time the favorites of the Naval Engineer Corps, and combine with some advantages the essential principles of the Evans' boiler.

In this evolution of the boiler of 1860 from the spheres of the ancient temples and other uses, there is a singular zig-zag progress. The virtues of high steam, proved by Papin, were long ignored by Watt and his contemporaries, and forced on him by Hornblower, Woolf, and Trevithick, in rival patents. The air blast, not needed for slow combustion, of Allen, was appropriated by Ericsson in his locomotive; and there was a short use of a self-feeding grate by Smeaton.

Watt's wagon top, with its side flues, was the first to give adequate fire surface for important work; it ranked well in evaporation, and its later patterns in strength.

The principles secured by the double return drop flue boiler of the Cornish and American marine type place it in the front rank of attainment. Its furnace within the shell has a direct effect on evaporation, air supply, expansion and contraction, tightness and durability;* by its use of the furnace flues as combustion chambers of an adequate size, and by making the greatest quantity of steam from the thinnest layer of water, important advantages are gained in steam delivery, as in water circulation; by its descending returns of the heated gases, constantly tending to rise in the tubes, it delivers their heat to the water in its upward circulation, without injury to the spaces of greatest efficiency, so that these gases can pass to the chimney, at comparatively low temperature, without injury to water evaporation in the lower returns. They are easily built, braced, transported, set, inspected, cleaned, and clothed; their strength is only a question of material and workmanship, and their durability unequalled; built with a dome or drum, superheating is easily secured, while their bottom feed and blow-off, dampers and regulators may have special advantages of position.

In evaporation, no higher standard could be expected than the better records made.

Taking the theoretical value of 1 lb. coal from 212°, at 14.277 lbs. water, the Par Consol boilers, in a test of 11,730 lbs., evaporated 11.428 lbs. water, or 82.09% (or 10.204 lbs. from 92° feed), and feed heat included, 83.308%; and going into the chimney

* Three of the Brooklyn boilers of 1861 were in use in 1885.

Objections.—Friction of many engines and pumps; low duty of Pittsburg type (178,050 ft. lbs.).

H. R. WORTHINGTON.—Three pairs horizontal, direct acting, compound engines; cylinders annular; hollow plunger pumps, with diaphragms; small rubber water valves. Capacity, 30,000,000. Advantages: three-fold expansion, direct motion, uniform steam and water load; full control of steam valves.

Objections.—Cambridge type shows want of mass in motion to promote expansion and prevent pump concussion; increase of stroke, 26 to 60 inches, head, 72 to 165 feet, speed, 50%, will aggravate this defect; six pumps on a chamber common to two mains will react on each other; Cambridge duty with 11 lbs. evaporation, 551,260 ft. lbs.; actual pump delivery not measured; several cylinders cannot compete with two large ones for same work; annular cylinders leak.

I. P. MORRIS & Co.—Four vertical "Bull" engines, plunger pumps, under cylinders, counterweighted balance beam. Capacity, 30,000,000. Advantages: simple motion and high expansion.

Objections.—Form of engine adopted to evade the Watt patents, with low duty record in the mines and here; time element in beam vibration important to initial cylinder and pump motion shown by long experience; two cylinders and pumps make more friction than one, and cost more for attendance, repairs, and otherwise; reverse of water motion at each stroke objectionable.

DOUBLE ACTING BEAM.

THOS. D. STETSON.—One cylinder beam; unequal centre; two plunger pumps, one each side of beam centre, with counterbalance boxes to each, of about 50,000 lbs.; boilers under 100 lbs. steam. Advantages: Cornish economy, with double action and high steam. Comparison reserved.

E. S. RENWICK.—Two Corliss crank and fly-wheel engines; 4 horizontal plungers, connected at different angles to produce a continuous stream; shafts geared to main shaft; steam, 75 lbs.; cut-off, $\frac{1}{10}$; capacity, 20,000,000.

Objections.—Cranks and gearing involve much friction; no gain in uniform water resistance; horizontal pumping strain a loss; horizontal air-chamber objectionable; piston speed, 720 ft., too great.

I. P. MORRIS & Co.—Four Thames-Ditton or "Simpson" engines; crank and fly-wheel; double acting bucket and plunger pumps; capacity, 30,000,000. Advantages: Double action, expansion, positive control of valves; capacity, 30,000,000.

Objections.—Usual friction and reaction of crank engines; 2 cylinders and pumps instead of 1; pump charges reversed in motion; bucket valves objectionable; record of engine, London, prominent, but not equal to other engines.

WOODRUFF & BEACH.—No. 1; vertical bucket pumps under each beam-centre, working through each other; capacity, 15,000,000. Comparison reserved.

No. 2; crank and fly-wheel, with 2 reciprocating bucket-pumps worked by cams on main shaft; capacity, 15,000,000. Comparison reserved.

SINGLE ACTING BEAM ENGINES.

Five proposed, as in table, of the ordinary water-works pattern.

PLANS ADOPTED.—Assuming, as was necessary in this case, that the contract engines would not be adopted, the choice was limited to the plan which best secured its principles of action.

While the standard of capacity exceeded any other pumping engine then in use, even the enormous *Harlem Meer* itself, by at least 16%, the advantages of using large engines as to friction, counteraction, attendance, operation, and maintenance precluded the recommendation of more than one for the same work.

Watt himself, having foreseen the advantage in capacity of the double acting engine and put it in use, it was evident that a gain in this respect would be made if all his other improvements could be retained.

As the Woodruff & Beach No. 1 was presented with hinged bucket valves and without counterweights, it was inadmissible; but, at my suggestion, their proposal was modified to include double beat valves on the rods and seats, full counterweight chests, and a large air-chamber.

With this change it was evident that the reciprocal, double acting pump motion by which the buckets continually moved in line with and held the water column was greatly superior, on inspection, to the reacting plungers of the Stetson or Simpson plan, while the same advantages in counterweight with the single acting engine and the great benefit of an ample air chamber could be secured. In boilers and various details of form and construction with reference to the engine-house plan, this proposal was also preferred, as in cost, and its adoption was strongly advised.

To put the steam in this way directly on the water column on so formidable a scale was an experiment; but the Cornish engines, with much smaller pumps, had been making higher lifts with the steam-stroke and with high expansion for many years, and without the especial control of and harmony with the water column secured by this reciprocal pump action. To bring 33 tons of water quietly to rest on a bucket-pump valve with ten

MODEL; BROOKLYN No. ONE.

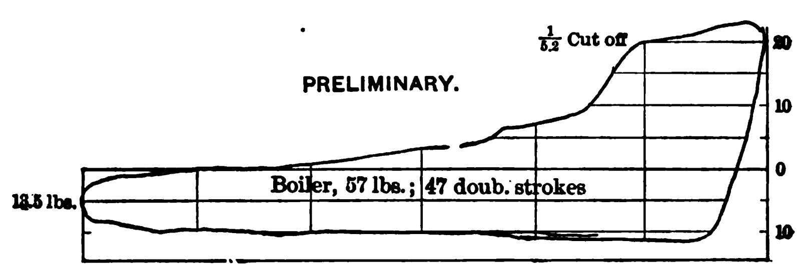


Fig. 3

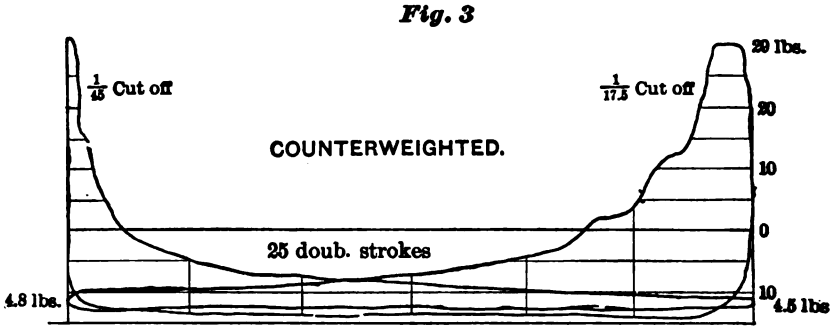


Fig. 4

double strokes per minute was an experiment, but at Huelgoat and Brittany, as in Cornwall, the action of such a column and of these double-beat valves (or equivalents) was too well defined to leave the result doubtful. To drop the usual water-works standpipe was also a change, but required less study on questions of reaction. The whole history of standpipes in this country has justified this conclusion.

In July, 1857, on the recommendation of Mr. Kirkwood, the Worthington proposal was adopted; but the contractors refused to be responsible for its performance, and in October the Woodruff & Beach No. 2 was adopted as strongly advocated by their superintendent, Wm. Wright, and put under construction. In January, 1858, before a committee of the Common Council, Chief Engineer Kirkwood and I testified that we did not approve this plan, and in March the commissioners authorized a return to the No. 1 plan.

Proceeding to Hartford to make the change, I found the firm of my opinion, but Mr. Wright strongly favored the plan under way. He yielded on condition that a large model should be built and tested against the Hartford head of over 100 ft., the result to control. A very complete and expensive model was built, of 24 x 14-inch cylinder, with $5\frac{5}{8}$-inch pumps. Mr. Wright, on the preliminary trial, wrote me, condemning it for not taking steam without a close throttle or permitting a close cut-off; but it had no proper counterweight, and when this was brought up to 2,200 lbs., by a cast-iron saddle on the beam, it gave extraordinary results, using 29 to 35 lbs. steam, with $\frac{1}{17}$ to $\frac{1}{40}$ cut-off, making 24 to 26 double strokes, with very smooth water load.

The following cards (Figs. 3 and 4) indicate the effect of increased weight on initial pressure and expansion, reduction of throttle, final and back pressures; final pressure from 13.5 (or 7.5 lbs. net), as first run, to 4.8 (or 3 lbs. net), with $\frac{1}{17.5}$ cut-off at one stroke and $\frac{1}{48}$ on the other.

In furnishing this conclusive testimony on the benefit of beam vibration and counterweight, the cards also make an effective prophecy of the action of the main engine itself, which lost half its duty, at least, by neglect of counterweight.

This return to the original plan advised was fortunate in several ways. As the engine house had been planned to take four single acting engines ultimately, the space necessary for the pump shafts of two No. 2 engines would have prevented the erection of a third engine. As to the cam motion, the Hartford experience was decidedly against its economy or efficiency.

The several plans adopted, and, in fact, the subsequent engine history, show the anomalous position of the engineer department, since the opinion of a shop superintendent, interested in patents, outranked that of the engineers in charge, and that of a contractor outranked a chief engineer. In important matters of detail, as will appear, the essential value of the engines was much reduced. It is only by the combination of professional men of education *and* experience, in society life, that they can make a stand against prejudice or ignorance in high places.

ENGINE No. 1.—The revised contract for engine No. 1 was made in March, 1858, and it was ready for preliminary trial in March, 1859.

Its dimensions were, cylinder, $90\frac{3}{16}$ inch by 10 feet 3 inches space, for about 10-feet stroke; pumps, 36 inches bore with double beat valves on the $8\frac{1}{4}$ inch rods, and annular barrels (devised by Mr. Wright) with 54-inch double beat valves; the valve seats were first filled with hard rubber, but it tended to lift, and they were packed with apple-wood, which acted admirably.

The air-chamber was of cast-iron, 6.5 feet diameter by 25 feet high. From experiments on the model, Mr. Wright thought an improvement in action would result by using a diaphragm to divide it in two chambers, with return valves which throttled off the upper contents. Its practical effect was to react, and increase the pump load very seriously (66.87 to 74.12 lbs. in one case), and the cylinder load (11.87 to 13.07 lbs.), and to produce distinct waves in the main, and it was abandoned.

The cylinder had a cast steam-jacket and double head, and was high enough to return the condensed steam to the boilers; the condenser and air pump, side pipes, double beat, filled out steam valves, and other parts were carefully proportioned and made.

The pump rods each carried a counterweight chest, one under the cylinder, the other under the outer beam centre. When it became necessary to increase the weight in September, 1859, Mr. Wright preferred to attach a vibrating segment to a shaft in the pump well, permitted for No. 1, but distinctly excluded for No. 2. The amount estimated originally was about 109 tons, subdivided for effective weights into beam 22, working parts 28, water load 32.9, counterweight 25.9 tons. Using the No. 2 beam, reduced 35 to 20.5 tons; the chests had but 13 tons, and the entire mass was about 9 tons less than the estimate for much lower piston speed. A single acting engine would have had not less than 147 tons in motion.

In the valve motions of Nos. 1 and 2, the feverishly inventive genius of a man of great mechanical ability found serious difficulty. The beautiful adjustments in pressure, load, and velocity, by which, with the equally simple cataract, the Cornish engines make their stroke and come to rest with great uniformity, were troublesome puzzles to crank engine men who were to build or to run this engine. Instead, then, of simply modifying the Cornish valve motion, complicated devices were used which did the work with some neglect of past experience.

It ought to be said, here, that no contractors could show a higher sense of contract honor than Messrs. Woodruff & Beach. No expense, care, or test, was spared to build engines of the highest class of material and workmanship. No better proof of this can be given than their repair account, in years of constant service. The following proximate schedule of materials in an engine of this size may be interesting:

Composition	6,050 lbs.
Cast steel	350 "
Finished wrought iron	48,700 "
Heavy forgings	21,200 "
Bolts, nuts, etc	32,250 "
Finished castings	137,500 "
Heavy "	471,000 "
Boiler iron work	99,800 "
	816,850 lbs.

The pump mains were of heavy patterns, carefully laid with solid lead joints, caulked inside and out; length 3,450 feet, 36 inches diameter. Two-thirds of the length has a gentle slope; the hill curve beyond, for about 1,200 feet, approximates closely that of "swiftest descent." Each has a check valve at the air-chamber, and another about 1,900 feet beyond it. They discharge at 168.8 feet above tide, reservoir level, so as to have a constant head. With a flow of 181 feet per minute the loss of head was 4.71 feet or 3.4%.

There were three boilers for each engine in a side wing, with chimney arranged for supplying two engines; shells, 8 feet diameter, 30 feet long; steam-drums, 4 feet diameter by 4 feet each; two furnaces, 6 feet 3 inches deep by 3 feet 1½ inches wide; doors fitted for air supply; four upper flues, 18½ inches diameter, 21 feet 9 inches long; lower returns, 8½ inches diameter; 7 in upper row, 6 in next, 3 in lowest; return, under boiler, to chimney conduit, 4 feet square, in brickwork; set in brickwork, tops covered with ashes, fronts felted; grate surface, each, 37.5 square feet; fire surface, 13.55 square feet, 903 effective; combustion, January, 1860, 11.86 lbs. per square foot; evaporation by tank

(212°), 9.95 (July, 1860); by volumes, 9.27 lbs. per pound coal burned; Hartford boiler, 8.39 lbs.; Belleville, Cornish boilers, 7.49.

Temperature experiments were made on the Hartford boiler of the same type to get flue action. Length, 22.66 feet; diameter, 7.5 feet; grates, 25.5 square feet; fire surface, 910 square feet; effective, 608. Thermometer showed, under slow combustion, lowest return (floor), 160° to 180°; side, 209°; centre, 240°; near shell, 284°; back connection, 334°; lower flues, 370°; middle, 400°; upper flues, 550°. This shows a rapid conduction of heat in the upper flues from about 2,000° furnace, and in 14 feet 4 inches travel, and also in the successive lower returns. With steam at 280°, the lowest return at the shell and at the back end was not cooling it, but preparing the feed supply for its upward travel to the most effective and topmost level, furnishing a fine illustration of the double return drop flue theory already stated. The Brooklyn boilers, with 8 feet diameter and 30 feet length, gave higher evaporation under more rapid combustion.

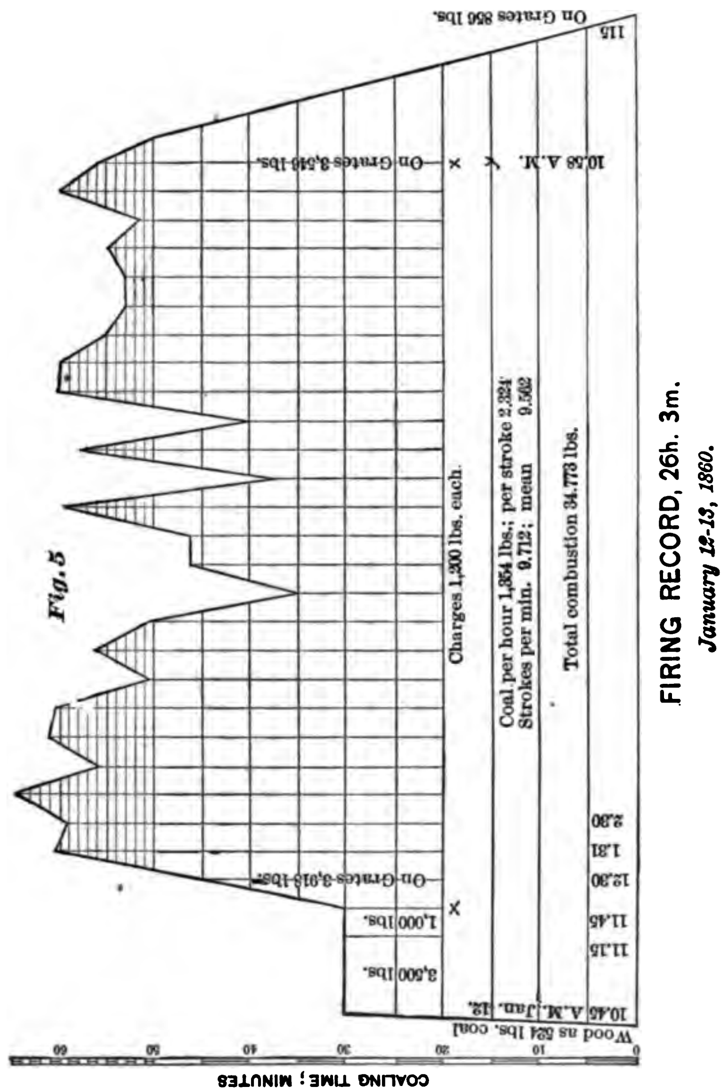
Duty Tests.—Engine No. 1 was embarrassed by being put under contract at $10,000 less than a fair price, as it was built, being a union of two different engines. The *pro rata* estimate for work in the shop was exhausted about December 1, 1858, and the balance for erection was not large. Expensive changes in valve motion and otherwise occurred. While, then, the contractors made complete construction a matter of pride, much was conceded to them within the strict accomplishment of the contract duty.

In June, 1859, the engine had been brought up to a speed which established its capacity and best motion, and the 25th to 27th, a forty-eight-hour duty test was made with a mean speed of 8.86 double strokes per minute and 425,000 ft. lbs. duty. It had been run occasionally at 10½ and 11 strokes, and was used for daily city supply, more or less.

The weight in motion, estimated, was, finished: wrought iron, piston buckets, etc., 72,000 lbs.; counterweight chests, 19,000; beam (effective), 30,000; water-load (effective), 65,000. Total, 186,000, of 243,000 calculated as required to secure proper action and expansion.

Some improvements being made September 15, the engine was tested 645 minutes, making 10.652 strokes per minute, with a duty of 487,500 ft. lbs.

Up to this time the maximum boiler pressure was about 9 lbs., and the engine was not only throttled at the steam-chest, but also by the valve motion. Mr. Wright, therefore, conceded increased



counterweight and higher steam; and December 22 and 23, on 28$\frac{43}{60}$ hours' run, with 9.342 strokes, the duty was 575,300 ft. lbs., with about 100 tons in motion.

A steam-jacket waste into the well, of about 30 lbs. coal use per hour, was relieved. Some improvements in valve motion and otherwise were made, and boiler steam was raised to 16

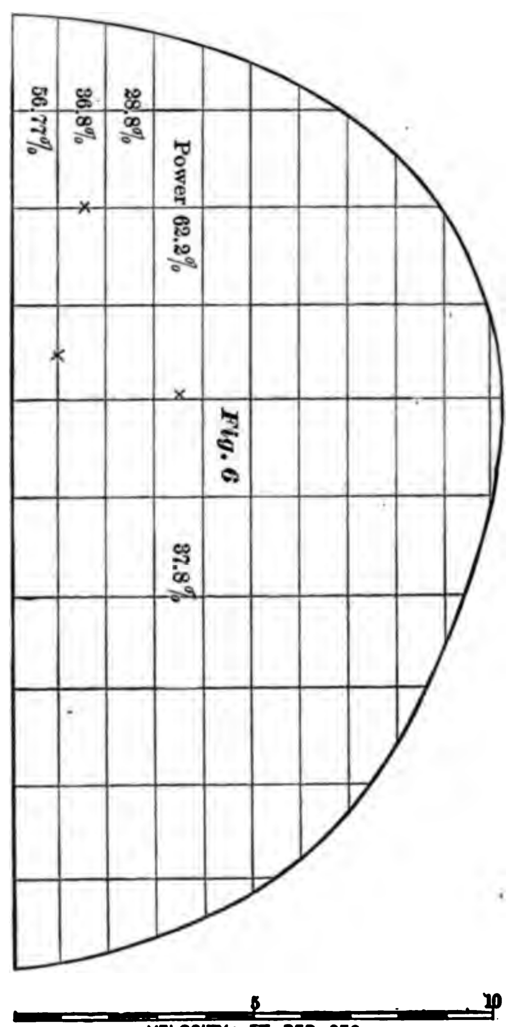
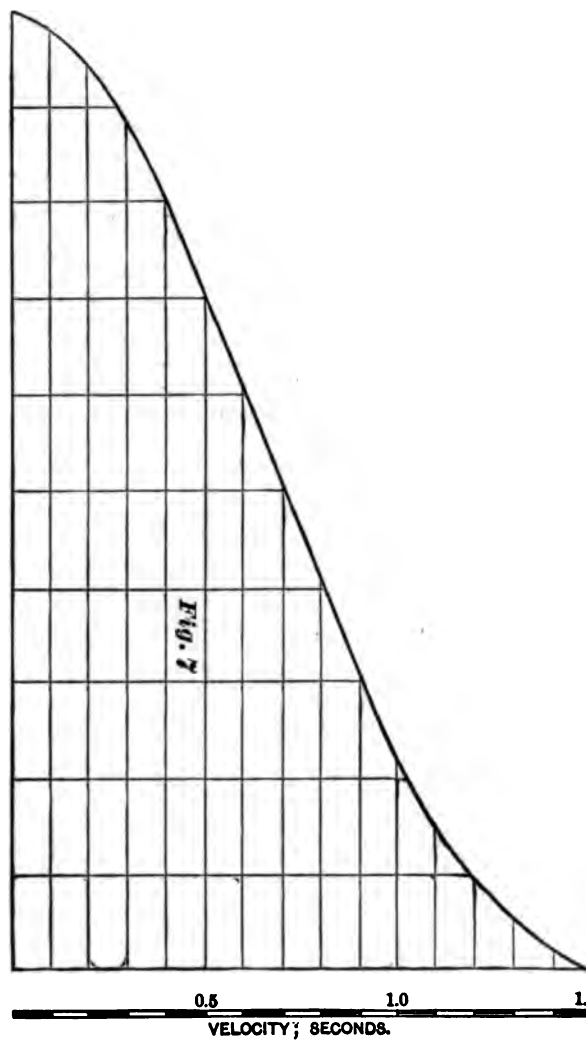


Fig. 6

or 18 lbs. A duty test of January 12 to 14, 1860, with 9.57 strokes per minute, was 611,114 ft. lbs., and the engine was accepted.

Duty Cards.—To illustrate the principles discussed in this paper, the following diagrams and cards are given:

Boiler Firing.—The objections to "parallel" coal estimates, which assume that the value of the grate contents can be determined by inspection, are shown by the firing record of the test



of January, 1860, as plotted (Fig. 5). In this case the coal charges were uniform as weighed, and used as the state of the grates required; nothing like parallel times and quantities occurs.

The system specified in the contract was to put the fire room,

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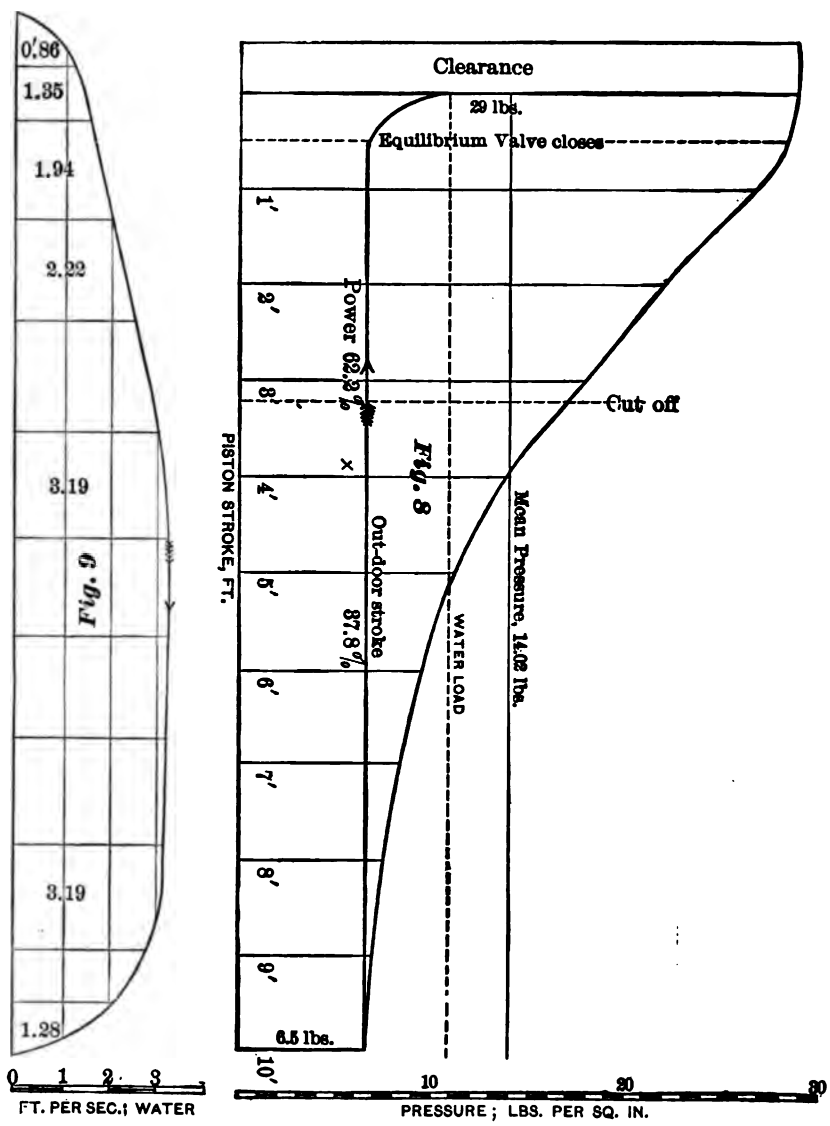
boilers, engine, and engine room at their usual working temperature; then blow off steam and clean the grates. Then starting fires, running not less than 24 hours, charging all fuel used to



the experiment less the value of grate contents, when the engine runs down on the last charge. By this process the fuel which has done the work is accurately weighed, and the contingencies of error are reduced to a minimum. Of course a longer test will be more satisfactory, but is not usually convenient. The

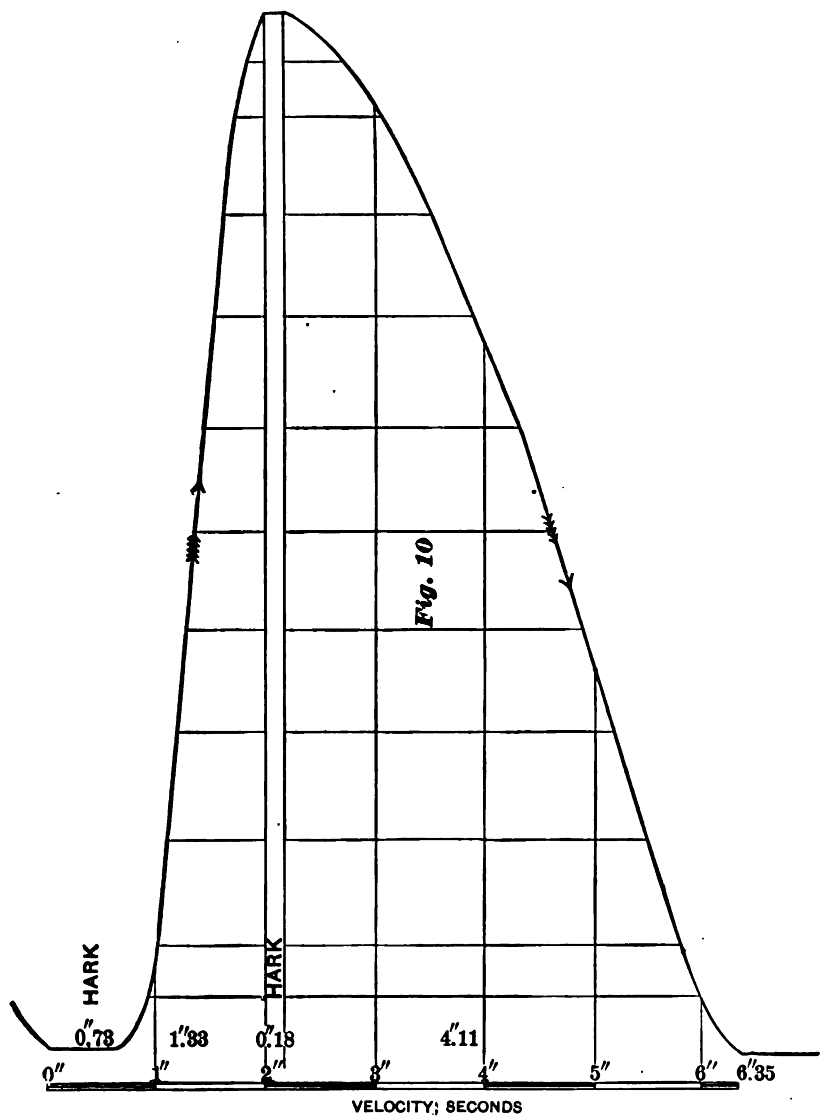
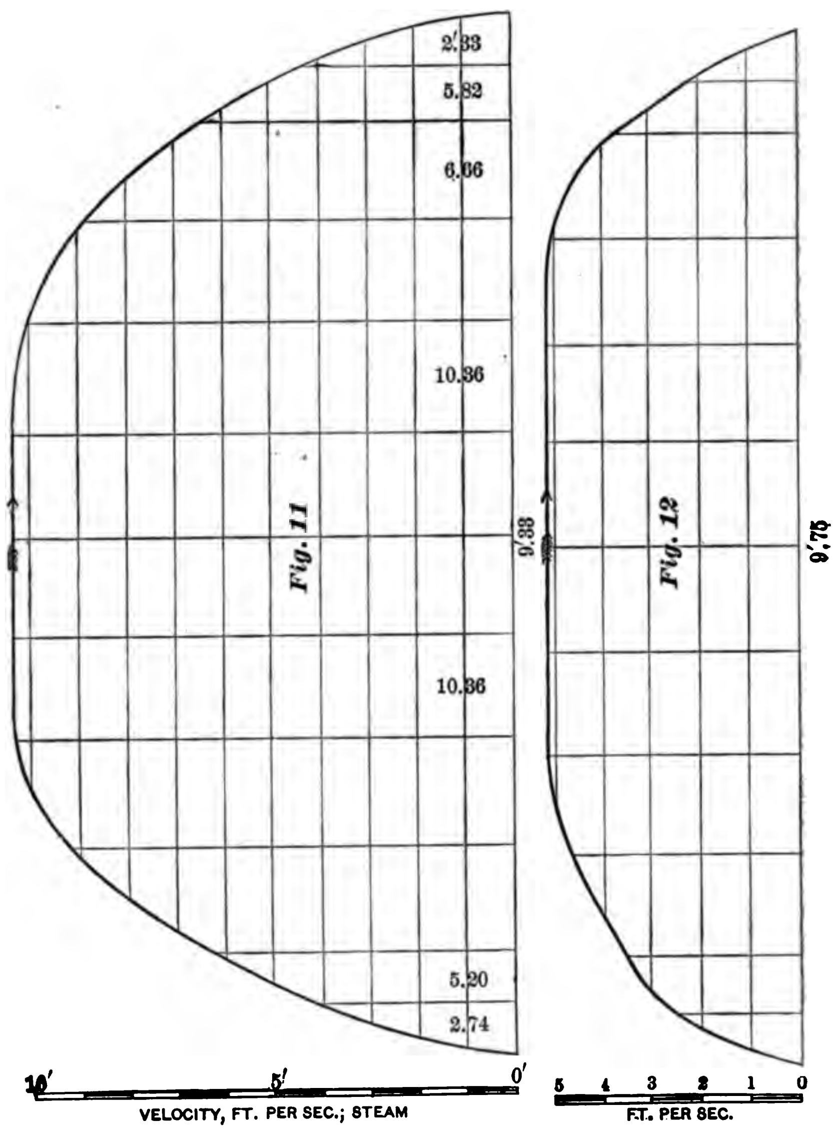


Fig. 10

Cornish system of taking all the coal used for a month, reduces the actual coal duty for banked fires and similar losses.

The propriety of using the market coal of the locality is also evident, and the objections to hand selected coal obvious.

The law by which an engine naturally reaches its maximum speed under due load and pressure, and the relative advantages of proper load to control the acceleration and retardation of



speed in travel, are shown in the velocity cards from the Old Ford engine (Figs. 6, 7, and 8), the Spring Garden (Figs. 9, 10, and 11), and the Brooklyn No. 1 (Figs. 12, 13, and 14).

7

r

N.

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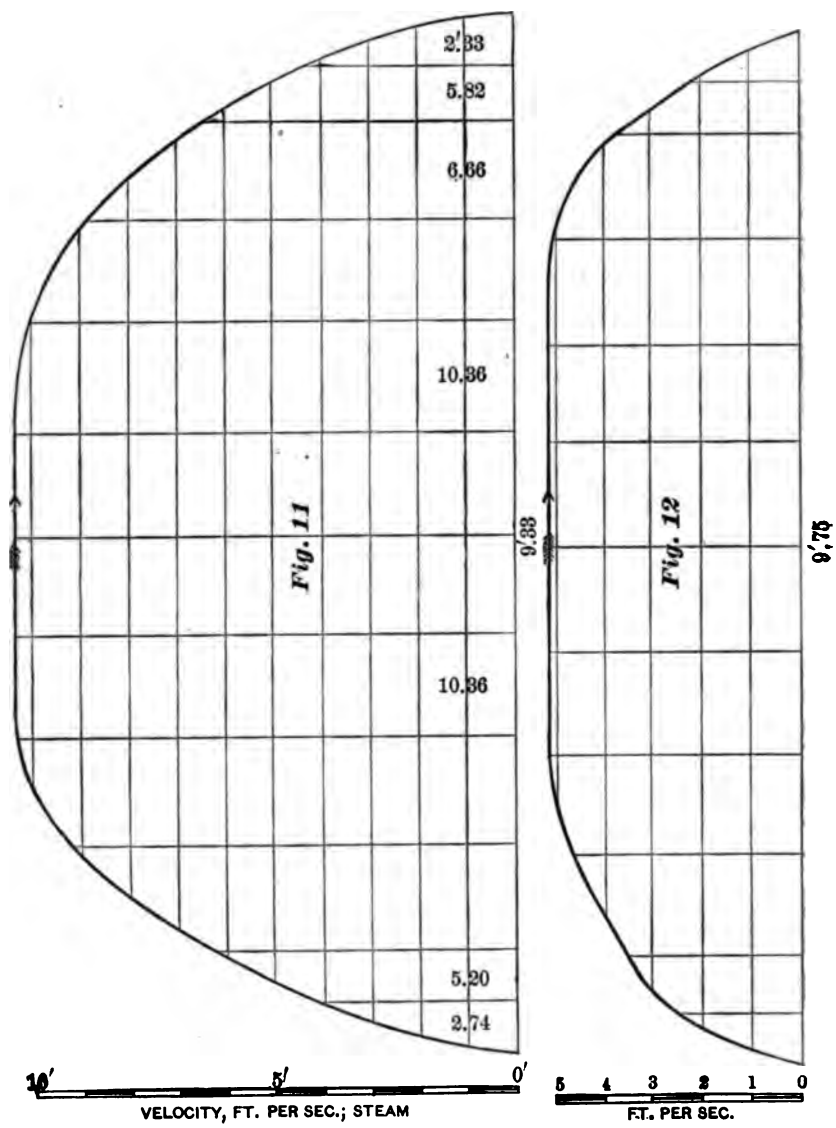
The law by which an engine naturally reaches its maximum speed under due load and pressure, and the relative advantages of proper load to control the acceleration and retardation of



speed in travel, are shown in the velocity cards from the Old Ford engine (Figs. 6, 7, and 8), the Spring Garden (Figs. 9, 10, and 11), and the Brooklyn No. 1 (Figs. 12, 13, and 14).

7

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N.

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.77
.09
.20

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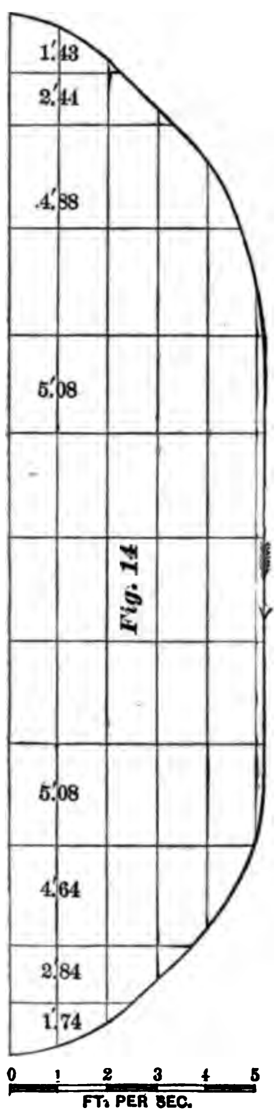
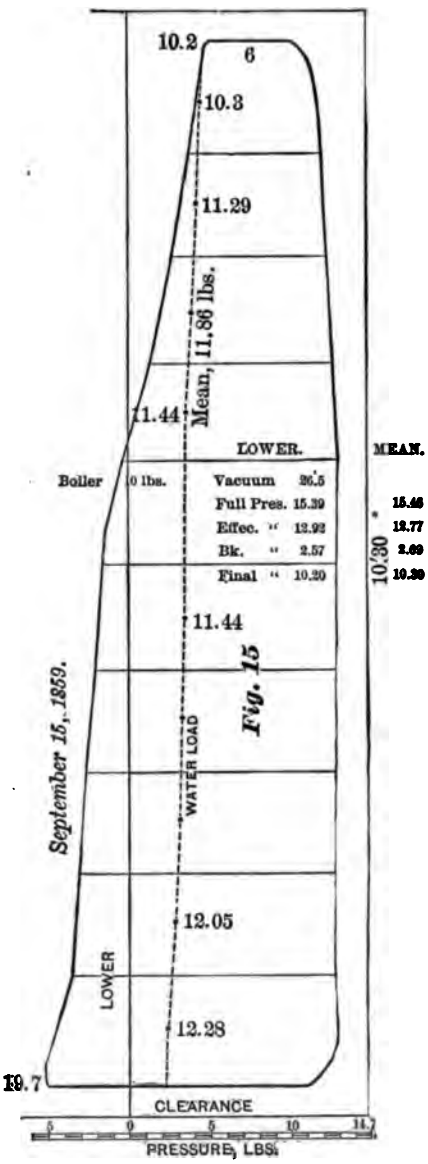
The maximum velocity in each case is attained within one-third of the stroke; but the form of the card is much superior





in the Old Ford, with higher initial pressure, to the low, throttled pressure line of the Brooklyn. The effect on final pressure is very material as a comment on short counterweight in the latter; and the water card of the Spring Garden shows defective weight there. The prolongation of maximum speed early attained, tends to increase final pressure and consequent steam

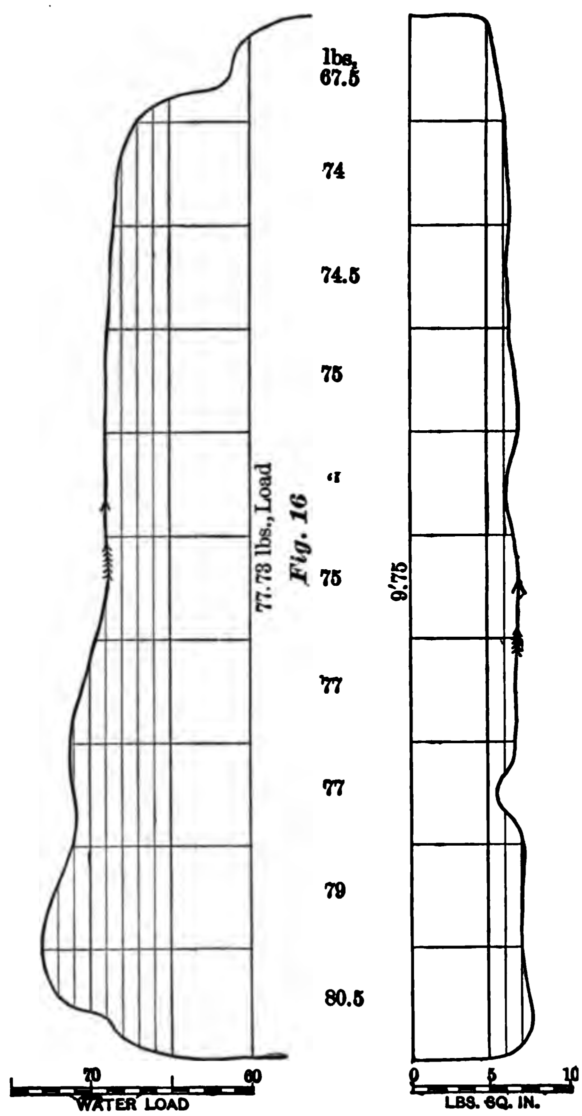


Fig. 16

use. The theory of uniform resistance is therefore improper, and the cut-off a mechanical necessity.

It is also demonstrated that improvement in initial stability of piston in pumping reduces mean resistance. Not only does the Old Ford card, opening with 29 lbs. steam, end with a full cylinder at 6.5, while the Brooklyn, with 19.7, ends with 10.2;

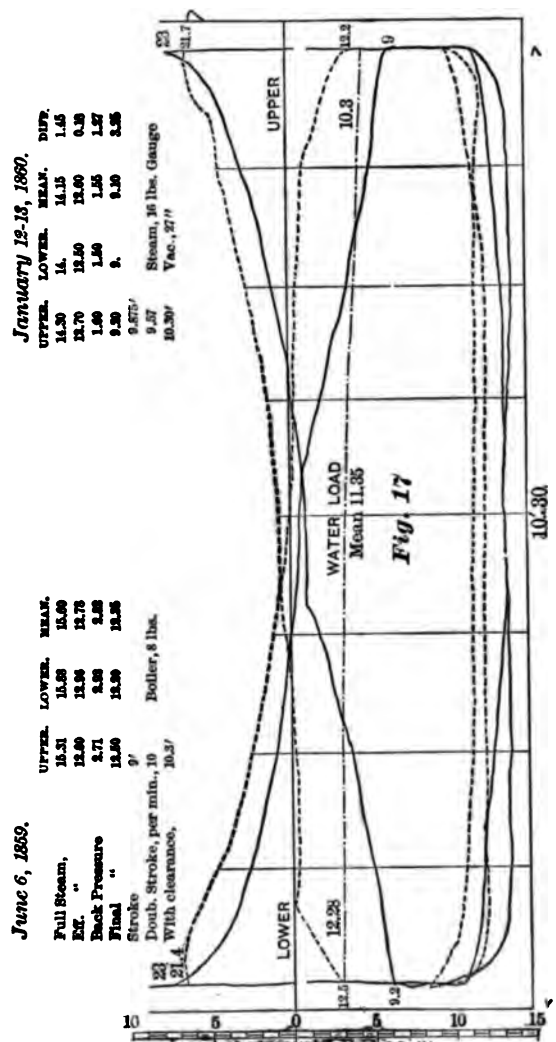


Fig. 17

but the cards of this and the model engine show reduced mean resistance with increased initial pressure, because working in harmony with the mechanical law of *vis viva* as to both engine and water mass. Another important gain is shown in the increased length of stroke due to better control of final motion, which gains in water delivered and in reduced clearance loss per stroke. In June the travel was generally 9.33 feet; in September, 9.66 to 9.75 feet; in January, 9.875; and with additional

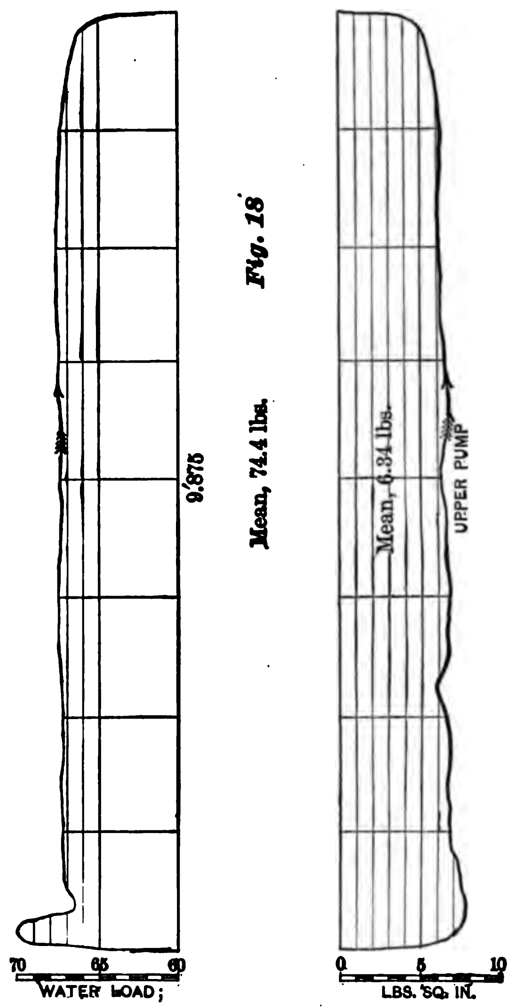


Fig. 18

load, to reduce final pressure, it could be made in 10.3 feet space, about 10.1 safely, with our carefully built spring beams.

The comparative steam cards of June 6, 1859, and January 12, 1860 (Fig. 17), further illustrate the effect of increased weight on final pressure.

The pump card of January (Fig. 18) also shows the improvement on the card of September 15 (Figs. 15 and 16).

make their stroke and come to rest with great uniformity, were troublesome puzzles to crank engine men who were to build or to run this engine. Instead, then, of simply modifying the Cornish valve motion, complicated devices were used which did the work with some neglect of past experience.

It ought to be said, here, that no contractors could show a higher sense of contract honor than Messrs. Woodruff & Beach. No expense, care, or test, was spared to build engines of the highest class of material and workmanship. No better proof of this can be given than their repair account, in years of constant service. The following proximate schedule of materials in an engine of this size may be interesting:

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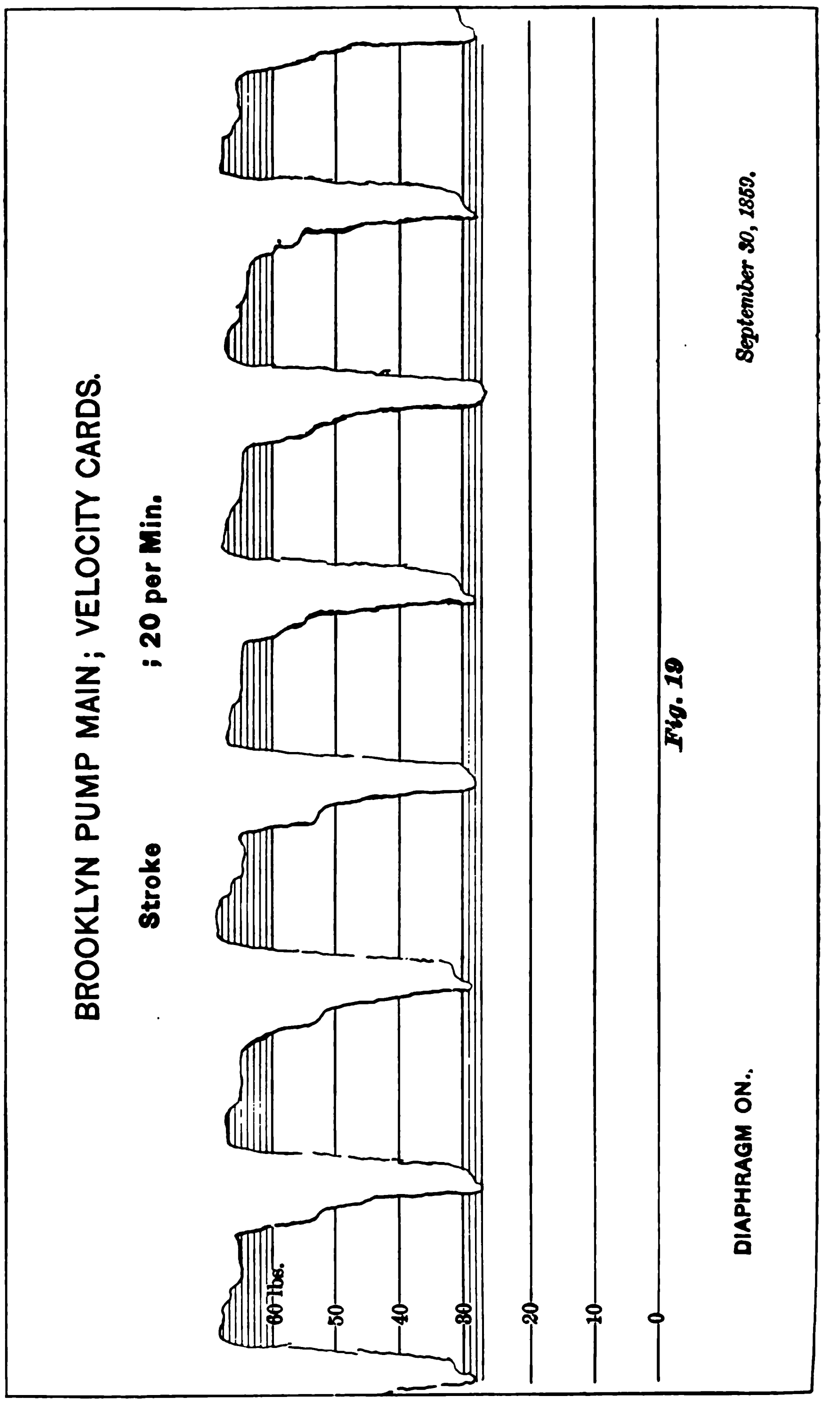

BROOKLYN PUMP MAIN; VELOCITY CARDS.
Stroke ; 20 per Min.
60 lbs.
50
40
30
20
10
0
DIAPHRAGM ON.
September 30, 1859.

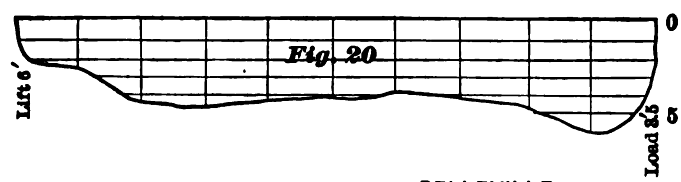
Fig. 19

This is shown by the velocity card of September 30, 1859 (Fig. 19).

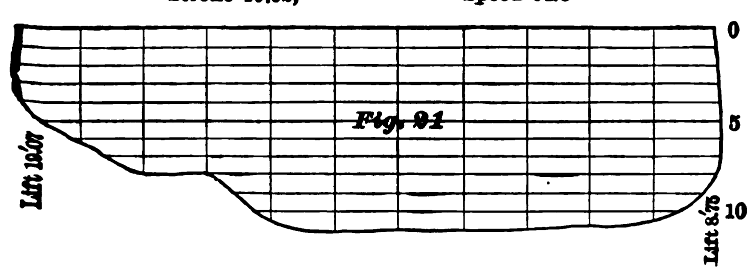
There is an important distinction between the upper and lower faces of any pump piston in action. While the load on

SUCTION CARDS.

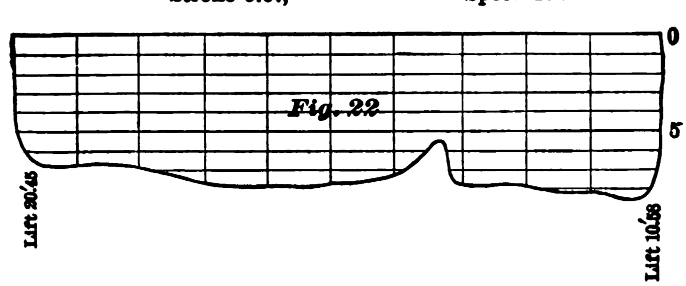
CORNISH ENGINE, PHILADELPHIA,
Stroke 9'.5, Speed 94 ft. per min.



BELLEVILLE.
Stroke 10'.92, Speed 64'.8



BROOKLYN.
Stroke 9'.87, Speed 184'



the upper side is the entire mechanical water lift, and depends on the relative level of the piston, below the piston the atmosphere makes the lift, and the resistance is a simple question of vacuum formed; and this is to an important extent independent of the relative height of the piston above the water, and shows the necessity of engine inertia to prevent too rapid a start.

The law of *vis viva*, by which the resistances are reduced instead of increased in the water column in motion as its lift increases, is shown in the annexed cards (Figs. 20, 21, and 22).

In the first card, the conditions of free delivery to the plunger barrel are very favorable, since there is a water head of 3.5 feet, and very large valves, with a short passage from the forebay; yet from too rapid motion or other cause, to start the column and maintain its flow, the indicator opens with a vacuum of over 3 lbs.; at the first foot, 6.25 lbs.—average, 4.25 lbs.; showing at the first foot a vacuum equal 14.37 feet lift under an actual load of 2 feet, and at the end of the stroke the vacuum is 2 lbs. or 4.6 feet, when the actual lift is 6 feet above the well. The pump might have been about 15 feet higher with so much less water load.

In the second card (also a plunger, single acting), it is evident, as the velocity water stroke of the first engine shows, that there is a lack of counterweight. Here the vacuum is 8 lbs, or 18.4 feet lift, against 8.15 actual; at the first foot, 24.15 feet against 9.15; at the end, 4 lbs., or 9.2 feet, where the actual lift is 19.07 as the *vis viva* effect of the column; and the mean vacuum is 9.34 lbs., or 21.48 feet, against 13.61 actual, another waste of power.

In the third card the upper pump does its work about 1.39 lbs. per square inch less than the lower, which starts with a bucket load of about 3 feet. The lift is 10.58 feet, and the vacuum 17.25 at the start; at one foot pull, 20.7 feet; lift, 11.58; final pull, 6 lbs., or 13.8 feet, with an actual lift of 20.45, or 8.9 lbs.; mean lift, 15.51 feet; vacuum, 17.52.

This engine was doing nearly five times the work of the second, and the economy of pump position is clearly shown, involving a gain peculiar to this form of engine.

The Hartford and Cambridge pump cards show similar action (Figs. 23, 24, 25).

In these pump cards the same law of *vis viva* developed by air pressure is proved in direct mass pressure or steam.

In addition to cards given, the following have certain important bearings:

The Belleville counterweight (standpipe), without air-chamber relief, opens the card with a load of 66.5 lbs., or 153 feet lift, against an actual of 139.76; the surplus work then reduces

Boiler Firing.—The objections to "parallel" coal estimates, which assume that the value of the grate contents can be determined by inspection, are shown by the firing record of the test

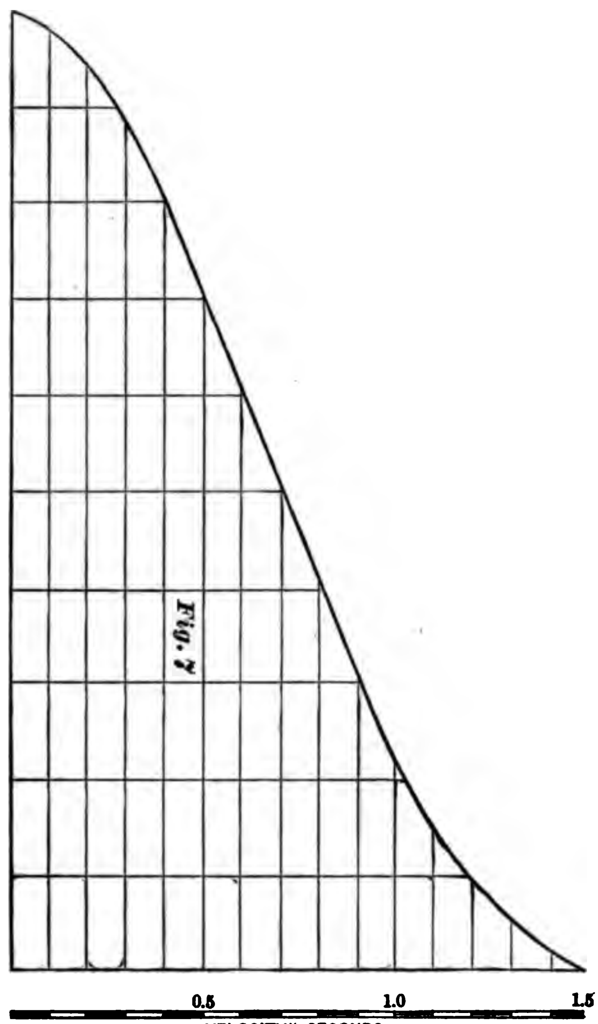
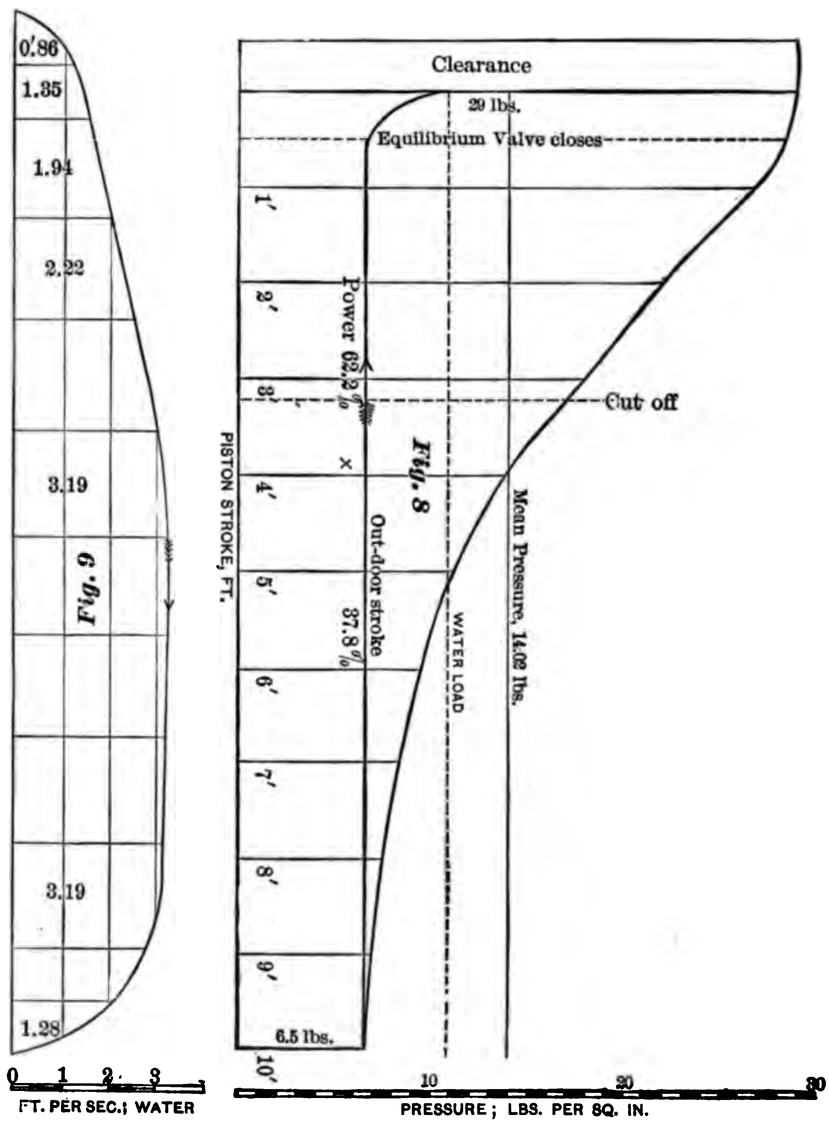


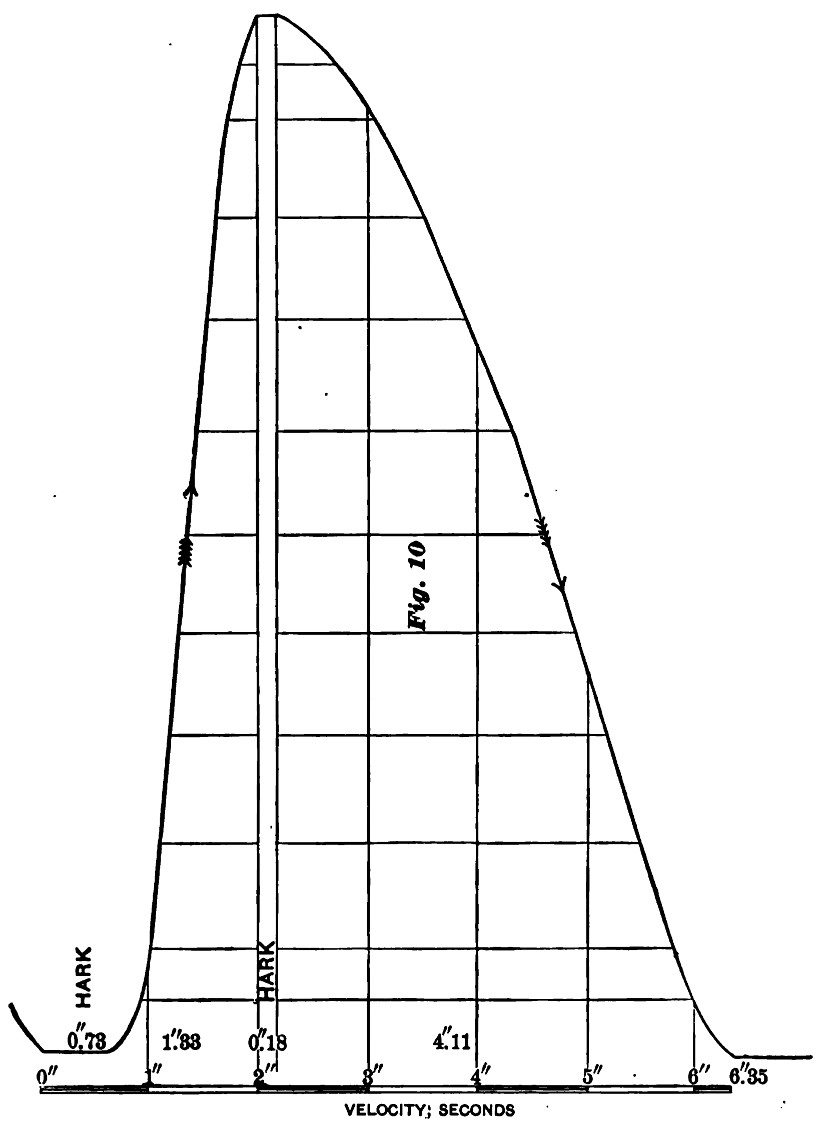
Fig. 7

of January, 1860, as plotted (Fig. 5). In this case the coal charges were uniform as weighed, and used as the state of the grates required; nothing like parallel times and quantities occurs.

The system specified in the contract was to put the fire room, boilers, engine, and engine room at their usual working temperature; then blow off steam and clean the grates. Then starting fires, running not less than 24 hours, charging all fuel used to



the experiment less the value of grate contents, when the engine runs down on the last charge. By this process the fuel which has done the work is accurately weighed, and the contingencies of error are reduced to a minimum. Of course a longer test will be more satisfactory, but is not usually convenient. The



Cornish system of taking all the coal used for a month, reduces the actual coal duty for banked fires and similar losses.

The propriety of using the market coal of the locality is also evident, and the objections to hand selected coal obvious.

the load to a final resistance of about 60.5 lbs., or 139 feet, the actual being 150.68.

With all its care in design and reciprocal pump relief, but with

BROOKLYN ENGINE; Lower Pump.

January 13, 1860.

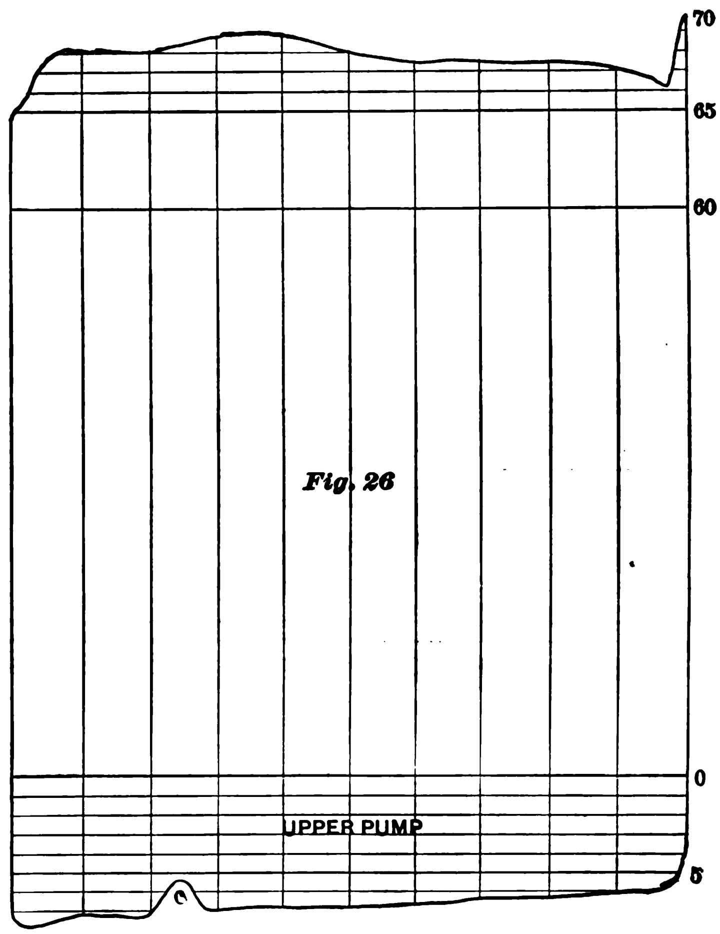


Fig. 26

4 pumps acting on the same 12-inch main, the Hartford pump opens with 61 lbs., or 140.5 feet lift, against an actual of 111.76, or 48.6 lbs.; the wave reduces the load to 49 lbs. at the close, or 112.7 feet, against an actual of 110.43.

The Cambridge horizontal pump, opening with a partly unfilled

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chamber, and without much weight, strikes solid water with a blow of 41 lbs., which includes some spring vibration, against 27.8 actual, and ends with 32.

In these, and abundant other cards, the same law of wave generation, acceleration, and retardation is demonstrated in full confirmation of theory. What is true of the cylinder card is true also of the pump card under a fixed law.

In further demonstration of the modifying effects of counterweight and air cushion on the pump wave, the following average card, taken in the capacity (Brooklyn) test, with about 10¼ strokes per minute is given (Fig. 26). While the same want of weight is shown in the initial blow, the rapid reduction of reaction is also shown, and the return control of the air-chamber (which absorbs part of the initial work) tends to equalize the wave motion, the practical reduction of final resistance being about 4½ lbs.; the air-chamber in this case acting on the main, in its turn, to propel the wave from the power stored in it at the beginning of the stroke.

ENGINE No. 2, after the acceptance of No. 1, was built substantially a duplicate, except that the cylinder bore was 85 inches, the capacity of the counterweight chests was enlarged (though not properly used), the valves of each annular pump barrel were changed to 8 double beat foot valves, instead of one upper 54-inch (without any special gain), with some minor changes. It was tried May 21, 1861, and tested February, 1862; duty reported being 619,037 ft. lbs.

My resignation being made in 1860, the engine was not built under my charge, and like No. 1, it never was properly loaded, being worked with low steam and a throttle.

It has been stated at various times that these engines were patented by Mr. Wright. The annular pump barrels were patented by him in 1859; the diaphragm, and some parts of the valve motion, were his inventions; but the engine itself, as a combination of counterweighted, double acting pumps, with double beat valves on the rods, was not his improvement, and not patented.

Prospect Hill engine was built with a fly-wheel; with two lifting pumps worked from the beam; 4.5 feet cylinder by 34 inches; pumps, 41 inches by 20; boiler, double return drop flue, 18 feet by 6 feet beam. It takes its supply from the 36-inch city main, at the corner of Underhill Avenue and Warren Street;

engine-room floor level, 119.5 above tide; reservoir flow line, 197 feet above; feeding main at pumps, about 106 feet.

The action of the pumps under trial was defective; with 50 lbs. steam and one-fourth to one-fifth cut-off, the duty was limited to 500,000 ft. lbs. Mr. Wright, therefore, put in a new set, with easy entrance and delivery lines, with a large air-chamber on the supply, embarrassed by surplus pressure. This brought the engine up in duty, so that under test of May, 1862, it made 649,577 ft. lbs. as reported, "parallel" coal estimates being used on 93 hours run.

Engine Experiments.—During the winter of 1857–58, trials were made of the Belleville, Hartford, and Cambridge engines. The results are collated in the following table. Messrs. Worthen, Copeland, Graff, and Morris were in charge.

PUMPING ENGINE TESTS.

Location.	Brooklyn.	Belleville.	Hartford.	Cambridge.
Class.	Double acting beam.	Single acting beam.	Geared.	Annular.
Cylinder, stroke, feet	10	11	5	4.5
bore, inches	$90\frac{3}{16}$	$80\frac{3}{16}$	$82\frac{1}{4}$	12, 14, 825
Pump, stroke, feet	9,875	10,923	2.688	(doub.) 4,362
bore, inches	36	34.75	$18\frac{3}{4}$ and $19\frac{5}{16}$	14
number	2		4	1
cubic feet, contents, effective	187.65	66.172	(4) 19.309	(doub.) 4.24
strokes per minute.	9.57	4.418	each 8.158	" 19.21
number	(doub.) 14,965	5,240	(4) 15,860	" 14,331
time, minute	1,563	1,178	486	746
discharge, cubic feet	2,000,000	344,360	76,560	60,762
gallons per day ( ),	14,418,641	3,287,860	1,771,396	915,510
friction per cent.. lift	6.25	7.25	17	15
loss of action, per cent	1.69	8	6.56	assumed 7
Coal burned, pounds	34,773	5,331*	1,117*	541.5
Pump main, length, feet	3,450	2,359	6.879	2,350
diameter, inches	36	36	16	12
feet per minute, velocity	181	avg.41.36	112.19	106.05
actual lift, feet	160	158.83	118.21	72.29
equivalent lift, feet	170	167.76	138.34	83.14
Total pumping friction, per cent. actual discharge and equivalent	5.89	5.92	14.56	18.29
Duty; head × strokes ÷ coal, ft. lbs.	611,114	680,660	591,505	551,261.

* Parallel estimates used; results in question.

Annual Operation.—The engine house and its appurtenances under the original contract provided for four engines, each of not less than 15,000,000 per day. The average consumption of 1889 was 52,191,128; the maximum, December, 55,112,699 gallons. With four superb engines working side by side, two of which could easily have been enlarged to lift 20,000,000 each, this engine house would have had no parallel; but the provisions of the original plan have been neglected. In the main room there are now three engines; in a side room excrescence, built on the front, there are four horizontal "Davidson" engines to do the work of one; and in a very costly engine house on the aqueduct, several hundred feet south, two vertical compound "Worthingtons" are being completed. Engines Nos. 1 and 2, under the charge of a man who believed in crank centres and throttling, and cared little for duty, did their work quietly and regularly, with remarkably light repair accounts. In 1866, No. 1 ran 3,824 hours; engine repairs, \$160.22; No. 2, 3,066 hours; repairs, \$171.84.

In 1866, Brooklyn began an extravagant system of water expenditures in various directions, under which, during 30 years' supply, with a revenue of \$26,645,902, the outlay has been for operation and maintenance, \$9,543,000; interest, \$16,698,000; original construction, \$5,440,000; extensions, about \$11,620,000; or about \$43,301,000.

In 1866, a full coat of ashes for the boilers was replaced by an expensive felt cover. In 1867, the 15-inch beam pin was enlarged to 20, and a new pump head made for No. 1. In 1868, an 80-inch cylinder was ordered as a substitute for No. 1, 90 inch, and a new set of pumps at \$27,000. In 1870, its alteration to a crank engine was decided. In 1871, three new boilers were set, though the other three were in use in 1885.

Of No. 2, the report for 1873 says: "With the exception of new brasses to the beam pillow blocks, and new valves to the pumps, no repairs are anticipated on this engine." In 1881, No. 2 ran 7,420 hours; in 1886, 7,198; 1889, 8,149. A continuous service of 30 years is shown by this engine of 1861.

These engines cost \$138,000; No. 3, of 1869, \$129,750; No. 4, 1883, with building and main, \$127,398; net, \$70,000; No. 5, 1888, 2 engines, 10,000,000 gallons each, \$190,471; house very expensive.

No. 3, as tested, had 178.43 feet lift (No. 1, 170 feet), 7½% loss of action (No. 1, 1.69); could not be continuously run; credited with 683,872 ft. lbs.; duty of 1874, 590,876; repair bill, 1872, \$5,725.

No. 4, lift, 179.21 feet; duty, by *capacity* of pumps, 657,561 ft. lbs.; 1886, 550,988. Fig. 27 shows the full elevation of No. 1 engine.

9

DISCUSSION.

Mr. Chas. E. Emery.—I have been very much interested in the paper. I recall the time when there was considerable discussion in relation to these engines, some claiming that they were not nearly as economical as they should be. It is obviously unfair to compare their performance with others which have since been constructed. On examining the paper with its statements of the actual results which had been accomplished previous to the time when these engines were designed, the conclusion is inevitable that the very best judgment was used, based on the information available at the time, to obtain a plant equal in economy and superior in mechanical detail to what had previously been accomplished. It is really very gratifying to note what can be done in a single building by employing a number of large engines. The view as I have seen it is an impressive one. The conditions have changed of late years and different apparatus is now available, not only more compact but more economical; but these facts do not in the least detract from the credit due for the sightly appearance and very excellent performance of the engines described in this paper.

Mr. Wm. Kent.—I would ask if there is any particular reason for mentioning duty per foot pounds sometimes, and in another place duty per 94 pounds of coal. I know there is a difference between American and English practice. But in some places the duty is sixty millions and in others ninety thousand. I would like to ask if there is any particular reason for having it expressed in different terms in different parts of the paper.

Mr. McElroy.—The answer to the question is simply this: The English practice differed. They changed the weight of their bushel. For a long time it was 94 pounds and then they altered it to 112. In quoting the record of experiments it was necessary to distinguish between the weights of the bushels used.

Mr. Kent.—What I mentioned was that in some places you give duty per pound of coal, and in other places per hundred pounds.

Mr. McElroy.—I reduced it in several cases.

CCCCLXX.*

TEST OF INDIANA BLOCK COAL AT THE CHICAGO WEST SIDE PUMPING STATION.

BY A. F. NAGLE, CHICAGO, ILL.
(Member of the Society.)

THE following test was made by the writer during the month of August, 1891, and the report of results presented to Mr. L. H. Clark, City Engineer of Chicago, for whom the test was made.

DESCRIPTION OF THE BOILER PLANT.

1. The boilers used during the test were what are known as the southerly batteries, consisting of three batteries of two boilers each. One battery is usually out of use for cleaning, the remaining two being large enough to run a pair of engines.

2. The boilers are of the ordinary return tubular type, exceptionally well-proportioned and well-set. The gases return over the tops of the boilers, and the steam-pipe is run back and forth in this flue chamber, thus superheating the steam slightly (14.14°). This, however, is all lost by the time the steam arrives at the engine, as is evinced by the fact that the steam-trap connected to the bottom of the steam-chest still blows some water.

3. There is a small flue heater for heating the feed-water. It is very small, and as it is not kept clean automatically, nor in fact can any cleaning be done except by the entire removal of the heater, it is of very little value. A small allowance, however, is made for it (9°).

4. There is also a "smoke consumer" consisting of air and steam jets above the fire, but it was not in use during the test.

5. The draught is regulated by hand dampers.

6. The condensed water from the steam-jacket usually drains directly back to the boilers, but during the test the steam-jacket was shut off entirely.

*Presented at the New York Meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

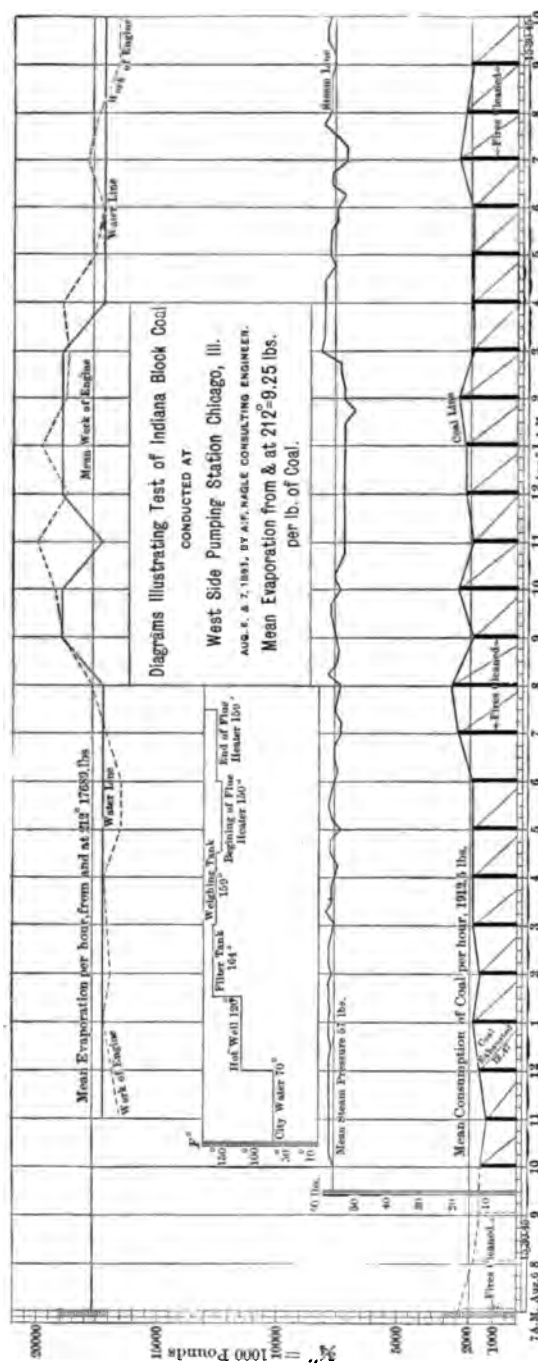
Diagrams Illustrating Test of Indiana Block Coal
CONDUCTED AT
West Side Pumping Station Chicago, Ill.
Mean Evaporation from & at 212°=9.25 lbs.
per lb. of Coal.
Mean Work of Engine
Water Line
Steam Line
Coal Line
Mean Steam Pressure 57 lbs.
Fires Cleaned
City Water 70°
Filter Tank 164°
Weighing Tank 150°

Fig. 28.

7. The feed-water is taken from the hot well of the engine (about 120°) and passed through two filters, where it is heated about 40° by the exhaust steam of the little donkey pumps. It can also be heated at this point by live steam, but this was not done during the test.

8. The water enters the boilers free from lime and other impurities. It is, however, necessary to blow off a little every eight hours, but this was not done during the test.

9. The dimensions of each boiler are as follows, there being four in use:

Diameter of shell	84 inches.
Length of shell and tubes	18 feet.
Number of tubes (4½ in. outside diameter)	68
Area of water-heating surface	1,602 sq. ft.
" " steam " "	356 "
" " flue heater "	128 "
" " grate	82 "
" " through bridge wall	6.70 "
" " " tubes	6.72 "
" " chimney per boiler	8.62 "
Height of chimney	125 feet.
Ratio of grate to water-heating surface	1 to 50
" " steam " "	1 to 11
" " flue heater "	1 to 4
Horse-power of boiler (basis of 12 sq. ft.)	184

CONDITION OF THE BOILER PLANT.

10. The boilers are eight years old, but new tubes were put in several years ago. They are now always practically free from scale. One battery, however, is withdrawn from use every three weeks and cleaned thoroughly inside and outside. At the time of the test, one battery had been in use about 4½ weeks and the other 1½ weeks, that is, the boilers were tested during the middle of their running period.

11. Just before starting the test, the tubes were blown out by a steam jet. The brickwork, gates, pipes, valves, and gauges were all in first-class condition.

TESTING APPARATUS.

12. The scales used for weighing the coal were the same as are in daily use, but adjusted by myself by standard weights kept at the station.

13. The feed-water was taken from two cast-iron tanks of

known capacity, and by a quick lever motion the valves were reversed to the pump. The tanks were filled to an overflow pipe and then drawn down to a plain copper ring fastened to the glass water gauge. In the southerly tank there were 1,566 lbs. at 74° Fahr. between these two points. The same tank gave 1,570 lbs. at 117° Fahr., an increase of 4 lbs. for 43° rise of temperature. As the mean temperature of water during the test was 159°, an increase of 42° above that measured, 4 lbs. again were added, or a total of 1,574 lbs. was taken as the weight of water in the tank at 159° temperature.

14. In a similar manner, the water in the northerly tank was found at 159° to be 1,581 lbs., or a mean of the two tanks of 1,577.50 lbs.

15. The scales used for the water-tanks had been recently repaired and tested by myself with standard weights and found to be correct.

16. The steam-gauge at boiler was tested in my presence with a test-gauge and found to be correct. Three pounds, however, must be deducted from the log readings, as a water-column formed in the pipe leading to the gauge.

17. The thermometers were furnished by myself, and the same instrument was used at the several points when readings were taken. Great care was taken to obtain correct temperatures at the several points. It may be well to explain that when the feed-water in the measuring tank had run at a uniform temperature for half an hour or more, six buckets of water were drawn from the feed-pipe at its entrance to the mud-drum, and after the first two the temperature would run uniformly, and found each time, within 1° or 2°, the same as at the tank. The water was, of course, restored to the tanks.

18. In the fire-room the fires are cleaned at 7.00 A.M. and 7.00 P.M. The test began at 10.00 A.M. As a matter of comparison and check, the coal used during these three preceding hours was also recorded. Coal was weighed out at the beginning of the hour sufficient to last during the hour, if possible, and thus hourly totals were obtained. The fires were light, hence frequent firing and less opportunity for overstocking the furnaces. The coal was weighed by an assistant, and he or his substitute never lost sight of the coal pile. The coal was taken indiscriminately from different parts of the coal shed. No error is believed to exist in the coal record.

19. The ashes were weighed both wet and dry, and a fair judgment was that the total weighed was equivalent to about 85% dry, and this net amount is recorded.

20. The water record was kept by two persons, the lever being operated by one man at the instant the word was given by an observer. It is believed that no errors of readings occurred greater than ½ of 1% if they had been cumulative, but usually errors of this kind neutralize each other.

21. A record was kept of the revolutions and water pressure of the engines as a check upon the steam used, the product of these two figures corresponding closely to the steam used.

22. The usual and proper precautions were observed throughout to obtain reliable and accurate data, and no errors of any magnitude are believed to exist.

RESULT OF TEST.

23. The results of the test are given in the following table. Date of test, August 6 and 7, A.D. 1891.

Duration of test	24 hrs.
Total coal consumed	45,900 lbs.
Ashes	3,337 "
Percentage of ash	7.25
Coal per hour	1,912.50 lbs.
Coal per hour per square foot of grate	15 lbs.
Combustible per hour	1,773 "

TEMPERATURE.

Temperature of city water	70°
Temperature of hot well	120°
Temperature at filter-tank	164°
Temperature at weighing-tank	159°
Temperature at flue-heater (estimated)	150°
Temperature at mud-drum (boiler)	159°
Temperature of escaping gases	557°
Temperature of steam, amount of superheating	14.14°

WATER.

Water evaporated from feed	388,065 lbs.
Water evaporated per hour from feed	16,169 "
Water evaporated per hour per square foot of heating surface	2.53 lbs.
Water evaporated per hour from and at 212°	17,689 lbs.
Water evaporated (feed) per pound of coal	8.45 lbs.
Water evaporated (from and at 212°) per pound of coal	9.25 "
Water evaporated (from and at 212°) per pound of coal, allowing for superheat	9.31 "

Water evaporated (from and at 212°) per pound of combustible...	9.98 lbs.
Water evaporated (from and at 212°) per pound of combustible, allowing for superheat....	10.05 "
Boiler pressure, net....	57 lbs.
Average revolutions of engine per minute....	11.02
Average water pressure....	100 ft.
Horse-power (basis of 34.50 lbs. of water from and at 212°)......	128 H.P.

EXPLANATION.

While the average feed-water in the measuring tanks was 159°, several careful tests indicated that it was no hotter than this after passing through the flue-heater and just before entering the mud-drum. As the coal is entitled, however, to the credit of imparting what little heat it may to this heater, as fair an estimate as possible was made of the heat lost between the tank and flue-heater, and it has been taken at 9°.

DIAGRAMS.

24. The lower lines in heavy black (Fig. 28) indicate the coal weighed out at the beginning of the hour, and a light sloping line indicates the time at which the coal was exhausted.

25. The water-line is to the same scale, and is corrected for equivalent water from and at 212° temperature.

26. The work of engine (revolutions multiplied by water pressure) is plotted on a scale corresponding to the mean water-line for its mean work, and the fluctuations above and below are noted in dotted lines. The discrepancies in these lines at 11.00 P.M. have been carefully looked up and verified. Such temporary variations are made possible by the large volume of water in the boilers compared with the amount used. A difference of $\frac{3}{8}$ inch in the water-level would furnish the extra steam temporarily required.

27. The steam pressures are recorded and plotted every 15 minutes, and corrected for water-column on gauge.

28. A little diagram in the centre illustrates the progressive increase of temperature of feed-water.

DISCUSSION.

Mr. Wm. Kent.—This paper does not contain, as I see, any particular point of interest, except that there was a test made of an ordinary form of boiler, with a coal which is not described by its analysis or any other designation except its name—Indiana block coal—and that it gives a result which is rather low. The flue temperature of the gases is rather high, and there is no explanation given why this temperature of the gases is so high when the boiler evaporation is so low. I think the chief effect of the paper may be to induce people to ask questions why the result was so bad. Instead of low temperature of flue gases and high economy, which we ought to expect from the low rate of driving, we find high temperature of flue gases and low economy. This low result of 10.05 lbs. evaporation from and at 212° per pound of combustible is what ought to be explained. We know that with anthracite coal we can get 12 lbs., with Cumberland coal 12½ lbs., under the best circumstances, but not with such high temperature as this. But we do not know what can be expected from Indiana block coal under the best possible circumstances. Mr. Nagle's paper leaves the question as it has been. We do not know what Indiana block coal is capable of doing, and this paper does not tell us.

*Mr. A. F. Nagle.**—" Indiana block " is a coal as well known in the Chicago market as Cumberland is in Boston, or Lackawanna in New York, and as I knew of no authoritative record of the evaporative power of this coal, I thought the foregoing report might be of interest to our Western members, even if it had none to some of our Eastern.

As to the results, it is no fault of mine that the " Indiana block," under the conditions of the test, did not produce as good results as anthracite coal does under its best conditions. Whether better results might be obtained under any other conditions, I don't know ; that would require numerous tests to determine satisfactorily; but we do know what can be obtained under good and known conditions.

The high flue temperature is undoubtedly due to the rapid rate of combustion ; namely, 15 lbs. per hour per square foot of grate.

The grate area was originally considerably larger, the ratio of heating to grate surface being then 38.6 to 1, and the rate of combustion about 11.50 lbs. What the evaporation then was I do not know, but I am disposed to credit the engineer at the pumping station with the requisite judgment that he would have made the change only as an actual improvement.

* Author's Closure.

It may be instructive in this connection to place beside this test the results obtained by Mr. Geo. H. Barrus in not very different cases and recorded in his book on *Boiler Tests.* See table.

TABLE SHOWING THE PROPORTIONS OF BOILERS AND THE EVAPORATIVE EFFICIENCY OF DIFFERENT BITUMINOUS COALS.

Test Number.	Kind of Coal.	Ratio of heating to grate surface.	Ash, per cent.	Rate of combustion. Lbs. per hour per square foot.	Tem. of escap'g gases	Evaporation per pound of combustible from and at 212°.	Expert.
19	Bituminous Cumberland	29.4 to 1	8.7%	10.9 lbs.	580°	10.60 lbs.	Barrus
26	Nova Scotia Culm. Bituminous	39.8 to 1	9.4%	14.4 lbs.	528°	9.51 lbs.	"
34	Walston..............	32.2 to 1	7.3%	12.3 lbs.	572°	10.00 lbs.	"
39	Walston.............	22.3 to 1	7.6%	9.52 lbs.	445°	9.48 lbs.	"
37	Bituminous Ohio Lump	39.2 to 1	7.6%	10.9 lbs.	501°	9.85 lbs.	"
a	Bituminous Indiana Block	50 to 1	7.25%	15.0 lbs.	557°	10.05 lbs.	Nagle

Tests 34, 39, and *a* may be fairly compared. If high flue temperature be the basis for judging of the obtainable evaporation, why did test 39, with 572° flue temperature, give 10 lbs. of steam, when test 34, with flue temperature of 445°, gave only 9.48 lbs.? We should naturally expect a *better* result, but we actually obtained a *smaller*.

Tests 34 and *a* compare very favorably if we take the flue temperature as the criterion. In test 34, with a rate of combustion of 12.3 lbs., and ratio of heating to grate surface of 32.2 to 1, we obtain a temperatureof flue of 572°, and yet in test *a*, with a greater rate of combustion—namely, 15 lbs.—we obtain a flue temperature of only 557°. Of course the heating surface has been increased greatly, but comparing 34 and 39, we find that the heating surface is not so important a factor as we would naturally suppose.

It may be a fair conclusion that a high rate of combustion is essential for these types of bituminous coals. If the rate be low, as low as 9.52 lbs., as shown in test 39, we get a low fire temperature. If the fire temperature were as high under a low rate of combustion as with the higher, having in this case only about two-thirds as much heating surface as in test 34, we should the original plan have been neglected. In the main room there are now three engines; in a side room excrescence, built on the front, there are four horizontal "Davidson" engines to do the work of one; and in a very costly engine house on the aqueduct, several hundred feet south, two vertical compound "Worthingtons" are being completed. Engines Nos. 1 and 2, under the charge of a man who believed in crank centres and throttling, and cared little for duty, did their work quietly and regularly, with remarkably light repair accounts. In 1866, No. 1 ran 3,824 hours; engine repairs, $160.22; No. 2, 3,066 hours; repairs, $171.84.

In 1866, Brooklyn began an extravagant system of water expenditures in various directions, under which, during 30 years' supply, with a revenue of $26,645,902, the outlay has been for operation and maintenance, $9,543,000; interest, $16,698,000; original construction, $5,440,000; extensions, about $11,620,000; or about $43,301,000.

In 1866, a full coat of ashes for the boilers was replaced by an expensive felt cover. In 1867, the 15-inch beam pin was enlarged to 20, and a new pump head made for No. 1. In 1868, an 80-inch cylinder was ordered as a substitute for No. 1, 90 inch, and a new set of pumps at $27,000. In 1870, its alteration to a crank engine was decided. In 1871, three new boilers were set, though the other three were in use in 1885.

Of No. 2, the report for 1873 says: "With the exception of new brasses to the beam pillow blocks, and new valves to the pumps, no repairs are anticipated on this engine." In 1881, No. 2 ran 7,420 hours; in 1886, 7,198; 1889, 8,149. A continuous service of 30 years is shown by this engine of 1861.

These engines cost $138,000; No. 3, of 1869, $129,750; No. 4, 1883, with building and main, $127,398; net, $70,000; No. 5, 1888, 2 engines, 10,000,000 gallons each, $190,471; house very expensive.

No. 3, as tested, had 178.43 feet lift (No. 1, 170 feet), 7½% loss of action (No. 1, 1.69); could not be continuously run; credited with 683,872 ft. lbs.; duty of 1874, 590,876; repair bill, 1872, $5,725.

No. 4, lift, 179.21 feet; duty, by *capacity* of pumps, 657,561 ft. lbs.; 1886, 550,988. Fig. 27 shows the full elevation of No. 1 engine.

CCCCLXXI.*

THE VALUE OF A WATER POWER.

BY CHARLES T. MAIN, LAWRENCE, MASS.

(Member of the Society.)

In estimating the value of a water power, especially where such value is used as testimony for a plaintiff whose water power has been diminished or confiscated, it is a common custom for the person making such estimate to say that the value is represented by a sum of money which, when put at interest, would maintain a steam-plant of the same power in the same place.

For example, when a power of 100 H.P. has been taken by right of eminent domain, or by right given by an act of legislature, or by any other legitimate means, in estimating the value of such a power and its consequent loss to the owner, it is reasoned that taking into consideration the cost of fuel at that particular place, and other expenses of running, a 100 H.P. steam-plant would cost say \$50 per year per horse-power to run. $\$50 \times 100 = \$5,000$ per year for running the steam-plant. This, capitalized at $5\% =$ \$100,000, which is said to represent the value of the water power.

At first glance this reasoning may appear to be sound, but upon examination it will be seen that it has no foundation, and probably there is no set of conditions under which it would absolutely hold good.

A water power may be of more value for one kind of business than another, and its value is very largely determined by its location. But passing these by for the present, the value of a water power depends upon:

1. The quantity of water, the fall, and the uniformity of flow during the year and for a succession of years. This is an axiom. and we should be obliged to go no further than this to dispose of the method of estimating values as stated at the outset.

(*a*) The effect of the fall is to increase or decrease the cost of construction per horse-power. If the fall is low the cost per

* Presented at the New York Meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

CCCCLXX.*

TEST OF INDIANA BLOCK COAL AT THE CHICAGO WEST SIDE PUMPING STATION.

BY A. F. NAGLE, CHICAGO, ILL.

(Member of the Society.)

THE following test was made by the writer during the month of August, 1891, and the report of results presented to Mr. L. H. Clark, City Engineer of Chicago, for whom the test was made.

DESCRIPTION OF THE BOILER PLANT.

1. The boilers used during the test were what are known as the southerly batteries, consisting of three batteries of two boilers each. One battery is usually out of use for cleaning, the remaining two being large enough to run a pair of engines.

2. The boilers are of the ordinary return tubular type, exceptionally well-proportioned and well-set. The gases return over the tops of the boilers, and the steam-pipe is run back and forth in this flue chamber, thus superheating the steam slightly (14.14°). This, however, is all lost by the time the steam arrives at the engine, as is evinced by the fact that the steam-trap connected to the bottom of the steam-chest still blows some water.

3. There is a small flue heater for heating the feed-water. It is very small, and as it is not kept clean automatically, nor in fact can any cleaning be done except by the entire removal of the heater, it is of very little value. A small allowance, however, is made for it (9°).

4. There is also a "smoke consumer" consisting of air and steam jets above the fire, but it was not in use during the test.

5. The draught is regulated by hand dampers.

6. The condensed water from the steam-jacket usually drains directly back to the boilers, but during the test the steam-jacket was shut off entirely.

* Presented at the New York Meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

more it costs to haul coal to it, the more valuable would be the power. If there is raw material to be brought to the mill and finished product to be taken away, it is a self-evident fact that the nearer the railroad or seaport the mill can be located the more valuable the power which drives it. This reasoning can be carried to *reductio ad absurdum* by saying that a water power is more valuable in the wilds of Maine, where there is no railroad and consequently where fuel is expensive, than in Lawrence, Lowell, or Manchester.

3. The value depends largely upon the fact whether or not the social conditions are or can be made such as to cause good operatives to locate and remain in the place; upon the sanitary conditions; and sometimes even, in the case of a developed power, upon the management of municipal or town government. All of which cannot be estimated in dollars and cents, but which determine to a certain extent the profits or losses.

4. There is in almost every business need for steam for other purposes than power, if for no other purpose, in colder climates, than for warming the buildings in cold weather. This steam can usually be used after being exhausted from an engine, requiring the consumption of little or no more fuel than is required to produce steam for the engine alone.

The plant required for producing the steam is a necessity when water is used for power, and should be included in the cost of power-plant, and the expense of running included in the cost of producing power. This item may be so large as to make a positive loss by running the boiler plant for steam for heating and using water for power, over and above the cost of producing the power by a steam-plant and using the exhaust steam for heating purposes. This item then in itself is enough to overthrow the old method of estimating the value of a water power in such a broad and general way.

I think it has thus far been shown conclusively that the old process of reasoning is wrong. It is always easier to criticise and to say that a thing is wrong than to say how to make it right, and it is much easier to show why the old method is wrong than to lay down a rule which shall cover the ground in a fairly correct manner.

In order to show a method of estimating the value of a water power it is necessary to consider the power first as undeveloped, and then, if it be developed, in its developed state.

By an undeveloped power is meant a natural fall or rapids, which, by the building of a dam or canal, or both, and by putting in water wheels, may be made to furnish power, but which is in its natural condition, no labor having been expended upon it.

There are but few kinds of business which demand a particular and restricted location. For this reason it is obvious that in nearly every kind of business a location can be selected which will furnish the best returns for the money invested. With an undeveloped power there need be no feeling that a certain amount having been expended it is a total loss to locate elsewhere. There is nothing to bind a foreign concern to this particular undeveloped power. It has the range of at least a large section of the country from which to make a choice of location, and in case it is necessary to locate on a stream and advantageous to use water power, there will still remain a choice of location.

There are exceptions to this, in cases where the power can be used where the raw material abounds, and the finished product finds a market in the immediate vicinity.

The essential points which must be considered—as to whether an undeveloped power can be developed and used to a greater profit than any particular business or the general run of business could be conducted elsewhere with a different source of power—are as follows:

a. Quantity of water during a dry year.

b. Uniformity of flow during the year, considering the storage capacity, natural and artificial.

c. Head of fall.

d. Conditions which fix the expense of building dam and canal, and flowage of land.

e. Conditions which affect the cost of foundations for buildings.

f. Geological conditions which determine the permanency of the falls.

g. Freight charges for fuel, supplies, raw materials, and finished product.

h. How much low-pressure steam can be used for heating purposes, and whether exhaust steam can be used for those purposes.

i. Is water needed for other purposes than power, and in what quantities?

j. The social and sanitary conditions which make it possible to procure and keep good help.

k. The greater uniformity of speed with steam than with water power.

All the above items except the last two can be estimated approximately in money value.

The power which has the most value is one which has a flow during a dry year which is nearly constant, or which can be made so by storage basins, and which requires no augmentation from other sources. It seems to me to be fair, in determining the value of such a power, to say that if the business which can be conducted there can be conducted elsewhere, where fuel is cheaper, the cost of that water power can be compared with the cost of steam power at such places which are suitable for the transaction of such business.

For illustration, let us consider the constant portion of the power at Lawrence as undeveloped. There is an amount of power which can be depended upon nearly always of about 10,000 H.P.

There is no question but what the business which is located along the Merrimac in Lawrence, which is the very business for which the development was made, could be equally well carried on in some other location where fuel and transportation are cheaper than in Lawrence. Let us consider it located where coal can be obtained at $3.00 per ton.

The amount of heat required per horse-power would vary with different kinds of business, but taking it for an average plain cotton mill, there is an amount of steam required for heating and slashing which is equivalent to about 25% of steam exhausted from the high-pressure cylinder of a compound engine of the power required to run that mill, the steam to be taken from the receiver.

Supposing this power produced by steam with plants averaging 500 H.P. each.

The coal consumption per horse-power per hour for a compound engine is taken at $1\frac{3}{4}$ lbs. per hour, when no steam is taken from the receiver for heating purposes. The gross consumption when 25% is taken from the receiver is about 2.06 lbs.

75% of the steam is used as in a compound engine	at	1.75 lbs. =	1.81 lbs.
25% " " " " high-pressure	"	3.00 lbs. =	.75 "
			2.06 "

19. The ashes were weighed both wet and dry, and a fair judgment was that the total weighed was equivalent to about 85% dry, and this net amount is recorded.

20. The water record was kept by two persons, the lever being operated by one man at the instant the word was given by an observer. It is believed that no errors of readings occurred greater than ½ of 1% if they had been cumulative, but usually errors of this kind neutralize each other.

21. A record was kept of the revolutions and water pressure of the engines as a check upon the steam used, the product of these two figures corresponding closely to the steam used.

22. The usual and proper precautions were observed throughout to obtain reliable and accurate data, and no errors of any magnitude are believed to exist.

RESULT OF TEST.

23. The results of the test are given in the following table. Date of test, August 6 and 7, A.D. 1891.

Duration of test	24 hrs.
Total coal consumed	45,900 lbs.
Ashes	3,337 "
Percentage of ash	7.25
Coal per hour	1,912.50 lbs.
Coal per hour per square foot of grate	15 lbs.
Combustible per hour	1,773 "

TEMPERATURE.

Temperature of city water	70°
Temperature of hot well	120°
Temperature at filter-tank	164°
Temperature at weighing-tank	159°
Temperature at flue-heater (estimated)	150°
Temperature at mud-drum (boiler)	159°
Temperature of escaping gases	557°
Temperature of steam, amount of superheating	14.14°

WATER.

Water evaporated from feed	388,065 lbs.
Water evaporated per hour from feed	16,169 "
Water evaporated per hour per square foot of heating surface	2.53 lbs.
Water evaporated per hour from and at 212°	17,689 lbs.
Water evaporated (feed) per pound of coal	8.45 lbs.
Water evaporated (from and at 212°) per pound of coal	9.25 "
Water evaporated (from and at 212°) per pound of coal, allowing for superheat	9.31 "

said to be about 25% of the total amount used, but in winter months the consumption is at least one and one-half times the average consumption, or 37½%. It is therefore necessary to have a boiler-plant of about 37½%, the size of the ones previously considered with the steam-plant, costing about $20 × .375 = $7.50 per horse-power of power used if the source of power is water.

The expense of running this boiler-plant is as follows:

Fixed expenses at 12½%. $7.50 × .125 =	$0 94
2.06 × .25 = .515 lbs. coal per hour average consumption per horse-power. .515 × 10.25 × 308 = 1626 lbs. per year @ $4.50 per ton =	3 26
One man at $2.00 per day =	1 23
	$5 43

A plant of minimum size, used for heating buildings alone, would be about 20% of the size required for power. $20 × .20 = $4.00 per horse-power of power.

The fixed expenses would be $4 × .125 =	$0 50
Coal, 0.5 ton per horse-power per year @ $4.50 per ton.	2 25
Attendance, one man at $2.00 per day for 150 days.	.60
	$3 35

The effect of item *g* can be estimated approximately by knowing the difference in charges for freight between our proposed location on the river and elsewhere equally suitable in other respects.

In an ordinary plain cotton mill, of average numbers about 30, the weight for raw material brought to the mill would be about 40 tons per 1000 spindles. The outgoing freight would be about 30 tons of finished product and 5 tons of waste. Calling all other supplies, etc., 5 tons, we have a total weight moved of about 80 tons. Taking the power required as 20 H.P. per 1000 spindles, the amount of freight per horse-power would equal 80 ÷ 20 = 4.00 tons per year. If a saving of 50 cents per ton can be made by locating nearer the base of supplies and markets, the saving per horse-power would be $2.00 per year. This amount of saving should either be deducted from the cost of steam power, or added to the cost of water power in getting their comparative value.

There is another portion of item *g* which cannot be expressed in money, and that is the advantage of nearness to markets for sale of goods.

ana block coal—and that it gives a result which is rather low. The flue temperature of the gases is rather high, and there is no explanation given why this temperature of the gases is so high when the boiler evaporation is so low. I think the chief effect of the paper may be to induce people to ask questions why the result was so bad. Instead of low temperature of flue gases and high economy, which we ought to expect from the low rate of driving, we find high temperature of flue gases and low economy. This low result of 10.05 lbs. evaporation from and at 212° per pound of combustible is what ought to be explained. We know that with anthracite coal we can get 12 lbs., with Cumberland coal 12½ lbs., under the best circumstances, but not with such high temperature as this. But we do not know what can be expected from Indiana block coal under the best possible circumstances. Mr. Nagle's paper leaves the question as it has been. We do not know what Indiana block coal is capable of doing, and this paper does not tell us.

*Mr. A. F. Nagle.**—"Indiana block" is a coal as well known in the Chicago market as Cumberland is in Boston, or Lackawanna in New York, and as I knew of no authoritative record of the evaporative power of this coal, I thought the foregoing report might be of interest to our Western members, even if it had none to some of our Eastern.

As to the results, it is no fault of mine that the "Indiana block," under the conditions of the test, did not produce as good results as anthracite coal does under its best conditions. Whether better results might be obtained under any other conditions, I don't know; that would require numerous tests to determine satisfactorily; but we do know what can be obtained under good and known conditions.

The high flue temperature is undoubtedly due to the rapid rate of combustion; namely, 15 lbs. per hour per square foot of grate.

The grate area was originally considerably larger, the ratio of heating to grate surface being then 38.6 to 1, and the rate of combustion about 11.50 lbs. What the evaporation then was I do not know, but I am disposed to credit the engineer at the pumping station with the requisite judgment that he would have made the change only as an actual improvement.

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* Author's Closure.

dropping to a very small sum if all of the exhaust could be used.

In a plain cotton mill about 25% of the exhaust from the high-pressure cylinder of a compound engine can be used on an average throughout the year. Establishments with dye-houses, or where drying is done, can use more, and it is safe to say that if it should not be known what particular business is to be carried on at a location under consideration, an average of 25% of exhaust steam could be used.

For powers which are to be used in larger or smaller amounts than 500 H.P. the relative costs would change; but a similar method of arriving at such values could be used.

The value may be expressed algebraically as follows:

$$S_p = E_f + E_r$$
$$W_p = E'_f + E'_r + S + (F_w - F_s) - W \pm B$$
$$V_{wp} = (S_p - W_p) \div P.$$

S_p = cost of steam power; E_f = fixed expenses, and E_r running expenses for steam power, taking into consideration the amount of exhaust steam which can be used.

W_p = cost of water power; E'_f = fixed expenses; E'_r = running expenses; S, cost of producing low-pressure steam required; F_w = cost of freight at location of water power, and F_s = cost of freight at location of steam power; W = cost of replacing such water as is needed for other purposes than power; B = extra cost for building in one location over the other.

V_{wp} = value of water power, and P is the percentage of interest to be obtained from such an investment.

The above formulæ are approximately correct for the average of cases, but there will be exceptions, and each case requires special treatment.

An undeveloped power has value when W_p is less than S_p, or when $S_p - W_p$ becomes a minus quantity it has no value. When $S_p - W_p$ is a plus quantity the value of the undeveloped power is represented by a sum of money which at interest will pay the difference.

Let us now consider the value of a power which is variable and which cannot be depended upon throughout the year, and, by the same reasoning, the value of the variable portion which there is in connection with a constant power previously considered. The variable power has of course less value than that which is constant.

naturally expect a high flue temperature, while, on the contrary, we obtained a low flue temperature, and also a poorer evaporation. This test is surely not without interest to those who are studying the subject carefully.

I may add that this "Indiana block" coal is a cheap fuel, costing in the Chicago market only about $2.65 per ton, against $5.50 per ton of anthracite, in car-load lots.

manent power. But the cost should be modified as before for the difference in cost of freights, for any advantage to be derived for the use of water for other purposes than power, and for any other advantages or disadvantages attending the use of one or the other power. The figures would be different for different sizes of plants, but the methods remain the same.

Almost all small powers are variable and of low heads, and although the cost per horse-power for producing steam power is larger for smaller powers, yet the cost will be in most cases less than the cost of running the double plant.

The value of an undeveloped variable power is, therefore, usually nothing if its variation is great, unless it is to be supplemented by a steam-plant. It is of value then only when the cost per horse-power for the double plant is less than the cost of steam power under the same conditions as mentioned for a permanent power, and its value can be represented in the same manner as the value of a permanent power has been represented.

Let us now consider the value of a developed power on which money has been expended in the construction of dam or canal, or both, and on wheel-plant.

To determine the market value of such a power it will be necessary to consider the power by itself independent of the plant; that is, to determine first the value of the power as though it were undeveloped, and then to determine the value of the improvements. The sum of both will represent the value of the power as developed.

It might happen in some cases that the value, considered in the undeveloped state, would be a minus quantity, but that the value of the improvements more than offset that, thus making it of value in the developed state.

The cost of developing a power originally will not always represent the value of the improvements, except in so far as it relates to the character of the work done. Considering the work properly and substantially done, the value of that work immediately after completion may not be represented by its cost. A certain power may cost to develop twice as much as another of equal power, the difference in cost being due to difference in head or some other natural cause; but, all other things being equal, the one which cost double has no more value than the other, because it produces no more.

horse-power of plant will be very much more than that for a high head. The value of that power of low head cannot be as great as that for the high head, other things being equal, for the first cost of plant and the fixed expenses, such as interest, depreciation, repairs, taxation, and insurance, will be greater for the lower head per horse-power; so also will be the running expenses; and to get the same return for the money expended, as more money is required in the construction of the plant with a low head, less value can be placed on the power itself.

(*b*) The effect of variable flow upon the value is more difficult to estimate, and to determine at what point of variability the power becomes of no value.

I am firmly convinced that to-day there are a great many concerns located upon streams which are so variable as to require an auxiliary steam-plant of a size equal to the water-power plant, or nearly so, to which in the past such water power may have been a saving, but which now, if they could begin anew, could produce their power more cheaply from a single steam-plant than from the double plant.

It is true that fuel is saved, if steam is not required for other purposes than power, during such times as the engine is not run; but it is also true that as the engine is only to run for a portion of the time, it is probably deemed advisable to purchase a low cost steam-plant in order to reduce the fixed expenses, which means a larger consumption per hour than there would be with a better plant. At times also the engine will be underloaded, which is not conducive to economy. To the running expense must be added the cost of maintenance of a double plant, so that the cost is almost sure to be more than that of a single new efficient plant.

If the stream is variable and the water-power plant is the only source of power, which must stop for a portion of the time, it would be of very little value under such conditions except for a very limited range of business. No business, employing any amount of labor, carried on in such a way, could compete successfully with concerns which have a continuous run.

2. Other things being equal, the value of a water power depends very largely upon its location.

On the basis of figuring indicated in the beginning, if the value of the water power varied directly as the cost of fuel, then the farther away from a railroad the power is located, and the

For example let us suppose the cost of development to be increased above that already estimated, owing to a lower fall, to \$260 per horse-power of plant.

Fixed expenses per horse-power per year = \$260 × .09 =		\$23 40
Running " " " " " say —		3 00
		\$26 40

Suppose all other expenses as before, then

$$W_p = \$26.40 + 5.43 + 200 - 2.00 = \$31.83\,;$$

$$V_{wp} = (S_p - W_p) \div P = (21.80 - 31.83) \div .06 = - \$167.17\,;$$

or it would take \$167.17 each year at 6% interest to make up for the difference between the cost of water and steam power. With such a cost as this it would be folly to develop a power, but if it had been developed it then might be run until the time for renewal, and at that time it would again become valueless for development.

The value of the improvements would be, if new, a sum which would bring the fixed expenses down so that the total cost of power should not exceed that of steam power, when S_p would equal W_p. Then, for water power,

$$E_f = S_p - [E_r + S + (F_w - F_s) - W \pm B] \text{ or}$$

$$E_f = \$21.80 - (3.00 + 5.43 + 2.00 - 2.00) = \$21.80 - 8.43 = \$13.37.$$

In the items of fixed expenses, repairs at 1% and insurance and taxes at 1% vary with the cost.

\$260 × .02 = \$5.20 for repairs, insurance, and taxes.

\$13.37 − \$5.20 = \$8.17 represents 5% interest and 2% depreciation, which need not vary with the cost, but which would vary with the value or amount paid for the improvements, if purchased after the improvements were made.

\$8.17 ÷ 07 = \$116.71 represents the value of the plant per horse-power if new, or the amount of money which could be expended in purchasing this power and improvements for the purpose of running through the average lifetime of the plant, at the end of which time it would become again valueless. For 10,000 H.P. the value would be new \$1,167,100, although \$2,600,000 had been expended upon it. If the plant has been run, then the value is \$116.71 per horse-power, less depreciation.

With a variable power and a double plant it may, and prob-

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The essential points which must be considered—as to whether an undeveloped power can be developed and used to a greater profit than any particular business or the general run of business could be conducted elsewhere with a different source of power—are as follows:

a. Quantity of water during a dry year.

b. Uniformity of flow during the year, considering the storage capacity, natural and artificial.

c. Head of fall.

d. Conditions which fix the expense of building dam and canal, and flowage of land.

e. Conditions which affect the cost of foundations for buildings.

f. Geological conditions which determine the permanency of the falls.

g. Freight charges for fuel, supplies, raw materials, and finished product.

h. How much low-pressure steam can be used for heating purposes, and whether exhaust steam can be used for those purposes.

i. Is water needed for other purposes than power, and in what quantities?

of its effect upon the value of the entire manufacturing plant, and the necessity of providing a power for that which has been taken away.

Is the damage in such a case equal to a sum of money which when put at interest would furnish a substitute power?

The nearest approach to a case in which the damage would be such an amount is when a portion of a permanent and developed power is taken away, leaving a portion to the owner.

If so small a portion is taken that no part of the water-plant can be dispensed with, then the fixed and the running expense of such plant continue. In nearly every case, when a portion is taken, there will be a boiler-plant perhaps large enough to produce the amount of power which it becomes necessary to replace. The fixed expense of such plant has to be borne, whether steam or water power is used. Any portion of a steam-plant which it is necessary to maintain and run when water power is used, can be fairly made to offset a portion of the fixed expense in the cost of steam power, if such plant can be made a part of the substitute plant.

If low-pressure steam is required for heating purposes and exhaust steam can be used, the heat thus saved should be credited to the cost of coal for running the substitute plant, and the cost of attendance should be proportioned between cost of power and cost of heating.

In figuring the cost of such a replacement of power any recognized improved methods should be figured on, even if such methods are not in use at the particular place under consideration.

In almost every power which we have considered as constant there are fluctuations, and the taking of a portion of the water, when there is so much water as to cause back-water, would be a benefit rather than a detriment.

If all of the water is taken away then cease the repairs, depreciation, taxes and insurance, and running expense, and such fixed and running expenses will properly go towards maintaining a steam-plant.

If a portion of a variable power, which is already supplemented by steam, is taken away, then a portion of the steam-plant can be made a part of the substitute plant, and if the steam-plant is of necessity of sufficient size to drive the work, then all of the fixed expense, and a portion of the running expense, should go to diminish the cost of replacing the water power.

In case all of the water power were taken away the fixed and running expenses cease, and, as before, can be applied to diminish the cost of steam power.

From what has been said I think it has been shown that it would be difficult to find a case in which such a broad statement of the value of a water power as has been made at the outset would hold good.

DISCUSSION.

Mr. Charles E. Emery.—The question discussed involves many changes of conditions which it is difficult to consider except when applied to a particular case. From such examination as I have been able to make I feel assured that the paper is very valuable, and I am very much obliged to the author for presenting it. It will be chiefly of service for reference in connection with any work that may arise of that character. There is one case in relation to condemnation of water-power which is not mentioned and which, in a prominent suit in Massachusetts, formed a very important factor. It was one of the Fall River cases which has become a classic reference on the general subject. The first condemnation for the use of the city was only about 5 per cent. of the current flow in the stream. As the flow was variable the several mills already had engines of large power for use during droughts, and it was urged by those of us who represented the city, that the steam-plant already in place should furnish the slight additional amount of power required when the water was low, and that as a basis for damages the city should reimburse the owner for the cost of that proportion of the steam-plant required to supply the deficiency caused by the diversion of the water, and that the city in addition should pay that proportion of the expenses. On the part of the mill owners it was urged that the damages should be based on the supposition that a small steam-plant, with engine-house, boiler, engine and engineer would be required to supply the deficiency, they thereby increasing the amount of damages very seriously. The Commission was a remarkable one. The county judge was chairman and Mr. James B. Francis, the noted hydraulic engineer, and Mr. Leavitt of this Society, were the other members. The Commission adopted the rule urged by the city in relation to such of the mills as used the water for power. There was, however, a certain quantity of water used in a print-works on the stream, and the representatives of these works urged that the only recompense which could be made them for the water diverted was the cost for which they could procure water to take its place, as their business required the water, and steam-power could not be substituted therefor. They, therefore, urged that their damages should equal the amount which it would cost them to replace the water at meter rates from the city mains. The Commission allowed the various mills using water for power $18,000, while the print-works, which was only entitled to one-eighth of the quantity condemned through one-seventh of the fall, and therefore had on a water-power basis an interest in only one-fifty-sixth of the whole, was allowed $10,000 for this small portion. In conversation afterwards some of the members of the Commission stated that they could not give a definite rule why so very much more was allowed in the second case, except that the conditions were entirely different. This incident shows the difficulty in making calculations for all the various conditions that arise.

Mr. Robert Cartwright.—I agree with Mr. Emery in relation to that thing, that no rule will apply for the cost of the water power. I have just put in at Rochester a 1,200 horse hydraulic plant for the Genesee River, and I tell you the honest truth when I say that parties are relinquishing their water rights upon the races and running their properties with steam power and saving money by selling their water rights and running by steam. Every case has to have its own individual figures. Just now we have got no water at all in the Genesee.

*Mr. Charles T. Main.**—The criticisms of both Mr. Emery and Mr. Cartwright are met in the paper.

The damage by the taking of water which is used for other purposes than power, is mentioned in several places in the paper.

The author had no idea of reducing the question to such a point as to be able to determine a fixed and unchangeable method of working out the value of a power, and has said in the paper that each case required special attention. I have simply endeavored in a general way to show that the methods commonly used in determining that value are wrong.

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* Author's Closure.

Item *i*. If water is required in large quantities for other purposes than power, for washing, etc., an estimate should be made of the cost of providing this away from the stream, and this amount deducted from the cost of water power or added to the cost of steam power. Let us assume that in the case under consideration the cost would be $2.00 per horse-power per year.

Considering all the other items of equal value in each case we should have the total costs as follows:

Steam power as given....................	$21 80
Water " , $18.70 + 5.43 + 2.00 − 2.00 =	19 13
Difference in favor of water power...	$2 67

$2.67 × 10,000 = $26,700.00 per year saving on 10,000 H.P.

Now it is fair to say that the value of this constant power is a sum of money which when put at interest will produce the saving; or if 6% is a fair interest to receive on money thus invested the value would be $26,700 ÷ .06 = $445,000.

I do not want it understood that this is my estimate of the value of that portion of the water power at Lawrence which is constant, for certain premises have been assumed for illustration simply without a knowledge of all the truth.

If there are no other considerations than power it will be profitable to develop a power so long as the conditions fixing the cost of construction and running expense do not bring the fixed and running expense so high that the total cost per horse-power for water power will be equal that of steam power. When that point is exceeded it would be folly to develop the water power.

The cost per horse-power for the dam would not increase in the inverse ratio as the head, but the cost of canal and that portion of plant from canal to river would increase very nearly in that ratio. So that if at 28 feet head the cost of plant is $130 per horse-power, at 10 feet head the cost would be nearly 2.8 times as much, or say $350. The fixed expense in this would be $350 × .09 = $31.50, and running expense say $5.00, making a total of $36.50 per horse-power per year, which is far beyond our cost given for steam power.

The other conditions affecting the cost and running expense would work for or against the development.

In almost every concern there is use for low-pressure steam to a greater or less extent. If exhaust steam can be used for these purposes the net cost of steam power is reduced, the cost sibly so situated between high and precipitous banks as to render it impossible to locate our manufacturing plant there, or to distribute the power furnished by a steam-plant in the immediate vicinity, which may, however, be so isolated as to reduce fire risks to a minimum while taking advantage of the most economical situation for coal and water supply. Let us assume, therefore, that we are provided with an electric current supply sufficient in quantity to operate whatever apparatus we may wish to connect, and of sufficiently low pressure or "voltage" to allay any fears of personal injury on the part of employees.

The "main shafting" naturally demands our first attention, and if the buildings are of more than one story, the advisability of dividing the plant into sections, each one of which may be started and stopped independently of any other, and without belt towers or belt holes (those efficient dust and fire conveyers), is evident to the most superficial observer; while in a single-story building such sub-division reduces to a minimum the friction of lines of shafting, run most of the time, it may be, for the operation of only two or three machines. Engineers will appreciate fully the tales the indicator will tell, in nine out of ten of our larger manufacturing plants, of the power devoted to revolving the immense "main shaft" and in transmitting power from one line of shafting to another when not a single machine tool is in operation. While it is not advisable to run any but the largest tools by separate motors, it will usually be found possible to group machines in systems requiring from 10 H.P. upwards, and so arranged that the average of power required will not vary between wide limits, and such variation will not occur suddenly. We can then connect with the shafting for each group a properly designed motor, which may be started and stopped almost instantaneously, and, if desired, allowing of delicate adjustment in speed.

Fig. 29 shows such a motor operating various classes of apparatus and placed on the floor precisely as a small engine would be, but requiring no special foundations (owing to the absence of reciprocating motion and consequent pound), and belted to the shafting in the ordinary way; while Fig. 30 shows a motor of 15 H.P. capacity, connected directly to the shafting, with the controlling apparatus placed within easy reach of the floor on wall or post, as may be most convenient. When the

We must here consider:

a. The maximum, minimum, and average quantity of water, and length of time when there is no water.

b. All the other items which entered into the value of a uniform power.

c. Necessity in nearly all cases for a supplementary steam-plant.

The quantities and falls at different times determine the power. If there is liability to excessive rises in the river, the head is diminished by back-water and the power reduced. If the quantity of water is reduced below its normal flow, the power is reduced and ceases altogether when the flow ceases. If the variation causes many days of complete or partial shut-downs, the power will be of no value for most kinds of business unless supplemented by steam power, and if the total lack of water is for many days, the steam-plant must be of the same power as the water power used.

We should then have the expense of maintenance of two plants, and the running of each a portion of the time.

Let us consider again a power of 500 H.P., or several plants of 500 H.P. each, with a head of 28 ft., costing about $130 per horse-power of water-plant. To this add $65, the cost of a steam-plant, thus making the total cost of plant $195 per horse-power.

It will make but little difference about the wear and tear whether the plants are run a portion or all of the time.

The fixed expenses are:

On water-power plant, $130 × .09 = ...	$11.70
On steam-power plant, 65 × .125 = ...	8.125
On both power plants..	$19.825
The running expense for water power is, say...........................	2.00
And for steam power, according to length of run, cost of coal, etc.......	
If coal is $4.50 per ton and the engine is run for two months with its proper load all the time, the cost of coal will equal for that time about	1.91
Attendance for boilers and engine for two months......................	.70
Having the boiler-plant already in, the fixed expenses do not have again to be added for heating purposes, but the running expense does.	
It is usually the case that low water occurs in the summer and fall when no heating is required of buildings. The least amount required for this purpose we have seen to be $2.25 for coal and .60 for attendance =	2.85
Total cost for power and heat	$27.285

The cost obtained as above should be compared with the cost of producing steam power elsewhere, as in the case of a permanent power. But the cost should be modified as before for the difference in cost of freights, for any advantage to be derived for the use of water for other purposes than power, and for any other advantages or disadvantages attending the use of one or the other power. The figures would be different for different sizes of plants, but the methods remain the same.

Almost all small powers are variable and of low heads, and although the cost per horse-power for producing steam power is larger for smaller powers, yet the cost will be in most cases less than the cost of running the double plant.

The value of an undeveloped variable power is, therefore, usually nothing if its variation is great, unless it is to be supplemented by a steam-plant. It is of value then only when the cost per horse-power for the double plant is less than the cost of steam power under the same conditions as mentioned for a permanent power, and its value can be represented in the same manner as the value of a permanent power has been represented.

Let us now consider the value of a developed power on which money has been expended in the construction of dam or canal, or both, and on wheel-plant.

To determine the market value of such a power it will be necessary to consider the power by itself independent of the plant; that is, to determine first the value of the power as though it were undeveloped, and then to determine the value of the improvements. The sum of both will represent the value of the power as developed.

It might happen in some cases that the value, considered in the undeveloped state, would be a minus quantity, but that the value of the improvements more than offset that, thus making it of value in the developed state.

The cost of developing a power originally will not always represent the value of the improvements, except in so far as it relates to the character of the work done. Considering the work properly and substantially done, the value of that work immediately after completion may not be represented by its cost. A certain power may cost to develop twice as much as another of equal power, the difference in cost being due to difference in head or some other natural cause; but, all other things being equal, the one which cost double has no more value than the other, because it produces no more.

The value would depend largely, however, upon the character of the work done and the condition of the dam, canal, and wheel-plant. If any portion required renewing soon, the value would be lessened; and if a general renewal of all the plant were necessary, the value would then be practically the same as though it were undeveloped.

In order to show a method for arriving at the value of the improvements, let us again go back to our original examples of 500 H.P. plants, where

$S_p = E_f + E_r = \$21.80$ = Cost of steam power per horse-power per year; and

$W_p = E'_f + E'_r + S + (F_w - F_s) - W \pm B = \$19.13$ = Cost of water power per horse-power per year; and

$V_{wp} = (S_p - W_p) \div P = 2.67 \div .06 = \$44.50$ = Value of undeveloped water power per horse-power.

In the above estimate of the cost of water power the cost of plant has been taken at $130 per horse-power, and its value now is represented by its cost, for we have shown that this outlay can be made, and that the fixed and running expenses upon this insures a saving over steam power, and that at the end of its natural average life, if some cheaper power has not been devised, its renewal will be justifiable.

The value, then, of the developed power and plant, if everything were new and in good condition, would be $44.50 + $130 = $174.50 per horse-power, or $1,745,000 for 10,000 H.P.

The actual value of a plant would depend upon the amount of depreciation which had taken place, or, better, upon the number of years which it would run without renewing. Thus, if we take the average depreciation upon the entire plant at 2% per year, or its average life as 50 years, and upon examination decide that it has an average useful life of 25 years to live, then the value of the plant would be one-half the cost, or $65 per horse-power, or the value of power and plant would be $44.50 + $65 = $109.50 per horse-power.

The value of the plant will be its cost, less depreciation, up to the point where the cost of water power equals that of steam power, for it would be justifiable to make an expenditure up to an amount which would give as good financial returns as any other source of power. Beyond this point, when water power costs more than steam power, the value of the improvements would not be represented by their cost.

comprising more than twenty separate motors operating at distances varying from 200 to 1,400 feet from the central engine

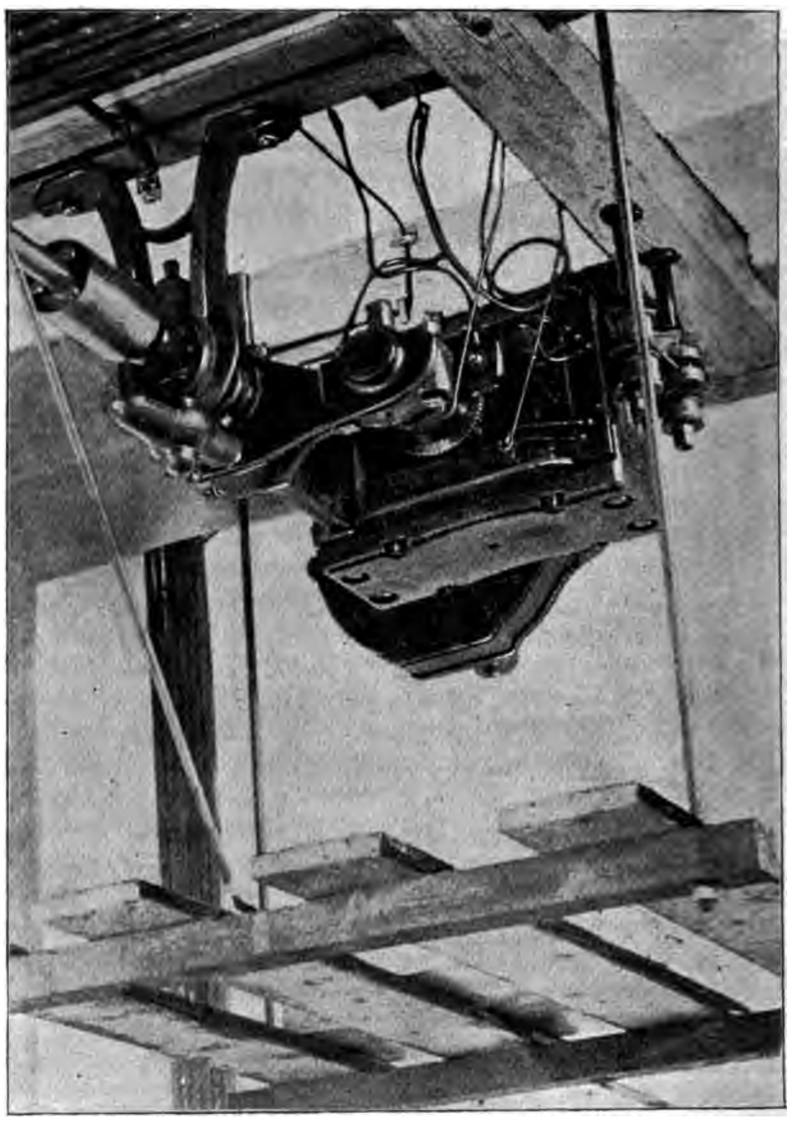


FIG. 30.

and generator. Another motor in the same factory and of the same size (15 H.P.), operates two 36-inch planers, two 32-inch ably would, be the case that it would be valueless in its undeveloped state, but that having the improvements, and these improvements being in good condition, it is then of value.

Take, for example, a case as given on page 149, where the cost of running the double plant is set at \$27.285 per horse-power per year. Take the cost of steam power as before, \$21.80, and there is a difference of \$5.485 in favor of steam power, so that the power by itself has no value.

The cost of developing this variable power with double plant was estimated at \$195 per horse-power. The value new will be such an amount as will bring the fixed expenses down so low that the cost of running the double plant will be no more than that of a single steam plant.

$$E_f = S_p - [E'_r + S + (F_w - F_s) - W \pm B] =$$
$$\$21.80 - (461 + 2.85 + 2.00 - 2.00) = 21.80 - 7.46 = \$14.34.$$

Repairs, insurance, and taxes @ 2% on water plant	= \$180 × .02 =	\$2 60
" " " " " 3% " steam "	= 65 × .03 =	1 95
		\$4 55

$$\$14.34 - \$4.55 = \$9.79;$$

which represents 7% for interest and depreciation on water-plant, and 9½% for same on steam-plant, or an average of 8.3% on both.

$$\$9.79 \div .083 = \$117.95,$$

which represents the value of the double plant per horse-power when new. After it has been in use it is worth \$117.95 per horse-power, less the depreciation.

The value of a developed power then is as follows: If the power can be run cheaper than steam, the value is that of the power, plus the cost of plant, less depreciation. If it cannot be run as cheaply as steam, considering its cost, etc., the value of the power itself is nothing, but the value of the plant is such a sum as could be paid for it new, which would bring the total cost of running down to the cost of steam power, less depreciation. That is, it is worth just what can be gotten out of the plant and no more.

If a portion or all of an undeveloped power is taken away from the owner, a fair compensation for the same would be its market value. But it might and probably would happen with a developed power, if a manufacturing plant had been established there and business carried on, that the damage would be more than the market value of the power or a portion thereof, because comprising more than twenty separate motors operating at distances varying from 200 to 1,400 feet from the central engine

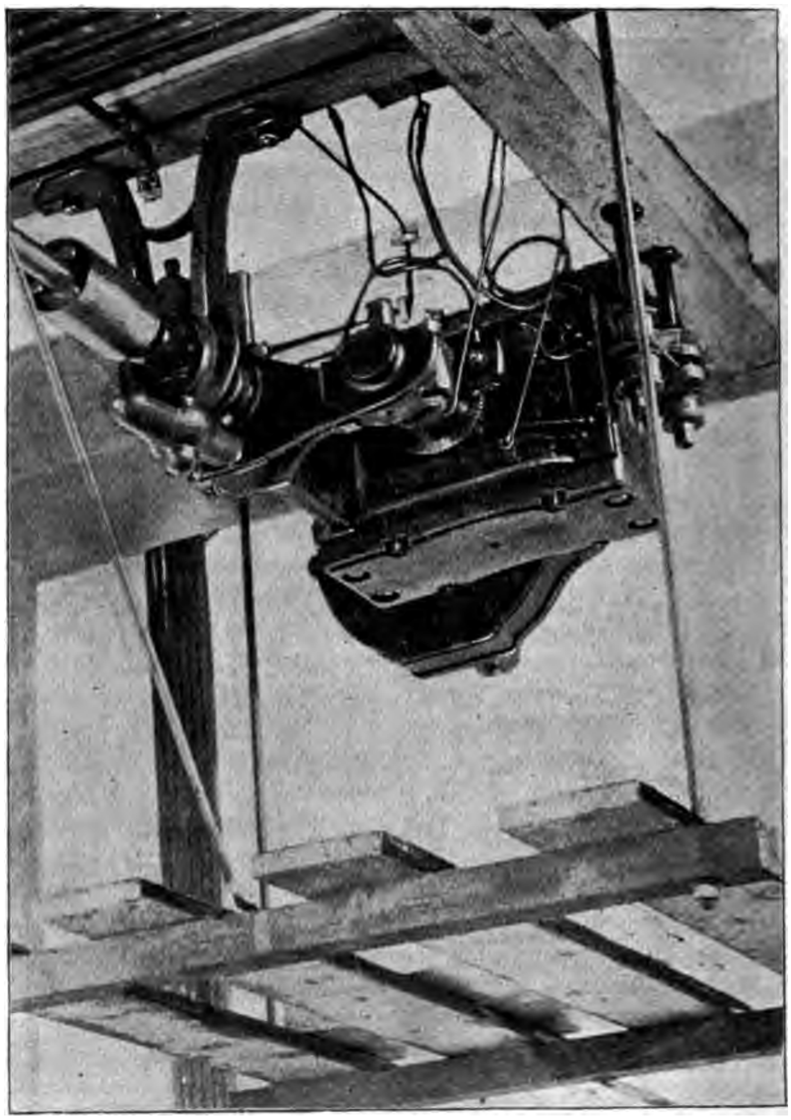


FIG. 30.

and generator. Another motor in the same factory and of the same size (15 H.P.), operates two 36-inch planers, two 32-inch planers, six 32-inch lathes, one 42-inch lathe, one 48-inch chucking machine, four 5-foot Prentice radial drills, and a large double grinder.

The application of motors to the operation of one or two

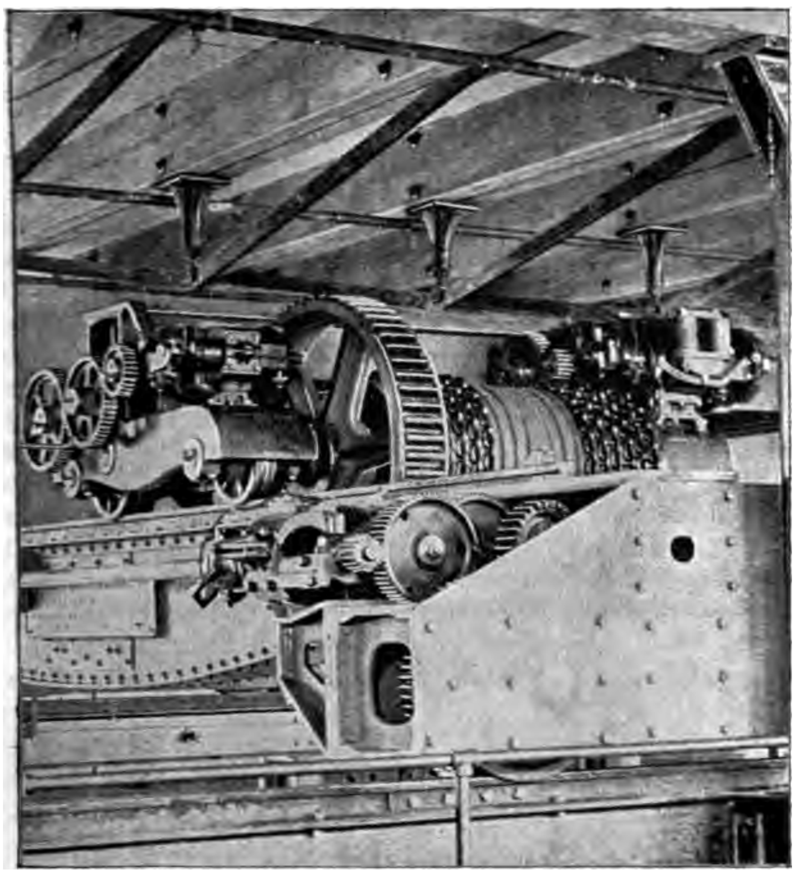


FIG. 31.

machines by means of countershafting is of course a small matter and calls for no special comment.

The increase during the past few years in the size of construction parts and machine tools has called for a corresponding increase in the power of devices for handling them, as demonstrated by the number of travelling cranes put in by the larger manufacturing plants. The friction losses in transmission by square shafts or other prevailing methods are excessive, and the possibility of operating the heaviest travelling cranes without mechanical connection between the crane trolley and the source of power, as provided by electrical means, appeals at once to the engineer, providing the losses in transforming from mechanical to electrical energy and back again are not excessive. Fig. 31 shows such a crane of thirty tons capacity, and some of its construction details. The following data concerning this piece of apparatus, which was built by the Morgan Engineering Co., and equipped with Thomson-Houston motors, may be of interest.

Three motors are used, one of 3 H.P. capacity for cross travel, and two of 10 H.P. capacity for hoisting and longitudinal travel respectively. The three movements may be obtained in any combination of direction and speed, as each motor is provided with a separate controller. The motors are built for 220 volts, and receive the current from two trolley wires running one on either side of the main floor, resting on insulators, a copper brush attached to either end of the bridge lifting the wire as it passes along, somewhat as in the method employed by cable cars in lifting their cables. The motors on the crane trolley receive their current through carbon brushes sliding along copper strips placed on the inside of the bridge. The total longitudinal travel of the crane is 165 feet; length of bridge between rails, 27 feet 2 inches. Total lifting space, 21½ feet. A "solenoid" brake is provided, which holds the hoisting gear automatically and instantaneously whenever the supply of current is shut off from the hoisting motor.

The extreme ease of operation, and the fact that the expenditure of power is almost exactly proportional to the work accomplished, and ceases absolutely when the crane is not in motion, are important features of this class of apparatus.

A simple and ingenious application of electro-magnetic principles is illustrated in Fig. 32, a load of almost five tons being held suspended by the attractive force of a magnet weighing only one-tenth of that amount. Such a device may often be used to great advantage in handling pig-iron, castings, boiler plates, etc., owing to the speed with which the operation of "picking up" and "letting go" may be accomplished. A simple "make and break" switch answers for a controlling device, and may be located either on the magnet itself or at any desired point. By a simple device, the quantity of current, and proportional

CCCCLXXII.*

ELECTRIC POWER DISTRIBUTION.

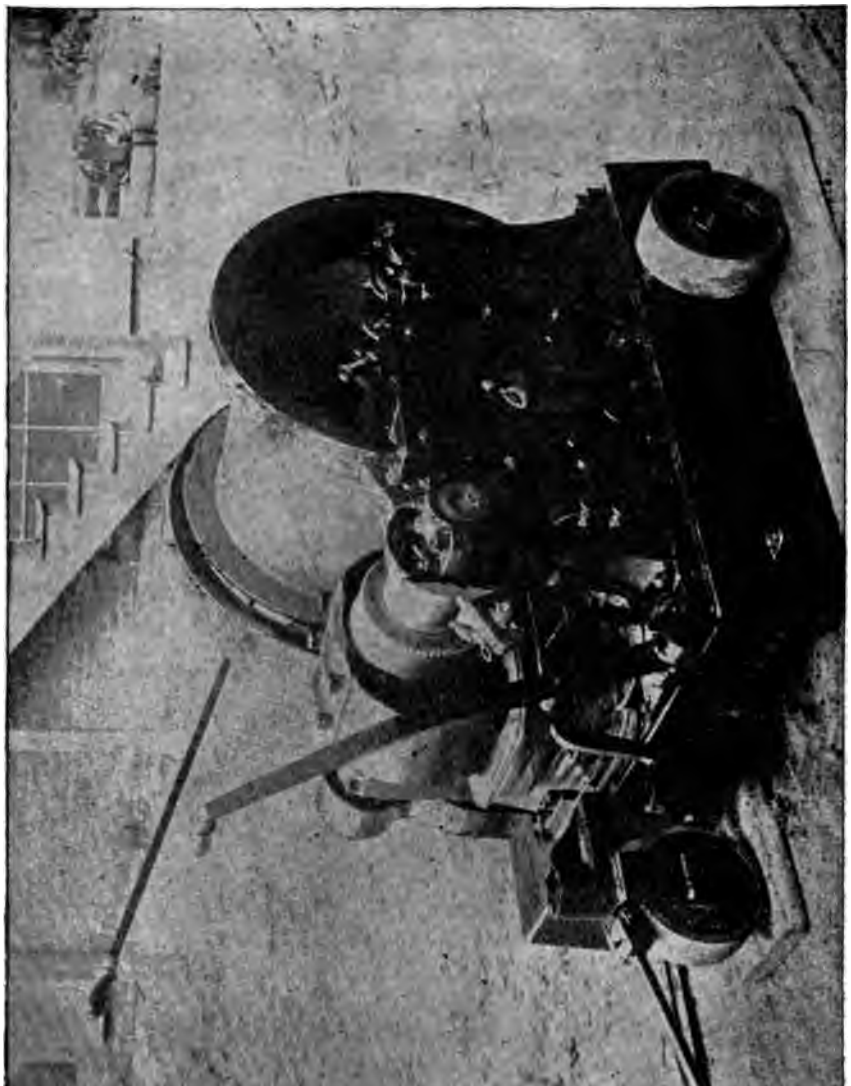
BY H. C. SPAULDING, BOSTON, MASS.
(Member of the Society.)

At a recent meeting of the Society a very lively discussion was brought out by a paper treating not of what electricity had done, but what it might do, in the line of passenger transportation, and the remarks of a number of members who took part in this discussion showed a deep interest in the development of electrical appliances as adapted to manufacturing processes.

The commercial possibility of transmitting power for comparatively long distances by electrical means, when the losses attendant upon such transmission by any other system would be prohibitory, is sufficiently well known and acknowledged at the present time to call for no extended comment. Curiously enough, our friends on the other side of the Atlantic are far in advance of us in the utilization of natural energy for lighting and power service, although the number of water-power installations is rapidly increasing in our Western States and Territories, where the high cost of fuel would naturally bring about such a result. In Switzerland alone there are already upwards of two hundred electric power stations utilizing water power for operating lights and motors over extended areas, while possibly the best-known illustrations of such service in our own country are at Spokane Falls and Aspen.

It is the purpose of this paper, however, to separate, so far as possible, the closely allied topics of power transmission and power distribution, electrically considered, and to treat the latter principally in the light of its adaptability to manufacturing and constructing operations. It is accordingly immaterial, so far as the scope of this article is concerned, whether we are to utilize a water privilege ten or fifteen miles away, and pos-

* Presented at the New York meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.



One of the most useful applications of electric motors is in connection with hoisting apparatus of various kinds, including passenger and freight elevators. A power hoist is often required in a storehouse or shipping room at some distance from line shafting is more than ten or twelve feet from the floor it will usually be found advisable to provide a light platform on which to stand when inspecting or oiling.

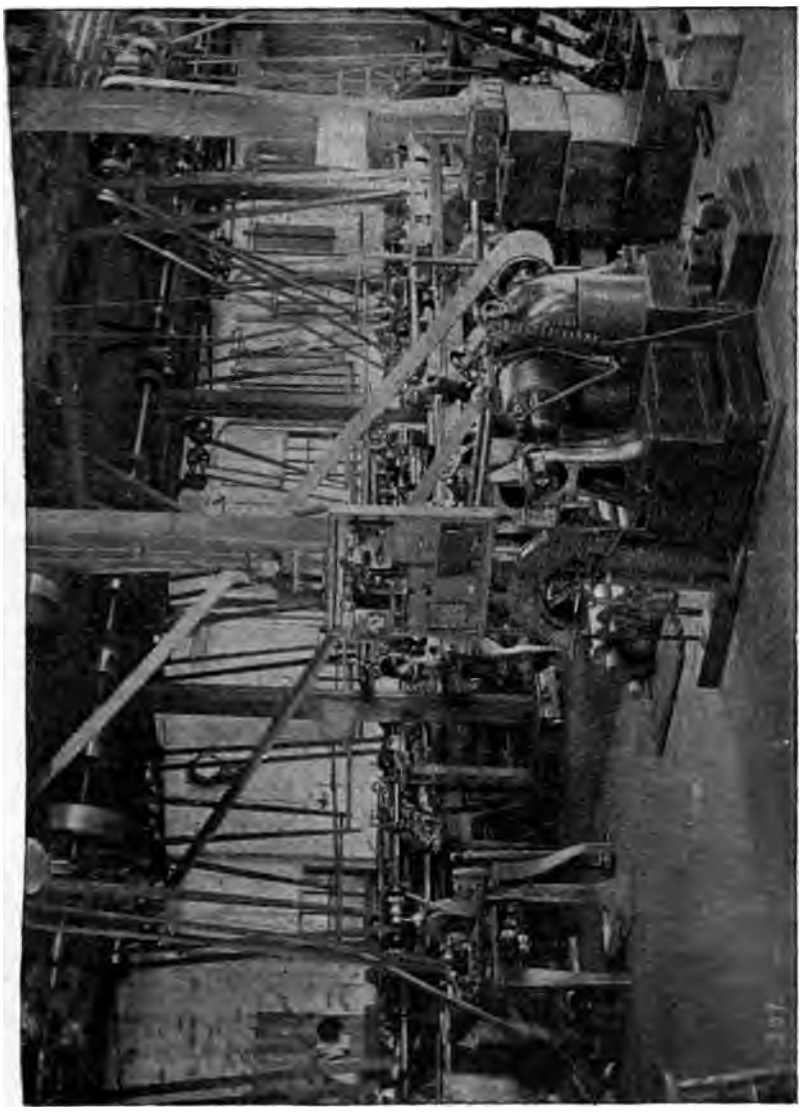


Fig. 29.

The motor shown in Fig. 30 operates two Pond planers of 84 inches and 72 inches respectively, one 5-foot Warren radial drill, also a large double grinder; and is one of a system care, and expense, and a maximum of convenience, economy, and safety. A portable hoist of 15 H.P. capacity, designed especially for dock work, is shown in Fig. 33; one of the same general type,

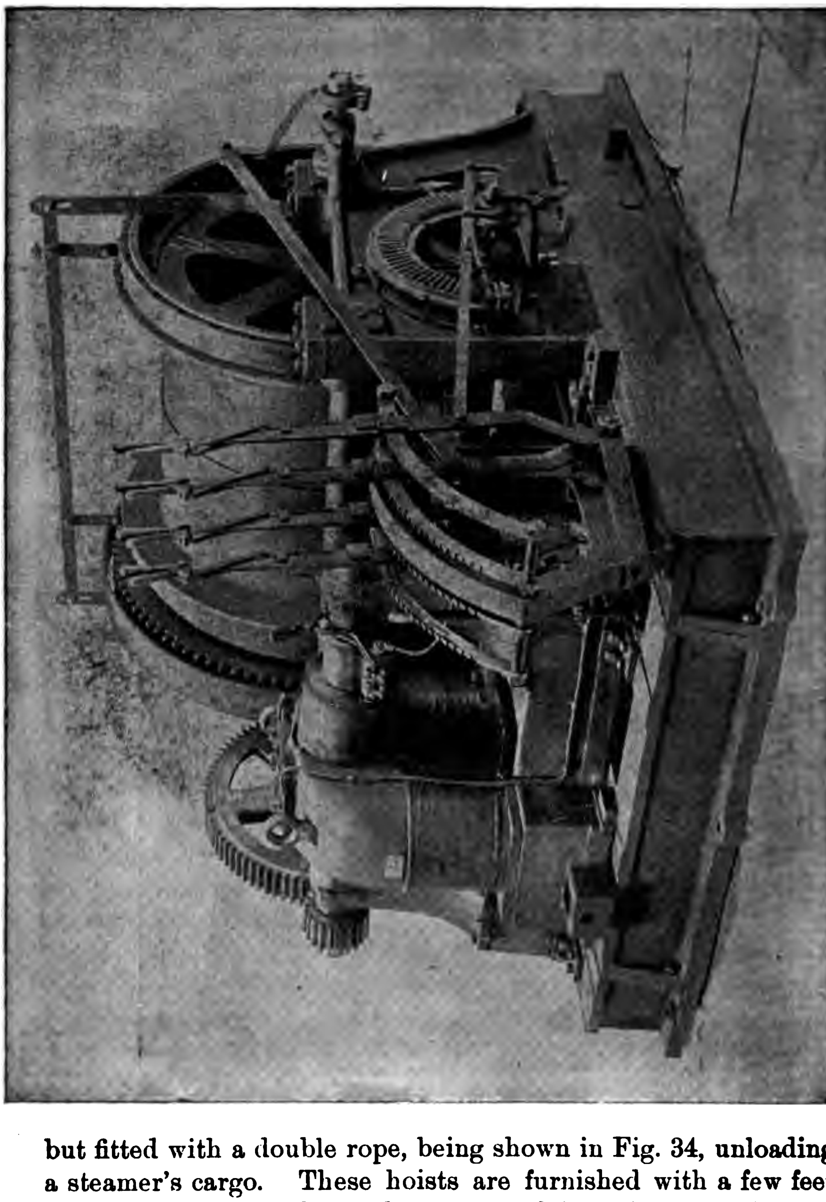


Fig. 35.

but fitted with a double rope, being shown in Fig. 34, unloading a steamer's cargo. These hoists are furnished with a few feet of flexible cable, and may be connected in a few seconds with "service plugs" located on various parts of the wharf, the current being taken either from the lighting wires on steamer or shore, or, better, from a special power circuit, if such be available. Hoists of this kind may be advantageously used for building operations, and are appreciated on account of the absence of smoke and noise.

Fig. 35 shows another hoist of the same general type, but of 35 H.P. capacity and designed for heavy work.

The earlier applications of electricity to elevator service were not specially successful, except for slow speeds, as it was not for some time considered feasible to start, stop, and reverse the motor itself, thus imposing the disadvantages inherent in "open and crossed belt" machines upon all electric elevator installations.

The only practical method of using motors for high-speed passenger work was for some time, therefore, in operating the pumps of hydraulic systems; but of late the development of quick-controlling and reversing apparatus, with special types of motors, has made possible results fully equal in smoothness of operation to the best hydraulic practice, while the running expense is usually considerably less. Designs are in use for a double-drum freight or slow passenger type of machine, in which the movements of the apparatus are controlled entirely by electric means, the armature shaft of the motor being run in either direction at will, and at any desired speed.

Fig. 36 represents an improved high-speed passenger machine now in operation in a large business block in Boston, the controlling mechanism being entirely in the car and operating through flexible cables. Automatic brakes are provided, which are thrown into operation by the shutting off of the current, while the usual safety devices against over-travel and breakage are present, with others of a purely electrical nature.

The enormous areas required by many of our industrial concerns for carrying on the various processes of manufacture to the best advantage necessitate new and improved methods for the transportation of fuel, supplies, and material in various stages of treatment from one part of the plant to another; and here again the invisible agent electricity offers its services in a safe, compact, and effective form.

While the economy of electric locomotive-truck service is naturally dependent upon local conditions and the amount of

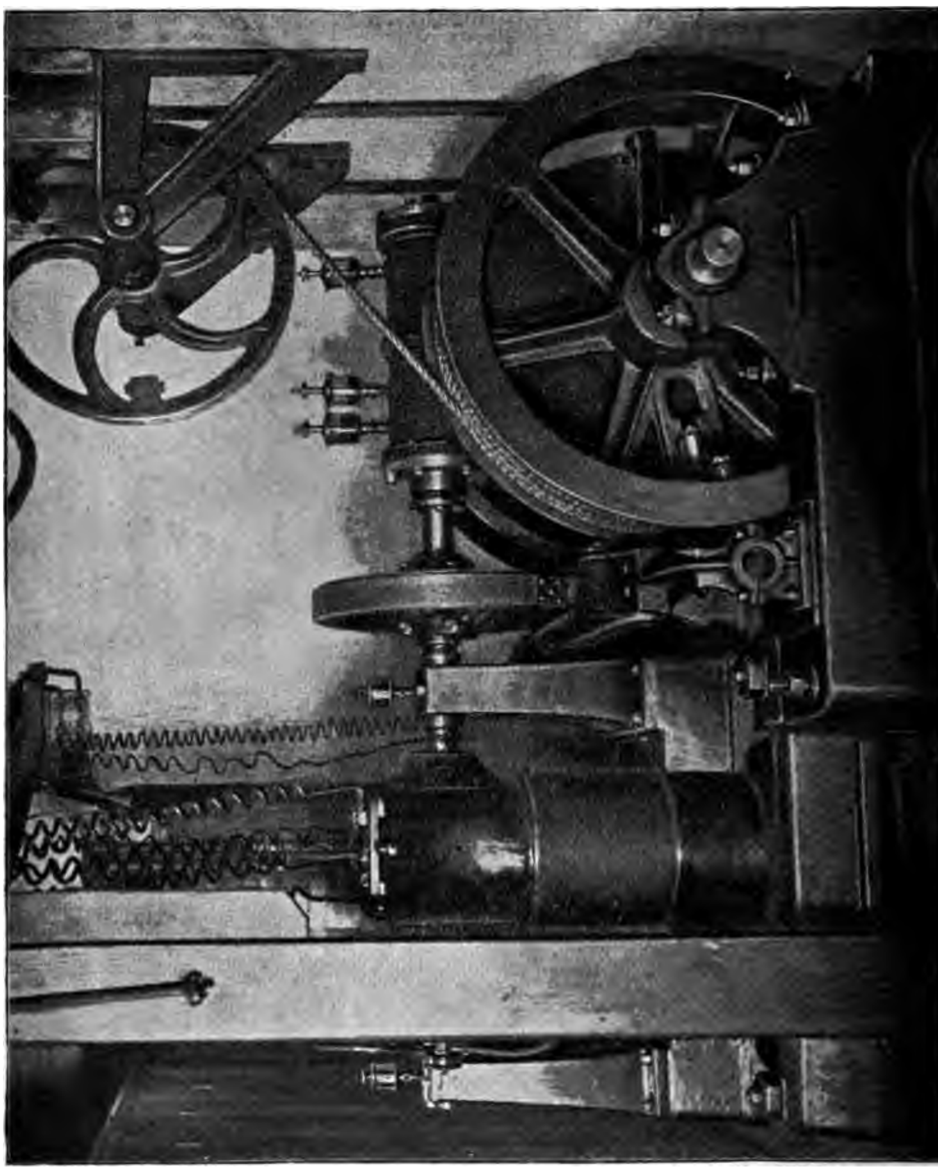


FIG. 36.

material to be carried, the saving over other methods has been entirely satisfactory to the score or more of corporations which have adopted it up to the present time. Fig. 37 shows an electric truck as applied to various lines of mill work, the motors being of 3 H.P. capacity, and using the regular lighting current,

while a larger size is shown in Fig. 38, suitable not only for general yard trucking, but having sufficient power to act as a

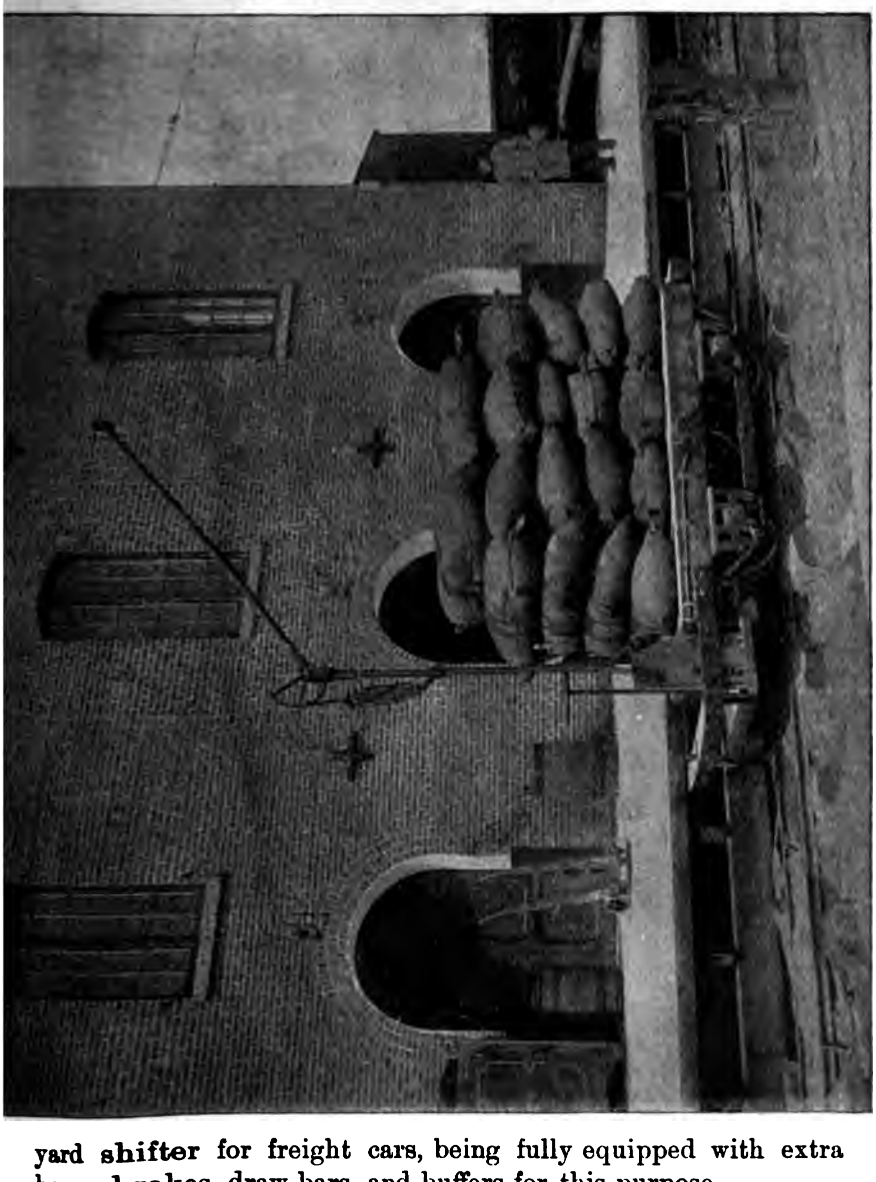
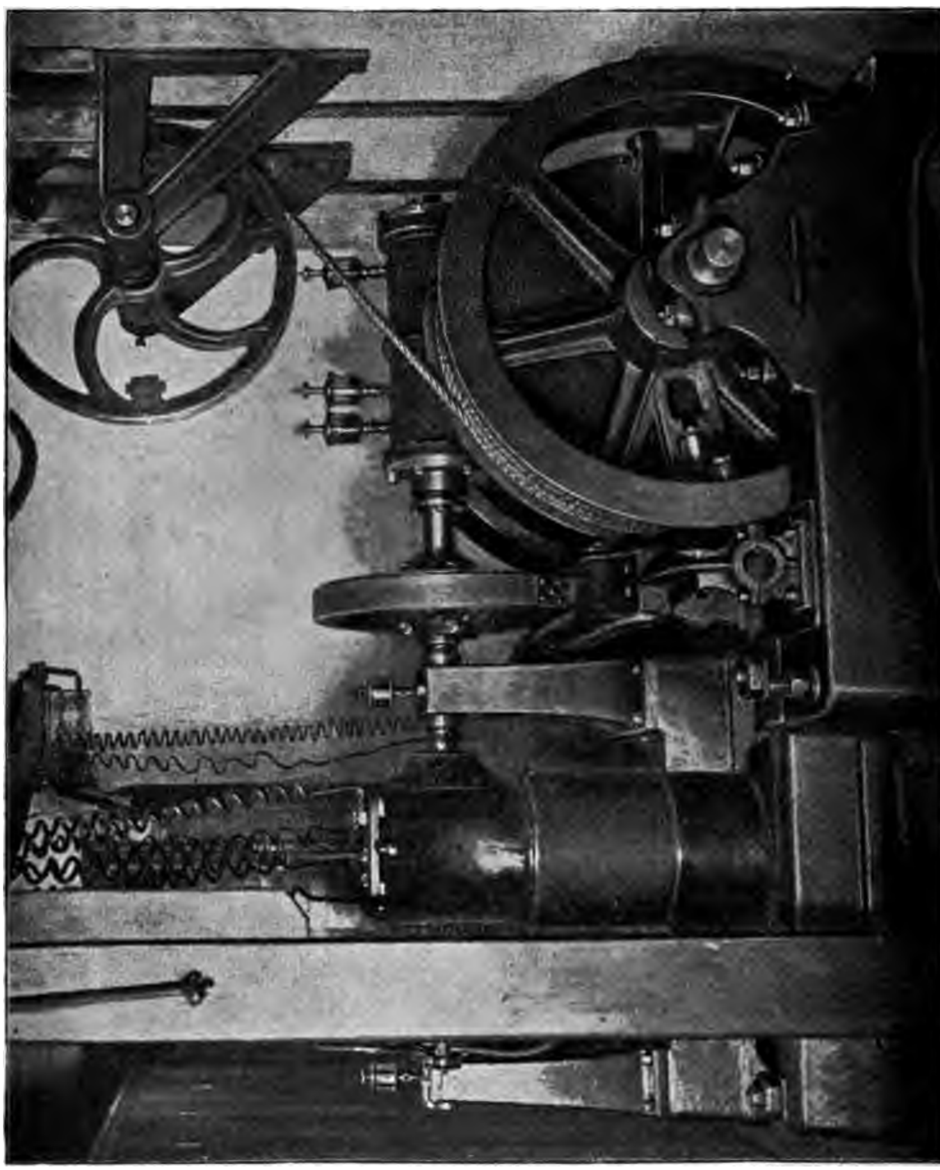


Fig. 37.

yard shifter for freight cars, being fully equipped with extra heavy brakes, draw-bars, and buffers for this purpose.

Among the numerous auxiliary uses to which electric apparatus may be put is that of pumping, either for boiler feed or general



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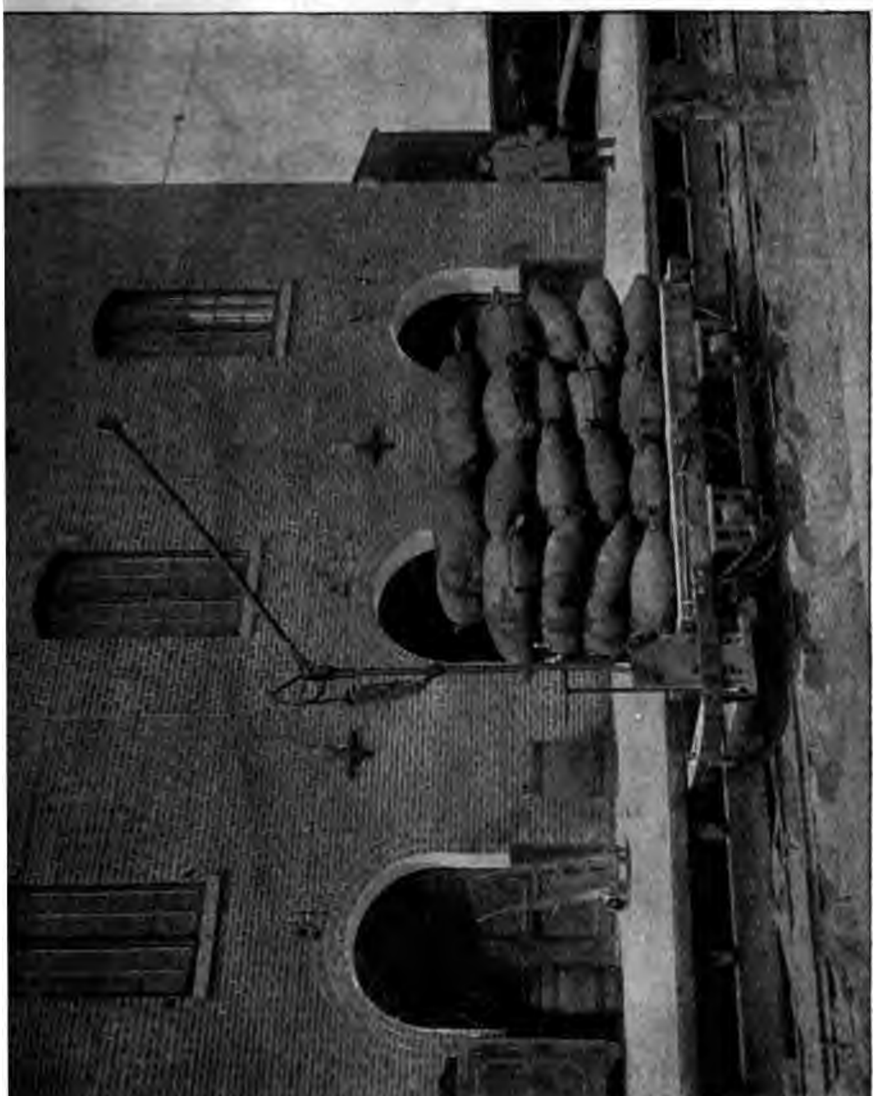

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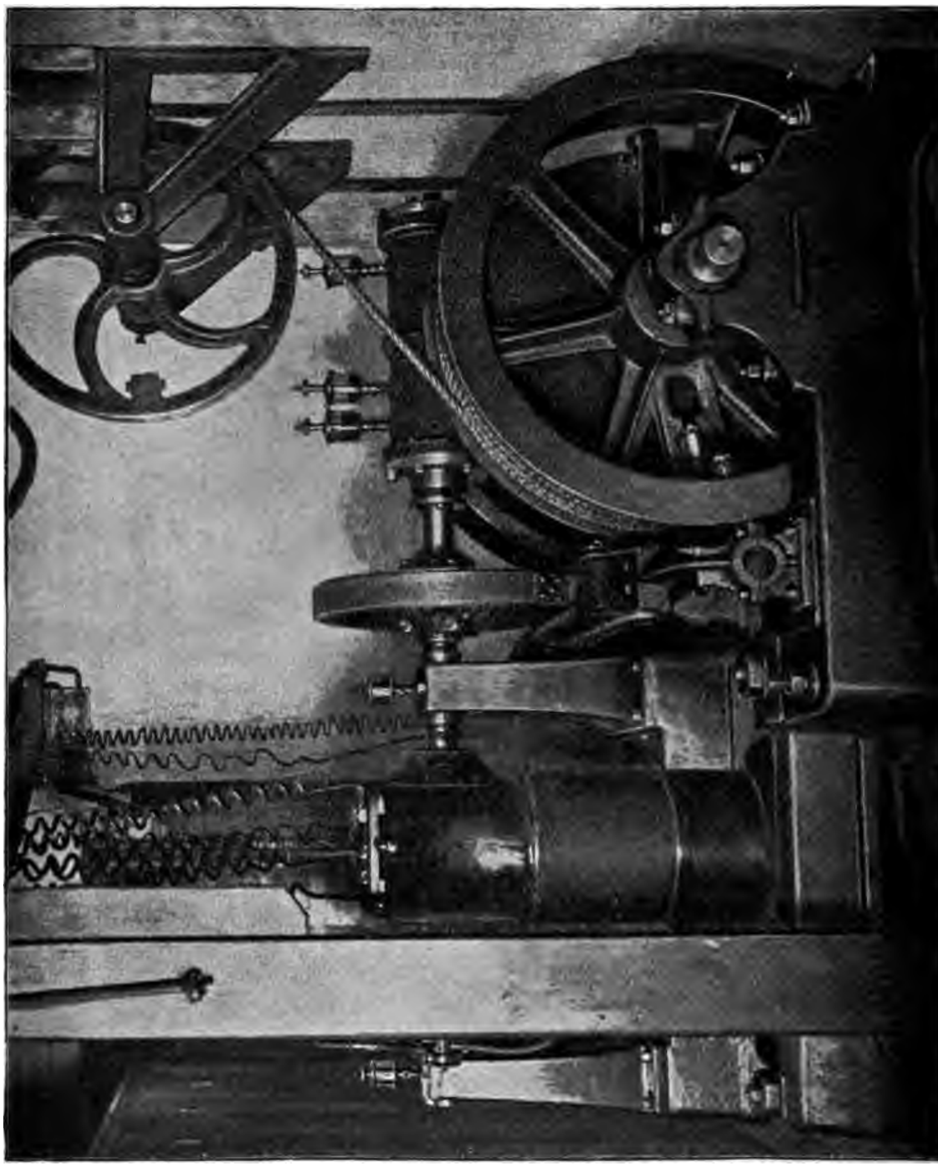

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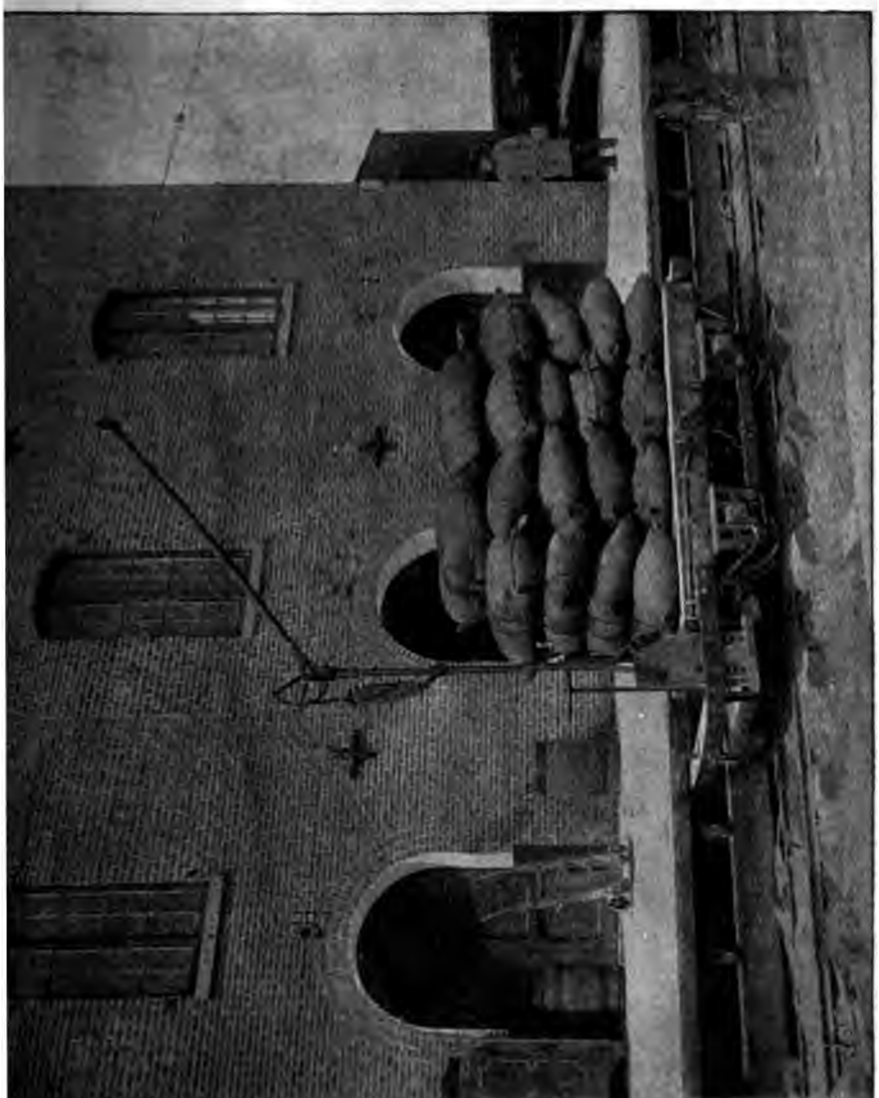

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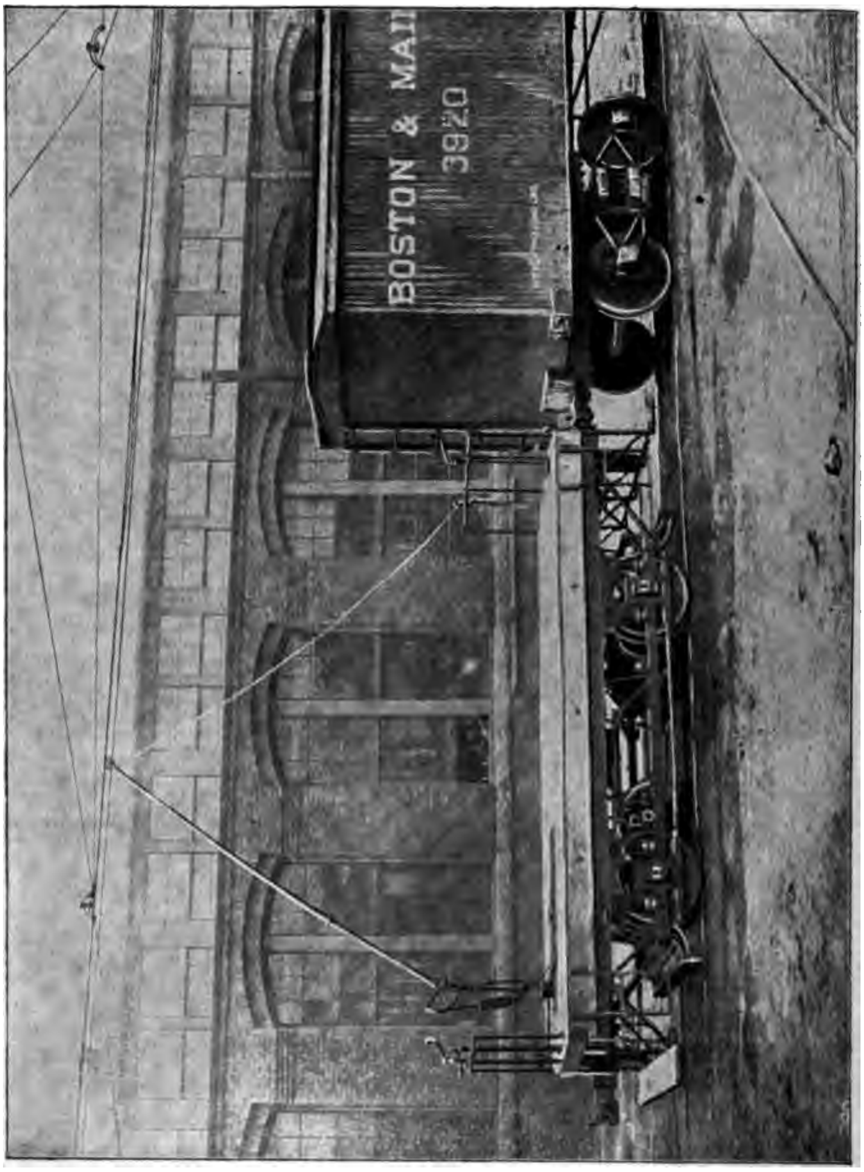

FIG. 88.

supply purposes. A number of automatic switches are now on the market, by means of which an electric pump may be automatically started or stopped by a change of water-level either in a gravity or pressure tank, a feature susceptible of numerous adaptations as suggested by circumstances.

Fig. 39 shows a form of pump specially adapted to general supply purposes, which is made in sizes of from 10 to 500 gallons per minute capacity.

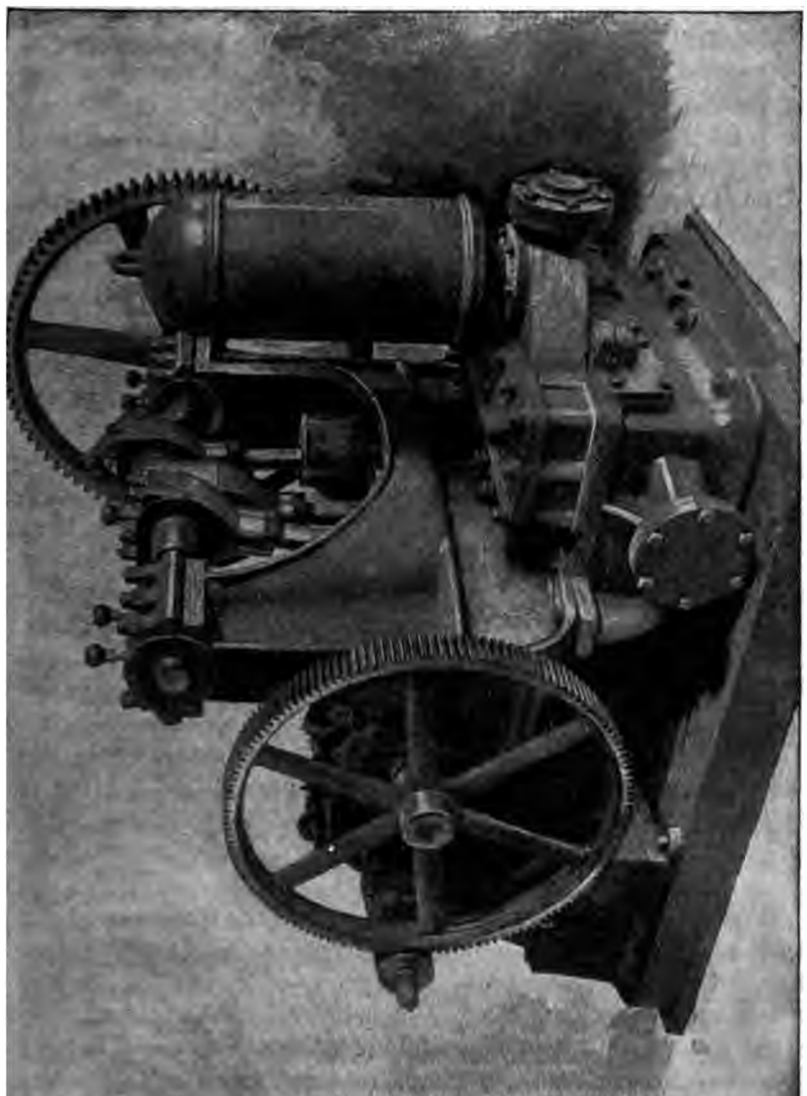

FIG. 39.

An objection sometimes offered to the adoption of electric apparatus is, that owing to the "youth of the art" the types of machines now built and on the market will in a few years be obsolete, like a coat outgrown though not outworn, so that the necessary investment is an unwise one on the part of the purchaser. This view seems to the writer not fully warranted, in the light of recent developments in electro-dynamics, since the efficiency of this class of machinery is now so high as to direct the ingenuity of electric engineers rather to the invention and improvement of auxiliary apparatus for controlling, measuring, and protecting the currents produced, than to the invention of new apparatus for their production.

The claim may now fairly be made also that the various "gauges" of the electric systems have been brought to standard, and that a piece of apparatus bought of any reliable company is adapted for use as well in conjunction with apparatus produced by another, and it may be competing, concern, provided the two systems are of the same general nature. This is just as necessary as that there should be one standard of measurement for engineering supplies of any kind, arbitrary, to be sure, but binding by courtesy throughout the engineering fraternity.

In conclusion, it is interesting to note the tendency to incorporate electric motors with various classes of machinery, thus forming complete mechanical units. It is to-day possible, for example, to equip a printing or publishing house equal in completeness to any now existing, without using a hanger, line shaft, or belt, each press being complete in itself as far as mechanical connection with the source of power is concerned, the entire transmission being accomplished by means of concealed wires.

The same is true of almost every line of industry, while the general application to naval practice is apparently a matter of the immediate future.

DISCUSSION.

Mr. Oberlin Smith.—I would like light on the question why it is not practical to use a motor for each machine, in a machine-shop, for instance, instead of for a group of four or five, as the author mentions. I have asked myself for a good while why we do not come to this system, and abolish line shafting and belting altogether in our shops, perhaps in many cases our counter-shafting also, because there is such extreme convenience with an electric motor in regard to instantly stopping, starting, reversing, and changing speed. I think there may be two reasons—there were three not long ago—why this has not taken place yet. One of them was that small motors were very inefficient, but the improvement in motors and generators of all kinds has been so remarkably rapid that the very small motors, say one horse-power and so forth, are almost as efficient as the large ones. At any rate the percentage of efficiency is so high that I have little question but that we can distribute power around an ordinary machine-shop to as good advantage as by the old way. Where we get generators of perhaps 90% efficiency, and motors of 70 to 90%, we will not have very much loss, at any rate not so much as is now due to long lines of shafting and belting. The "line loss" in cases of this kind may be disregarded, as conducting distances are so short. The great disadvantage of shafts and belts running all the time would be gotten rid of. With an electric motor we get our power exactly *when* we want it, and we get exactly *as much* as we want.

The second reason is purely conservatism—this is too *new* a thing for the majority of shop-owners. They may have already gone so far as to attach a motor to a main line of shafting, but there seems to be a holding back about getting rid of the line-shafting altogether.

There is a third and more practical reason why we do not do this thing, and that is the excessively high cost of motors. They now undoubtedly cost too much to equip each individual machine-tool with one, but this is simply because the industry is new. Our generators are reasonably cheap now, but the small motors do not seem to have gone down in proper proportion. The cry of the electricians, when appealed to on the subject, is that copper is an expensive article, and it is impossible to bring it down. In conversation on this subject with an eminent electric inventor, perhaps the leading electrician in this country, I got him to admit that enough copper could be bought for a one-horse motor for $5 or thereabouts. It would seem, therefore, that such cheapening is chiefly a question of labor; and I think one of the great fields for the mechanical engineer, during the remainder of this century, is to develop the small motor into proper shape to be manufactured just as are sewing machines and guns. Thus, taking present cost of copper and other materials, the labor could be so very much reduced, and these small motors thereby made so cheap that we could have them everywhere, and attach them to any machine we wanted to, stopping it and starting it whenever we chose by simply "touching the button." I wish to put this opinion on record, as outlining the *coming method* for machine-shops. It would, of course, require a good deal of nerve and faith and capital to manufacture and put upon the market so large a "batch" of these machines as really to make them cheap—for they must be made thousands at a time, with special tools. Moreover, the demand for them has not yet been created, and will not be until after they are cheapened.

Mr. S. J. McFarren.—I do not expect to add anything to the matter that Mr. Smith has given, except as to the use of the word "practicability." He says he does not see why it is not practicable to have a motor to each machine. It is entirely practicable; but, as I understand it, it is not yet economical, not only as he stated, from the price of the small motor, but from the fact that you cannot run two small motors with the same current with which you can run one motor to do the same work. This is perhaps only temporary, and may be removed by improvements now in progress.

If there are any electricians present, it must do them good to hear Mr. Smith talk in this convention. He made a remark which reminded me of something I heard in Philadelphia. I saw a motor of small power threading $\frac{3}{16}$ bolts and tapping nuts in hard rubber. I was told that this little machine had been in constant use, without any repairs excepting brushes, for two years and eight months. The number of reversals per minute when I was looking at it, with no appliance for steadying either the nut or bolt, which were both held in the operator's hands, was something like 10 or 12. I was told that it had made full time for two years and eight months, and it seemed as good then as the day it was made, and a better machine for the particular work it was doing than any possible combination machine that could be conceived of so far as I could see.

Mr. Smith.—I will say, Mr. President, that the difficulty formerly experienced in motors of all sorts by their copper brushes frequently getting out of order is in a great measure overcome now by the carbon brushes which are used almost everywhere, and which are becoming more and more popular. Thus the difficulty referred to by the last speaker, about the repairs of brushes, may not be so great after all.

It may be remembered that at one of our discussions, I think at the Cincinnati meeting, I spoke of the electric motor in connection with railroad use as being "simple as a grindstone or a coffee-mill." I have been somewhat criticised in the public prints for that expression. Many of the learned railroad men have spoken of the extreme complexity of a motor, compared with a pair of reversing engines, etc.; but I still adhere to my old doctrine, that in a machine which has only one revolting shaft in two journal-boxes, for its moving parts, we have the ideal mechanism for rotating driving wheels—especially as we can eliminate both shaft and boxes (considered as part of the motor) by utilizing the axle and bearings of the vehicle itself, which have to be there anyway. If practical difficulties remain in the way, let us overcome them. That's what we are for.

Mr. McFarren.—What Mr. Smith says about carbon brushes is the truth in street railway work. I doubt whether the development of the street railway electric systems could have reached the present point at all without the carbon brush. That reminds me of another illustration in the same business. I claim it is capable of easy demonstration that two motors—I know many people who differ on this—that two motors cannot be run on one car—that is, with a motor on each axle—with the same economy of current and the same mechanical economy that one motor can; and, further, that there are no real objections to the use of the one motor as compared with the two, except the difficulty in getting the entire weight for traction, and the contact of all four wheels instead of two wheels for adhesion.

Mr. Smith.—If we take a group of five machine-tools, and put five one-horse-power motors on them, there is not, of course, as high efficiency as there would be with one five-horse-power motor. Retail is not as good as wholesale, but, as a compensation, it must be remembered that these machines are frequently stopping and starting, and that the five-horse motor is running all the time, while perhaps only two or three machines will, on an average, be running steadily. Thus there is an element of waste which may be greater than in the other case, where each machine is using only the current needed for what it is doing, and is going only just when it is wanted. One moment, perhaps, it is using a horse-power, and the next, one-tenth of a horse-power. In such a way these little motors may probably be more economical than the big ones, because there is nothing but actual work drawing upon the current, and the shaft and belt friction is abolished.

CCCCLXXIII.*

INFLUENCE OF THE STEAM-JACKETS OF THE PAWTUCKET PUMPING ENGINE.

(Third Paper.)

BY WILLIAM KENT, NEW YORK CITY.

(Member of the Society.)

PAPERS with the same title as the above have been presented to the Society by Prof. Jas. E. Denton (*Trans.*, Vol. XI., page 328), and by Prof. D. S. Jacobus (*Trans.*, Vol. XI., page 1038). In these papers tests of the Pawtucket pumping engine have been described in great detail. Concerning the influence of the jackets upon the economy of the engine, the leading conclusion of Prof. Denton was that "the most that can be claimed for the jacket is that it probably caused no loss, and may possibly have caused a saving, not exceeding 3% of the total consumption." Prof. Jacobus's conclusions, made after a comparison of his own tests with those of Prof. Denton, and of Mr. John Walker, the superintendent of the water-works, are: "(1) The effect of jackets on the form of the expansion curves of both the high- and low-pressure cylinders is so small that the errors involved in the most accurate measurements of the indicator diagrams make it impossible to show any difference in the laws governing the same

"(2) The economy of the plant when working under the three sets of conditions is about 1.7% in favor of using the flue reheater, and 2.5% in favor of using both jackets and flue reheater, over that obtained when neither the flue reheater nor jackets are used."

As the results and conclusions in these two papers have been a surprise to many engineers, and as in their discussion some doubt was expressed upon the accuracy of the observations or of the apparatus used in the tests, Prof. Denton desired that

* Presented at the New York meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

further tests of the efficiency of the steam-jackets be made by an observer other than himself, and that in these tests all the possible precautions should be taken which had been suggested by the discussion of the papers above mentioned. At his request the writer consented to make a series of tests jointly with Prof. Jacobus, and the object of this paper is to place the results of these tests on record.

Tests were made by us on May 14, 15, and 16. The indicator diagrams were taken by Prof. Jacobus and the writer simultaneously, four indicators being used on the steam-engine, one on each end of each cylinder. When diagrams were taken from the water cylinders at the same time, they were taken by the engineer of the pumping station. All other observations were taken by ourselves. On Tuesday, May 19, Prof. Jacobus returned to Pawtucket and made another test under the same conditions as that of May 14.

Since the tests were made the original records have been in the hands of the writer, and he has calculated from them the average results shown in the appended tables.

These results show the effect of the steam-jacket on economy, within the limits of accuracy of the apparatus, of the observations, and of the measurements of the indicator diagrams. All the tests were made under uniform conditions as to steam pressure and work done, and the same indicators were used in the same positions in each test. Instrumental corrections will not therefore change the relative results, nor the percentage of gain made by the steam-jackets. As all of the instrumental errors are quite small, however, none of them exceeding 2%, and as those in the case of the indicators, which were accurately standardized, were found to balance each other, the error in the total horse-power developed, and in the economy of the engine, or steam consumption per horse-power per hour, due to constant instrumental errors, is probably not over 1%.

No attempt has been made to study the effect of the steam-jacket upon the law of the expansion curves in the two cylinders, nor upon cylinder condensation in the cylinders. As the original diagrams and records are preserved, and will be turned over to Prof. Jacobus, I trust that he will study these questions and present them to the Society in another paper.

Prof. Denton in his paper lays great stress on the errors of indicator diagrams and their measurement, so that he says the apparent saving of from 0.13 to 0.35 lb. of steam per horse-power per hour is within the limit of error to which the determination of indicated horse-power and cut-offs are subjected, so that he will not admit more than that the jackets only *possibly* caused a saving. It seems to the writer, however, that the variable errors in diagrams taken by several different indicators, and by different observers after proper corrections for known errors of the springs, should be expected to tend to balance each other, and that although the actual error of any single diagram may be 3%, yet if a series of diagrams during a long test shows an average result of from 0.13 to 0.35 lb. saving, or from less than 1 to less than 3% in favor of the steam-jackets, that result, small as it is, should be accepted as highly probable rather than merely possible, until other results overthrow it. Prof. Jacobus's tests, reported in his paper, also have the same tendency to show uniformly a very slight saving by the use of the jackets, and the tests now reported confirm the results both of Prof. Jacobus and Prof. Denton, in showing that the jackets do give a saving, but that the saving is very slight, ranging from about 1 to less than 4%.

The engine is described in Prof. Denton's paper, but a brief description may be repeated here. It is a horizontal cross compound engine, steam cylinders 15 and 30⅛ inches bore; water cylinders, 10.52 inches; stroke of all pistons, 30 inches; clearance, high-pressure cylinder, 4%; low, 3.7%. Diameter of rods, 2¾ inches. Ratio of volumes of cylinders, 4.085. Average cut-off in high-pressure cylinders, one-fourth, and in low, one-third. Jackets envelop the barrels but not the heads of both cylinders, and steam of full boiler pressure is used in each. The heads are not jacketed, but contain passages leading to and from the ports. The condensed steam from the jackets is pumped into the feed-pipe at a point between the boiler and hot well. The condensed steam collected in the receiver is continuously pumped through a heater placed in the chimney flue, and thence returned to the top of the receiver. Out of a total of about 155 lbs. thus circulated per hour, in actual work, one-third only is evaporated and returned to the receiver as steam; the other two-thirds gradually accumulate in the receiver and are blown to waste every three hours.

The conditions under which the tests of the four different days were made are as follows :

In the first test, on Thursday, May 14, the jackets on both cylinders were in use, as also was the reheater in the chimney, through which was passed the water that collected in the receiver.

To prove the absence of air and the thorough heating of the jackets by live steam, a thermometer, in a mercury well, was inserted into the upper part of each jacket. These thermometers registered uniformly 346° to 350° Fahr., or from 5° to 8° less than a thermometer in the steam pipe, or just about the temperature due to the pressure of the steam. Pet cocks were also let into the upper part of the jackets, and these were frequently opened to a very slight extent, when instantly steam appeared, there being not the slightest evidence of air at any time.

In order thoroughly to drain the condensed water from the jackets, pipes were led from the lowest points down to the cellar, beneath the engines. Water gauges were inserted in a horizontal extension of these pipes, and an assistant kept constantly at the exposed extension of these pipes, who opened a cock and discharged the accumulated water through a cooling coil into a pail whenever it reached a certain level in the water-gauges. This water was weighed from time to time, and it weighed 35 lbs. per hour for the high-pressure cylinder, and 77 lbs. per hour for the low-pressure cylinder, a total of 112 lbs. The object of cooling the water was to prevent loss by evaporation. This was accomplished by passing it through two Nason feed-water heaters, through the steam spaces of which cold water was circulated. We are indebted to the courtesy of Mr. C. W. Nason for the loan of these heaters. A water-gauge was also attached to the receiver, to indicate whether any water came back from the reheater unevaporated. The water collected in the receiver, above that which the reheater was capable of evaporating, was blown off three times during the day.

The engine was run under these conditions nearly five hours before the taking of records, begun at 10.45 A.M., so as to have everything heated to normal conditions.

The water fed to the boiler was weighed in a tank upon a platform scale, from which tank it was discharged from time to time into a second tank, from which it was then fed by a pump to the boiler. At the end of each half hour the water was brought to as nearly as possible the same level in this second tank, and the weight in the first tank recorded, corrections being apparent saving of from 0.13 to 0.35 lb. of steam per horse-power per hour is within the limit of error to which the determination of indicated horse-power and cut-offs are subjected, so that he will not admit more than that the jackets only *possibly* caused a saving. It seems to the writer, however, that the variable errors in diagrams taken by several different indicators, and by different observers after proper corrections for known errors of the springs, should be expected to tend to balance each other, and that although the actual error of any single diagram may be 3%, yet if a series of diagrams during a long test shows an average result of from 0.13 to 0.35 lb. saving, or from less than 1 to less than 3% in favor of the steam-jackets, that result, small as it is, should be accepted as highly probable rather than merely possible, until other results overthrow it. Prof. Jacobus's tests, reported in his paper, also have the same tendency to show uniformly a very slight saving by the use of the jackets, and the tests now reported confirm the results both of Prof. Jacobus and Prof. Denton, in showing that the jackets do give a saving, but that the saving is very slight, ranging from about 1 to less than 4%.

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On the second test, Friday, May 15, the jackets were not in use, and whatever water collected in the receiver passed into the low-pressure cylinder, as the reheater also was not in use. The jackets were opened at their lowest points to prevent any possible presence of water in them. The thermometers in the jackets registered 250° to 260° Fahr. in the high-pressure, and 196° to 198° in the low-pressure cylinders.

At the third test, Saturday, May 16, the conditions of the first test were repeated, except that the water condensed in the jackets was not caught and weighed, but was allowed to pass into the feed-pipe between the tank and the boilers, as in the ordinary way of running the plant. On this account an addition of 112 lbs. per hour, which was the amount of condensation determined on Thursday, was added to the tank record of water used.

On the fourth test, Tuesday, May 19, the conditions were in all respects the same as at the first test. The water condensed in the jackets was 36.2 lbs. in the high-pressure cylinder, and 75.2 lbs. in the low, or 111.4 lbs. total.

The following table gives the principal results of the tests:

	Jackets in use. May 14th.	Jackets off. 15th.	Jackets in use. 16th.	Jackets in use. 19th.
Engine-room temperature, deg. Fahr	83.3	83.0	83.5	80.3
" " barometer, inches	30.111	30.180	29.90	
Feed-water temperature, Fahr.	63.8	63.9	80.4	60.5
Boiler pressure, lbs. above atmosphere	123.5	123.5	123.1	123.0
Steam pressure in high-pressure jacket	121.5	0	121.1	121.0
Temperature of steam due to boiler pressure	351.9	351.9	351.7	351.6
" " " 2 feet from steam-chest	355.1	362.8	359.8	355.0
" " " high-pressure jacket	348.0	257	349	349.7
" " " low " "		197		
Receiver pressure, lbs. above atmosphere	9.1	9.2	10.0	8.8
Vacuum by gauge, inches	27.9	27.8	27.1	28.3
Revolutions per minute, avge	48.0	48.0	48.5	48.3
Horse-power, indicated "	142.90	140.78	139.95	141.91
Total water pressure, including suction, lbs., avge	119.37	117.51	115.96	118.74
Water condensed in jackets, h.p. per hr., lbs.	35	......	......	36.2
" " " " l.p. " " "	77	......	......	75.2
Steam consumption per hour, avge	1,956	2,007	1,965	1,980
" " H.P. per hour	13.687	14.256	14.041	13.952
Saving due to jackets, per cent	3.99	... ..	1.51	2.18

Saving due to jackets, average of 3 tests, 2.54 per cent.

Of the indicator diagrams herewith presented, Nos. 1 to 4 (Figs. 40 to 43) inclusive are selected from those taken on May 14 and May 15 to represent the average cards of these days, the mean effective pressures of these cards corresponding nearly to the average mean effective pressures of all the cards taken during these two days. For comparison, diagrams Nos. 5 and 6 (Figs. 44 and 45) are also given, from the cards of May 14, to show the maximum variation in the cards taken on one day when the jackets were in use. These cards show that the differences between the average cards of the two days which gave the greatest difference in economy, the jackets being used the first day and not used the second, are no greater than what appear to be the accidental differences between cards taken on the same day and under the same conditions. An inspection of the tables of mean effective pressures of all the cards taken will show large variations in mean effective pressures in cards taken near the same time, and when the work done, according to the water-pressure gauges, was the same. It is noticeable also that the outside end of the cylinder will sometimes show a greater mean pressure than the inside end,* and sometimes the reverse. The causes of these fluctuations in the energy developed in the cylinders are not apparent in all cases, but are partly due to the action of the governing device, which does not produce precisely

* By the inside end is meant the end of the steam cylinder nearest the pump cylinder, corresponding to the "crank end" in an ordinary engine.

made for the very slight variations in the water level in the second tank, and the height of the water in the three vertical tubular (Corliss) boilers was recorded. Corrections were made in the water record for variations in the water level. The apparent irregularity in the water record of water consumption per half hour is due, to some extent, to the fact that with the same amount of water in a vertical boiler the apparent water level may vary on account of differences in the condition of the fire. If records were taken at less frequent intervals, the apparent irregularity would decrease.

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Saving due to jackets, per cent	3.99	... ..	1.51	2.13

Saving due to jackets, average of 3 tests, 2.54 per cent.

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* By the inside end is meant the end of the steam cylinder nearest the pump cylinder, corresponding to the "crank end" in an ordinary engine.

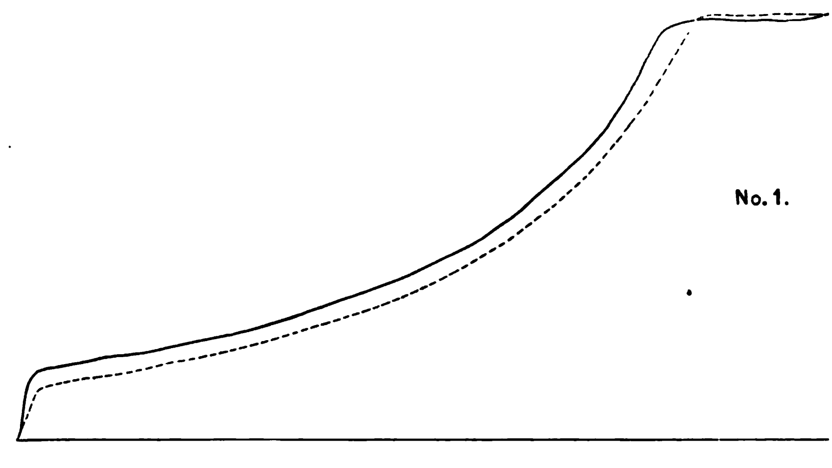

Fig. 40.

AVERAGE CARDS MAY 14 AND 15.

Full Line, 3.52 P.M., May 15, Inside End, M.E.P., 61.50. Jackets off.
Dotted Line, 3.35 P.M., May 14, Inside End, M.E.P., 58.75. Jackets on.

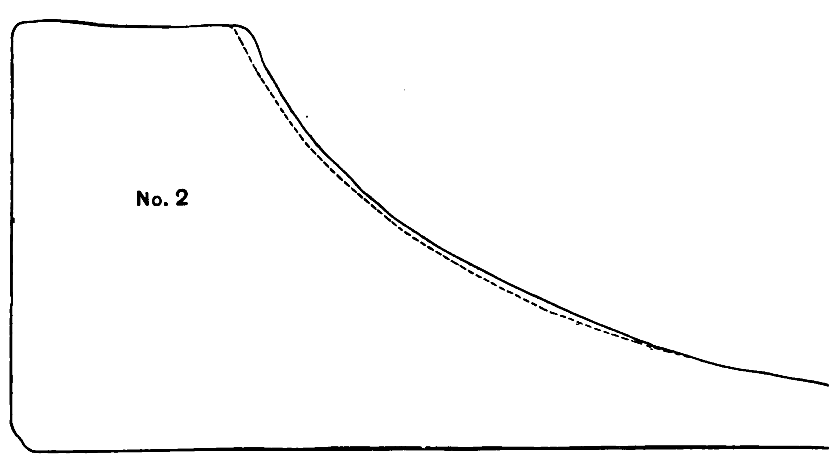

Fig. 41.

AVERAGE CARDS MAY 14 AND 15.

Full Line, 3.52 P.M., May 15, Outside End, M.E.P., 58.75. Jackets off.
Dotted Line, 3.35 P.M., May 14, Outside End, M.E.P., 57.50. Jackets on.

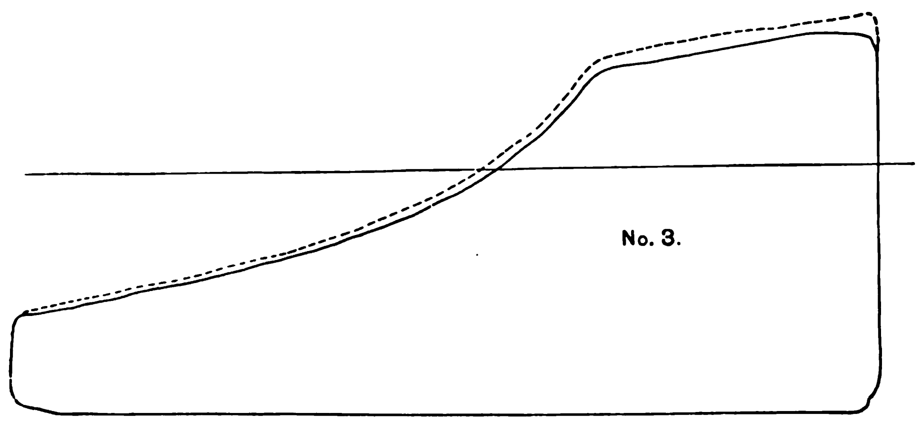

Fig. 42.

AVERAGE CARDS MAY 14 AND 15.

Full Line, 3.52 P.M., May 15, Inside End, M.E.P., 12.25. Jackets off.
Dotted Line, 3.35 P.M., May 14, Inside End, M.E.P., 12.65. Jackets on.

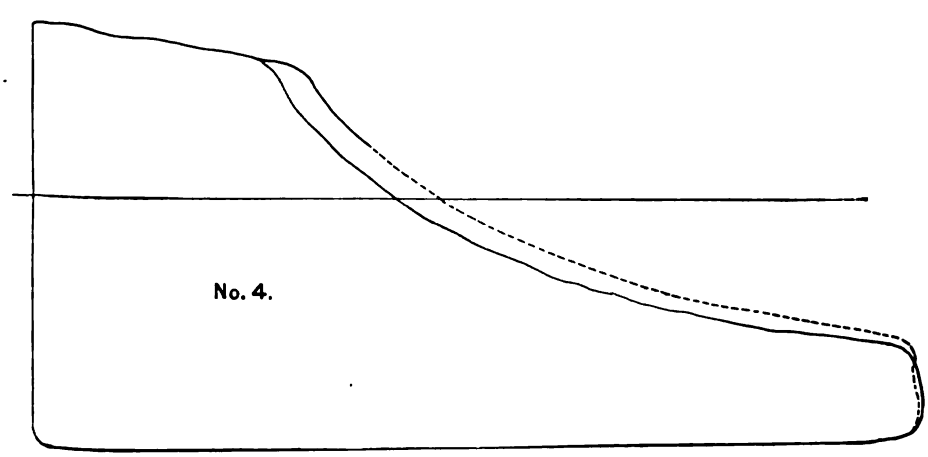

Fig. 43.

AVERAGE CARDS MAY 14 AND 15.

Full Line, 3.52 P.M., May 15, Outside End, M.E.P., 12.85. Jackets off.
Dotted Line, 3.35 P.M., May 14, Outside End, M.E.P., 13.75. Jackets on.

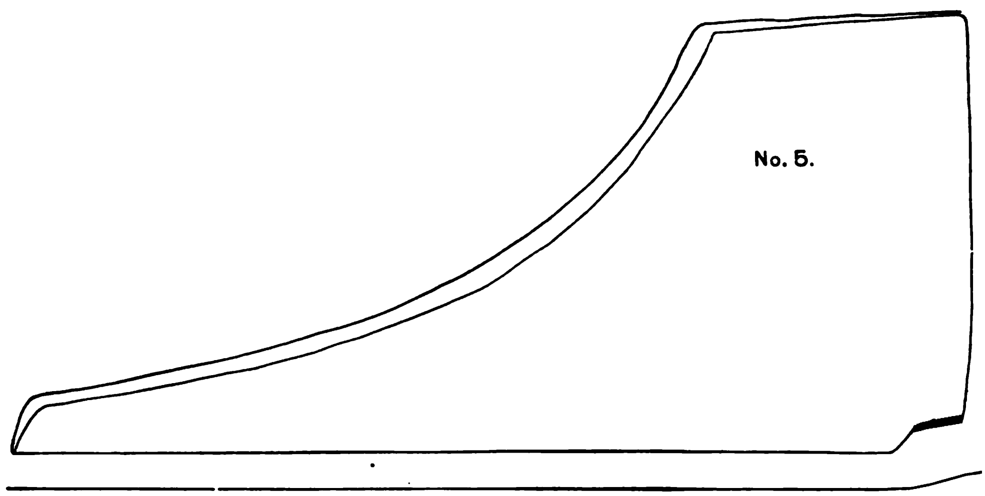

Fig. 44.

MAXIMUM AND MINIMUM CARDS MAY 14. INSIDE END.

Maximum, 12.35 P.M., M.E.P., 61.50.
Minimum, 5.02 P.M., M.E.P., 55.25.

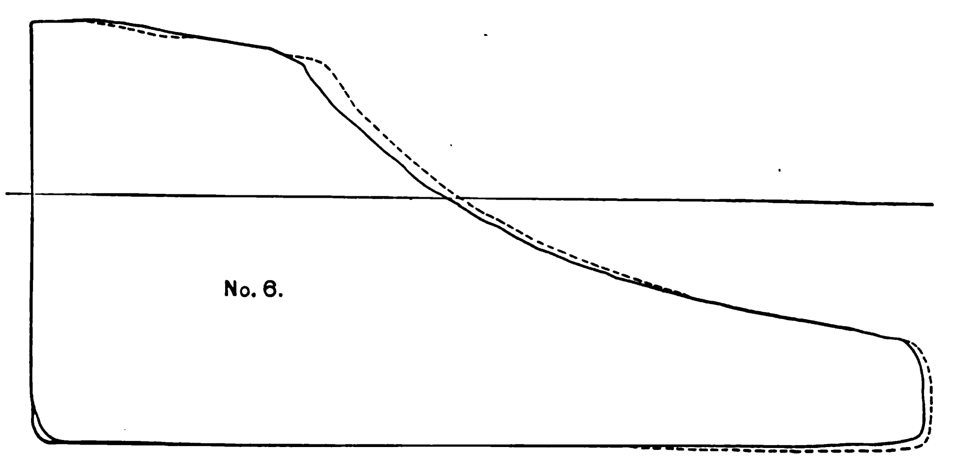

Fig. 45.

MAXIMUM AND MINIMUM CARDS MAY 14. OUTSIDE END.

Full Line, 1.10 P.M., M.E.P., 13.30.
Dotted Line, 4.55 P.M., M.E.P., 14.

the same variation in cut-off at the two ends of the cylinders for a given variation of load or of steam pressure.

The tests above reported, taken in connection with those previously reported, I think prove that in the use of the Pawtucket pumping engine the use of the jackets gives a saving of between 1% and 4%, but they do not lead to any more general conclusion than that jackets may be expected to give this saving in a cross compound Corliss engine of 140 H. P., running at about 50 revolutions per minute, supplied with dry steam of 125 lbs. gauge pressure, and cutting off at about one-quarter stroke in the high and one-third stroke in the low-pressure cylinder. Before this conclusion can be expanded to apply to other engines, there should be tests made with equal precautions and refinements to those made with the Pawtucket engine, on such other engines, with different dimensions and different conditions, such as pressure of steam, moisture or superheat in the steam, speed of revolution, number of expansions in the two cylinders, etc.

In view of the rapidly increasing use of compound engines with high pressures and higher speeds than the Pawtucket engine, it is much to be desired that a series of tests should be undertaken in order to learn whether in such engines the steam-jacket causes any saving sufficient to compensate for its extra cost.

It is gratifying to be able to confirm the general results as to the economy of the Pawtucket engine, which have been found by Profs. Denton and Jacobus, as well as those that have been made by Mr. Walker, the engineer of the water-works. It is worthy of notice, also, that this engine, built nearly fifteen years ago, has not yet been surpassed in economy by any two-cylinder compound engine, nor as a pumping engine has it yet been surpassed by any type of engine whatever.

THURSDAY, MAY 14. JACKETS IN USE.

Time.	Water consumed per half hour, lbs.	Engine revolutions.	Height water lifted.	Water pressure, lbs. per sq. in.	Total water pressure, lbs. per sq. in.
11:15	917	1,444	13 ft. 10¼ ins.	112.5	118.50
11:45	1054	1,444	13 10½	112.5	118.50
12:15	981	1,442	13 10¾	116	121.99
12:45	960	1,444	13 11½	114	120.03
1:15	947	1,445	14 1	113	119.09
1:45	1026	1,443	14 2	112	118.13
2:15	886	1,416	14 2	112	118.13
2:45	1006	1,445	14 3	114	120.17
3:15	1033	1,443	14 4	113	119.20
3:45	998	1,435	14 5	114	120.24
4:15	951	1,437	14 6	114	120.28
4:45	989	1,443	14 6	113	119.28
5:15	974	1,442	14 6¼	112	118.30
	Av. per hour, 1956. Av. per H. P. per hour, 13.687 lbs.	Av. per hour, 2,880 = 48.0 per min.			Av..119.37

Average boiler pressure by gauge.. 123.5 lbs.
" temperature due to pressure .... 351.9°
" " of steam in pipe 2 feet from steam-chest........... 355.1°
" degrees of superheating.. 3.2°
" of steam in jackets (both) .... 348.0 lbs.
" pressure in receiver, pounds above atmosphere............... . 9.1 "

INDICATOR, AVERAGE MEAN EFFECTIVE PRESSURE.

Time.	High Pressure.		Low Pressure.	
	Inside.	Outside.	Inside.	Outside.
11:40	57.25	60.75	12.45	13.65
12:05	61.00	60.	12.85	13.70
12:35	58.75	62.25	13.	13.85
1:10	57.25	59.50	12.95	13.30
1:55	58.50	60.25	12.60	13.50
2:10	61.	60.	12.75	13.40
2:35	58.25	59.25	12.80	13.40
3:07	58.25	58.75	12.75	13.40
3:35	58.75	59.25	12.65	13.75
4:05	58.50	57.	13.20	13.85
4:32	58.00	58.75	13.	13.80
4:35	59.25	61.	12.79	13.47
4:55	59.50	59.25	13.05	14.
5:02	55.25	58.50	12.08	13.45
Average........	58.53	59.61	12.83	13.61
H. P. Const....	.6211	.6427	2.5704	2.592
Horse-power...	36.35	38.31	32.97	35.27

Total horse-power.. 142.90
Average pressure of steam in high-pressure jacket........................ 121.5 lbs.
" temperature of feed-water.. 63.8°
" " " engine-room.. 83.3°
Height of barometer.. 30.111 ins.
Average vacuum by gauge .... 27.9 "

FRIDAY, MAY 15, JACKETS NOT IN USE.

Time.	Water consumed. Lbs.	Engine revolutions per half hour.	Height water lifted.	Water pressure. Lbs. per sq. inch.	Total water pressure. Lbs. per sq. inch.
8:30	938	1,443	14 ft. 1½ ins.	111½	117.61
9:00	1002	1,440	14 1	110½	116.59
9:30	935	1,435	14 0½	112	118.07
10:00	1081	1,439	14 0	111½	117.56
10:30	1017	1,440	13 4	110½	116.52
11:00	1078	1,439	13 10½	111	117.01
11:30	900	1,440	13 10	111	116.99
12:00	1008	1,438	13 9	112½	118.46
12:30	1014	1,435	13 10	113½	119.49
1:00	1088	1,438	14 00	113½	119.56
1:30	984	1,440	14 0½	110½	116.58
2:00	1009	1,438	14 1	110	116.09
2:15	534				
2:30	502	1,443	14 1	111	117.09
3:00	988	1,442	14 0	111½	117.56
3:30	999	1,442	13 11½	112½	118.54
4:00	997	1,443	13 11	111	117.02
4:30	1021	1,442			
4:45	528				Av. 117.51
	Av. per hour, 2007. Av. per H. P. per hour, 14.256 lbs.	Av. per hour, 2,879 = 48.0 per min.			

Average boiler pressure by gauge ....128.5 lbs.
" temperature due to pressure..351.9 deg.
" temperature of steam in pipe 2 feet from steam-chest................362.8 "
" degrees of superheating..9.9 "
" of steam in jackets, high..257 "
" " " low..197 "
" pressure in receiver, pounds above atmosphere........................ 9.2 lbs.
" " " high-pressure jacket .... 0 "

INDICATOR, AVERAGE MEAN EFFECTIVE PRESSURE.

Time.	High Pressure.		Low Pressure.	
	Inside.	Outside.	Inside.	Outside.
8:20	59.25	60.50	12.80	12.80
8:45	60.75	60.75	12.25	12.55
9:22	60.50	60.75	12.30	12.40
9:48	61.50	61.50	12.30	12.50
10:50	60.75	61.	11.90	12.65
11:55	61.75	58.75	12.25	12.60
11:55	63.	60.50	12.10	12.30
12:45	62.25	59.25	12.65	13.10
1:12	63.75	57.75	12.45	12.70
1:42	61.50	57.50	12.45	12.55
2:57	59.75	56.75	12.60	12.55
3:52	61.50	57.50	12.25	12.85
4:45	61.	56.50	12.70	12.95
Average.........	61.82	59.25	12.38	12.65
H.P. constant...	.6211	.6427	2.5704	2.592
Horse-power....	38.09	38.08	31.82	32.79

Total horse-power 140.78
Average temperature of feed-water.. 63.9°
" " " engine-room.. 83.0°
Height of barometer .... 30.180 inches.
Average vacuum by gauge .... 27.3 "

traffic. Hard-burned clay blocks or brick, if sufficiently vitrified to stand the wear, do quite well on a level street. On the least grade, however, they afford but slight foot-hold for the horse, and are too brittle to endure the heaviest traffic. Asphalt will not stand severe usage, wearing into depressions and hollows which need constant attention. It is also when wet very dangerous to horses, because so slippery that no safe foot-hold can be obtained. Granite and Belgian block do stand the traffic, but at what a cost of wear and tear to vehicles. They are very noisy, and injure the feet and limbs of horses by their rigidity, also when somewhat worn and wet they give a dangerous foot-hold.

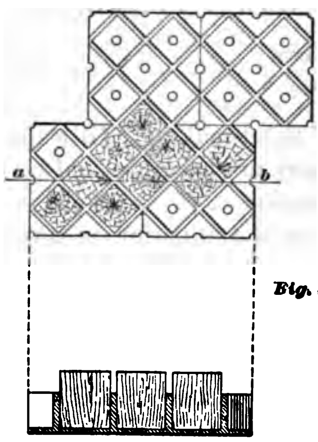

Fig. 46.

The desideratum for the best pavement appears to be one which would give a good and safe foot-hold to the horse, which would afford a good surface for traffic and which would be durable. One which could combine lack of noise with these other advantages would appear to be well in the front rank for meeting the requirements of to-day.

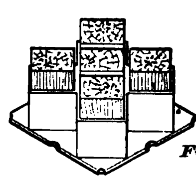

Fig. 47.

There is a small piece of pavement, 9 feet by 27 feet 4 inches, in the city of my residence, at a place where the traffic is very heavy, a crossing over the sidewalk from our principal business street into the freight yards of the " Big Four Line," which has attracted considerable attention. It is a combination pavement of cast iron pockets and bottom plate with oak blocks, making an even bottom surface and thus avoiding settling into holes. These iron pockets are filled with green oak blocks $3\frac{7}{16}$ inches square on the top and about 5 inches long. These blocks are driven into the iron pockets some 3 inches, and receive the traffic on the end of the grain of the wood. Each full-sized bottom plate has 5 pockets, 4 half-pockets and 4 quarter-pockets. See detail drawings, Fig. 46 and Fig. 47. Thus at

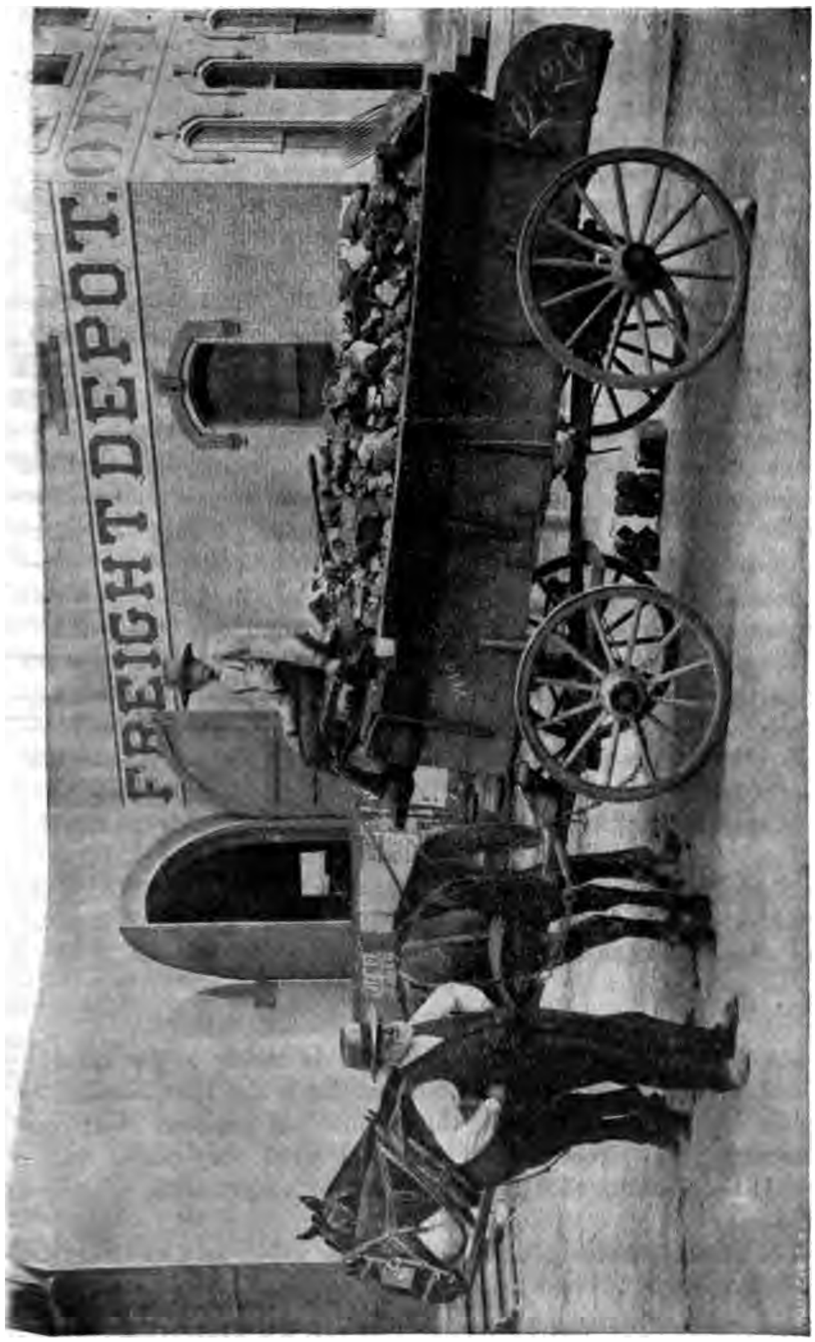

FIG. 48.

DISCUSSION.

Prof. D. S. Jacobus.—After examining the tests made on the Pawtucket engine, including those by Mr. Kent, Mr. Donkin, of the Committee on Steam Jackets of the British Association of Mechanical Engineers, expressed the opinion that the small effect of the jackets may be due to the fact that the steam entering the engine is slightly superheated. The same idea has been suggested by Mr. Kent and Mr. Barrus. Mr. Nagle, also, at a previous meeting,* advanced the decided opinion that the superheating is the cause of the small effect of the jackets.

It being desired on the part of Prof. Denton and myself that no question be left open regarding the performance of this remarkable engine, Prof. Denton arranged to have an additional test made with steam containing a small percentage of moisture, and the privilege of making these measurements was accepted by the speaker. The results are now reported as follows:

On November 6th the engine was tested for eight hours without jackets in use, and with steam superheated 17° Fahr. two feet from the cylinder. The results gave 14.16 lbs. per hour per horse-power, which is 0.8% lower than Mr. Kent's test, under similar conditions, but with only 11° of superheating.

A portion of the steam pipe between the boiler and the engine, about 15 feet long, was then stripped of the non-conduction covering. The extra radiation loss from the pipe thus permitted reduced the superheating to about 5° Fahr.; but as it was desirable to destroy all superheating, water from the boilers was injected into the steam pipe in such an amount as to destroy the remaining superheating and produce a small degree of wetness. The exact degree was not determined, but from what follows it is fair to conclude that the moisture averaged at least 0.1%. The engine was then again tested for seven hours on November 7th, and the result was 14.40 lbs. of steam per hour per horse-power, a result differing from that obtained without jackets, and 17° of superheating by 1.7%, which is less than the amount of variation ascribable to the accidental errors of the most careful tests. The detailed results are given in Tables I. to IV., from which we draw the following general conclusions:

1st. The gain in economy by using the jackets and flue

* *Trans.* American Society of Mechanical Engineers, Vol. XI., page 1064.

reheater with steam superheated about 5° Fahr. (Mr. Kent's conditions), over that obtained when the jackets and flue reheater are not in use, and with steam containing a slight percentage of moisture, is about 31%.

2d. If the engine is run with and without jackets, with slightly moist steam, the probable difference in favor of the jackets, allowing the same gain in economy per degree of superheating, with the jackets in use as was observed in the tests without the jackets, will be 3%.

3d. The small advantage of jackets in the Pawtucket engine, as compared to the large gain by jackets reported for other engines, is not ascribable to the small amount of superheating in the steam.

The increase of efficiency for 17° of superheating, calculated from the water pressure and revolutions, is 1.5% against 1.7% by the indicator cards. This agreement is a satisfactory check on the indicator measurements, but does not verify the figure itself, which depends directly on the measurement of the steam used by the engine.

In the test with moist steam it was necessary to estimate the amount of entrained water. This was done by means of the heat-gauge portion of the Barrus Universal Calorimeter. This instrument gave excellent results, and by its use it was possible to accurately regulate the moisture carried over by the steam, so that the average for the entire test is about 0.1%, a quantity much less than the error involved in using any other form of calorimeter. A small globe valve was used to regulate the rate of flow of this water. The readings of the Barrus calorimeter responded quickly to any change of opening in this valve, thus exhibiting its ability to indicate very small amounts of entrained water in a large volume of rapidly moving steam.

Radiation of the calorimeter was allowed for by stopping the engine and noting the reading of the two thermometers as the steam gradually lost its superheat and became saturated. The pressure of the steam during this experiment was preserved at precisely that maintained during the test.

The pipe leading to the calorimeter projected about 2 inches through the lower side of the steam pipe, at a point where it leads horizontally to the engine. There were no small holes made in the side of the calorimeter pipe, as is recommended by Mr. Barrus, because it was desired to obtain as dry a sample of steam as possible, so that any moisture shown would be the minimum for the whole amount of steam.

The calorimeter drew its steam from a point in the pipe about one foot from the thermometer placed in the latter, and about three feet from the engine. The amount of steam used by the calorimeter per hour was determined both before and after the test, and is deducted in the tables from the engine consumption.

The indicator springs were standardized directly after the test, and the proper scale employed in working up the diagrams. As the accuracy of the test does not depend on the absolute readings of the gauges, they were not calibrated.

The average cards, corrected for variations in the scales of the indicator springs, are shown in Figs. 61-64.

The water was weighed in a tank and emptied into a second tank, from which it was pumped to the boilers. The capacity of the weighing tank was such that it was filled only once each half hour, at which time the water was brought to a certain level in the second tank. The reading of the height of water in the boilers was taken at the time that the water tank was filled to the mark, so that the rate may be estimated for any portion of the run.

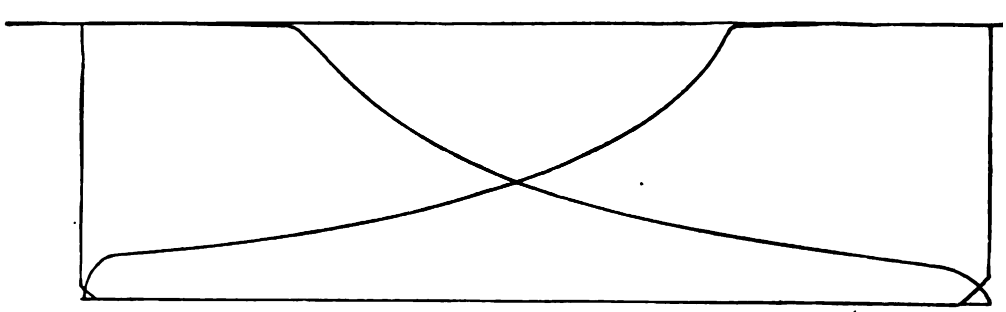

FIG. 61.

Test made November 6th. Superheated Steam. H. P. Cylinder. Scale, 80 pounds per inch.

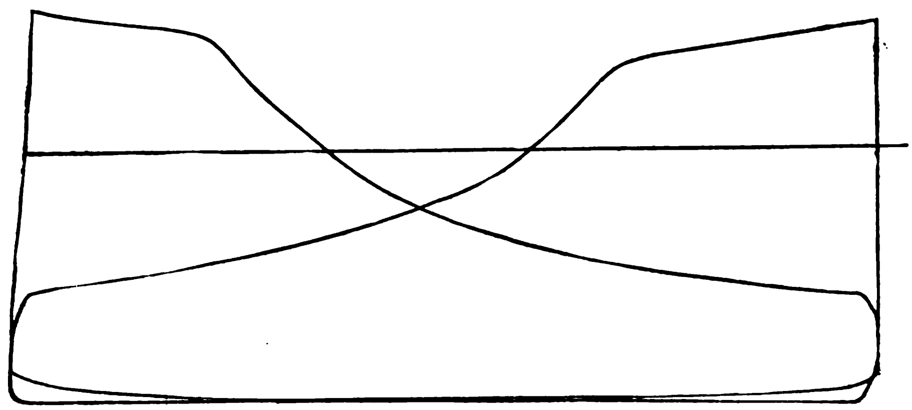

FIG. 62.

Test made November 6th. Superheated Steam. L. P. Cylinder. Scale, 10 pounds per inch.

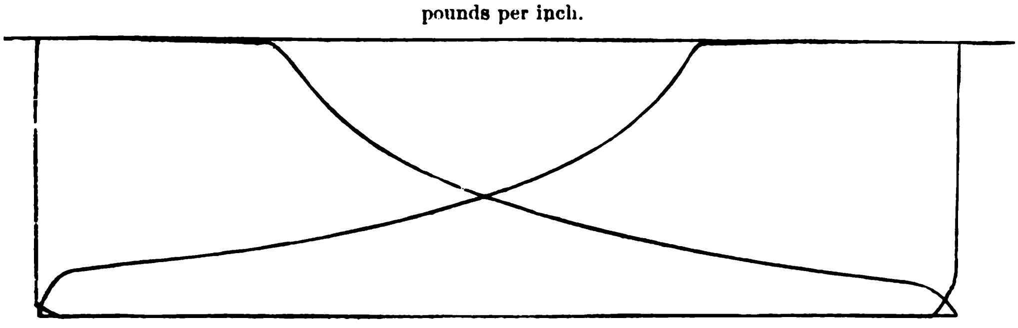

FIG. 63.

Test made November 7th. Moist Steam. H. P. Cylinder. Scale, 80 pounds per inch.

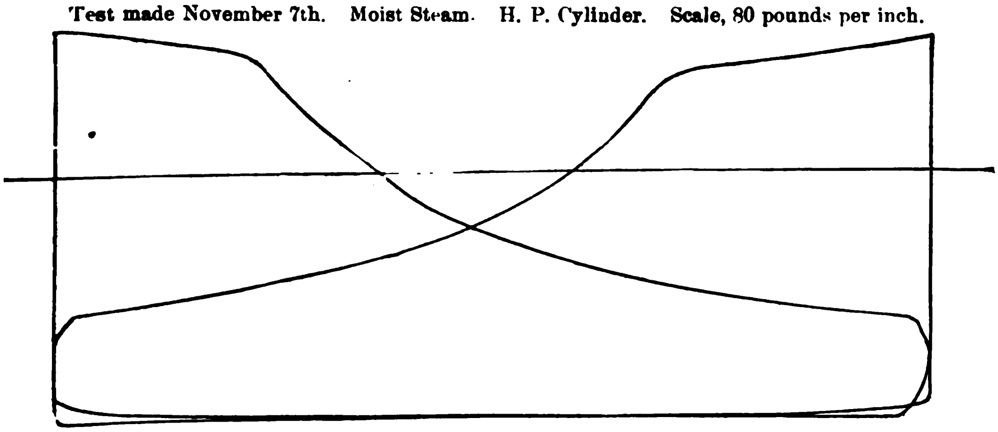

FIG. 64.

Test made November 7th. Moist Steam. L. P. Cylinder. Scale, 10 pounds per inch.

It will be noticed that the high-pressure admission line is more perfect than that of the cards shown at pages 182–184. This is due simply to using an indicator piston having less leakage.

TABLE I.

STEAM CONSUMPTION OF THE PAWTUCKET PUMPING ENGINE, WITH SUPERHEATED AND MOIST STEAM.

Bore of steam cylinders, 15″ and 30½″. Stroke of pistons, 30″. Diameter of all rods, 2¾″. Jackets and flue reheater not in use.

Condition of steam	Superheated. 17° Fahr.	0.1 per cent. of priming.
Date of test	Nov. 6.	Nov. 7.
Duration of test in hours	8	7
Average pressures:		
Boiler in pounds per square inch above atmosphere	124.0	124.0
Water " " " "	110.18	111.40
Vacuum in inches of mercury	27.8	27.3
Suction in feet	13.48	13.95
Barometrical pressure in inches of mercury	30.18	30.07
Average temperature in degrees Fahr.:		
Reading of thermometer in steam pipe two feet from steam chest	367.2	350.5
Reading of thermometer for saturated steam at boiler pressure	350.5	350.5
Degrees of superheating	16.7	0.0
Temperature of engine room	72.0	75.8
Mean effective pressure in steam cylinders in pounds per square inch:		
High-pressure cylinder. Head end	60.24	60.40
" " Crank end	62.96	63.68
Low-pressure " Head end	11.384	11.419
" " Crank end	12.058	12.250
Calculated quantities:		
Revolutions per minute	48.15	48.18
Total horse-power	138.73	139.95
Steam used per hour	1964.4	2014.6
Steam per hour per horse-power	14.16	14.40

TABLE II.

TEST IN WHICH STEAM IS SUPERHEATED 17° FAHR. MADE NOVEMBER 6, 1891.

Time, A.M.	Height of Water in Gauge-glass in Inches.				Water fed to Boilers in Pounds.		Pressure in Pounds per Square Inch.			Vacuum in inches of mercury.	Height that suction water is lifted in feet and inches.	Reading of thermometer in steam pipe in degrees Fahr.	Temperature of engine room.	Revolution counter five minutes later than other records.
	Boiler No. 1.	Boiler No. 2.	Boiler No. 3.	Sum of heights.	Total.	Net per hour corrected for water level and steam used by Barrus' calorimeter.	Boiler.	Receiver.	Water.					
8:30	3.8	4.4	4.0	12.2	....	....	124	8.0	112	26.8	13.7	366	71	399,871
9:00	5.1	4.5	3.5	13.1	1,018	....	124	7.8	109	27.1	13.6	365	72	401,605
9:30	4.6	2.6	5.0	12.2	1,977	1,977	124	7.8	108	27.0	13.6	367	72	402,761
10:00	3.9	4.6	3.4	11.9	2,950	....	124	7.8	109	26.9	13.5	369	72	404,206
10:30	4.5	2.4	4.3	11.2	3,891	1,959	124	7.8	108	26.9	13.2	367	72	405,653
11:00	4.1	4.3	3.6	12.0	4,928	....	123½	7.6	109	28.1	13.1	368	72	407,101
11:30	4.3	4.1	4.4	12.8	5,938	1,975	124	7.6	109	27.8	13.0	367	72	408,549
12:00	2.0	1.9	4.5	8.4	6,656	....	124	7.6	110	27.6	13.0	367	72	409,997
P.M.														
12:30	2.3	2.5	6.6	11.4	7,822	1,947	124	7.8	111	27.3	12.11	369	72	411,481
1:00	5.7	5.2	6.2	17.1	9,119	....	124	7.8	110	27.1	13.6	370	73	412,876
1:30	5.9	5.2	6.1	17.2	10,086	2,003	124½	7.8	109	26.9	13.6	367	73	414,323
2:00	4.8	4.6	4.9	14.3	10,941	....	124	7.8	112	27.3	13.8	367	72	415,769
2:30	4.2	4.8	4.5	13.5	11,881	1,958	124	7.8	112	27.1	13.7	365	72	417,212
3:00	4.5	4.1	4.8	13.4	12,854	....	124	7.6	112	26.9	13.11	367	72	418,655
3:30	5.2	4.2	3.9	13.3	13,854	1,947	123½	7.5	110	28.0	14.4	367	72	420,097
4:00	4.6	4.6	4.1	13.3	14,846	....	124½	7.6	112	27.8	13.11	367	72	421,541
4:30	4.1	4.2	4.8	13.1	15,832	1,949	124½	7.5	111	27.5	13.8	368½	72	422,985
Aver. and per hour.	4.3	4.0	4.6	13.0		1,964.4	124.	7.7	110.18	27.3	13.48	367 2	72.	2,289.2

TABLE III.

TEST IN WHICH STEAM CONTAINS 0.1 PER CENT. OF MOISTURE.

Time, P.M.	Height of Water in Gauge-glass in Inches.				Water fed to Boilers in Pounds.		Pressure in Pounds per Square Inch.			Vacuum in inches of mercury.	Height that suction water is lifted in feet and inches.	Reading of thermometer in steam pipe in degrees Fahr.	Temperature of engine room.	Revolution counter five minutes later than other records.
	Boiler No. 1.	Boiler No. 2.	Boiler No. 3.	Sum of heights.	Total.	Net per hour corrected for water level and steam used by Barrus calorimeter.	Boiler.	Receiver.	Water.					
1:15	9.6	14.3	11.0	34.9	....	....	124	8.4	114	27.8	14.4	350¼	75	446,778
1:45	12.4	11.5	11.7	35.6	1,116	....	124	8.6	114	27.1	13.10	350	74	448,221
2:15	13.1	11.7	10.8	35.6	2,187	2,109	124	8.2	114	27.0	13.10	350¼	74	449,664
2:45	11.1	12.8	12.3	36.2	3,270	....	124	8.0	113	27.8	13.11	350½	74	451,107
3:15	11.6	13.1	10.7	35.4	4,276	2,062	124	8.2	114	27.8	13.11	351	74	452,549
3:45	12.9	11.6	11.7	36.2	5,261	....	124	8.2	113	27.1	13.11	350½	75	453,990
4:15	10.6	11.5	11.5	33.6	6,318	2,0?0	124½	8.5	114	27.1	13.11	350½	75	455,430
4:45	12.0	13.4	10.4	35.8	7,426	....	124	8.2	114	27.5	13.11	351	76	456,870
5:15	12.9	12.8	12.1	37.8	8,573	2,021	124	8.0	112	27.3	13.11	350½	76	458,311
5:45	13.9	13.8	11.9	39.6	9,658	....	124	7.5	109	27.1	14.0	350½	78	459,762
6:15	11.1	11.5	10.0	32.6	10,317	1,940	124	7.0	108	27.1	13.11	350½	79	461,214
6:45	9.1	10.0	10.2	29.3	11,096	....	124	7.2	109	27.8	13.11	351	78	462,664
7:15	10.3	11.9	12.1	34.3	12,363	1,941	124	7.2	106	27.1	13.11	350½	78	464,114
7:45	10.5	12.1	10.5	33.1	13,276	....	124	7.2	108	27.1	14.0	351	76	465,564
8:15	11.5	10.9	11.0	33.4	14,284	1,959	124	7.2	109	28.1	13.11	350	75	467,014
Aver. and per hour.	11.5	12.2	11.2	34.9		2,014.6	124	7.84	111.4	27.8	13.95	350.53	75.8	2,890.9

TABLE IV.

BARRUS UNIVERSAL CALORIMETER.

Record of thermometer readings observed during test made November 7, 1891. Reading of lower thermometer for dry steam, 290½ Fahr.

Time, P.M.	Temperature in Degrees, Fahr.		Time, P.M.	Temperature in Degrees, Fahr.	
	Upper Thermometer.	Lower Thermometer.		Upper Thermometer.	Lower Thermometer.
1.20	349	287	3.50	349	288
1.30	349	286	4.00	349	288
1.40	349	288	4.10	348	288
1.50	348	288	4.20	348	288
2 00	348	290	4.30	348	289
2.10	348	290	4.40	348	289
2.20	348	290	4.50	349	289
2.30	348	290	5.00	348	288
2.40	348	290	5.10	348	289
2.50	348	290	5.20	348	289
3.00	348	290	5.30	348	288
3 10	348	289	5.40	348	288
3.20	348	289	5.50	348	288
3.30	348	289	6.00	348	288
3.40	348	289	Average......	348.2	......288.7

Average number of degrees of cooling........................1.8
Corresponding per cent. of priming............................0.1

Mr. Horace See.—The simple compound engine of the steamship *Ohio* and her sister ships, built in 1873, had their cylinders steam-jacketed. They were run for days with and without steam in the jackets to ascertain whether any good could be derived from the use of the former. The coal consumption was found to be no less with these engines, which made 60 revolutions per minute, with the jacket in use than with the steam shut off.

The triple compound engines of the steamship *Aberdeen*, built in 1882, had the intermediate and low pressure cylinders jacketed, but not the high. They were tested by Dr. A. C. Kirk, their designer and builder (who, by the way, should be made an honorary member of this society for his valuable work in the development of the steam-engine), assisted by Mr. Parker, of the British Board of Trade.*

The tests were made with and without steam in the jackets,

* Utilizing Steam of the Higher Pressures. Horace See, Engineers' Club. Philadelphia, 1884.

and the engines making 65 revolutions per minute. The drain water was measured in the former case. The result was found to be practically the same whether the same amount of steam was used in the jackets of the intermediate and low pressure cylinders or utilized directly in the cylinders.

One of the conclusions reached by Dr. Kirk was that "when we take into account the thickness of the interior chamber of the cylinder and the speed of the piston, it is not conceivable that the interior surface of the chamber can be maintained, by the steam in the inside of it, at a uniform temperature. In fact, the slower an engine runs the more efficient the jacket becomes."

The tests made by Messrs. Kent, Denton and Jacobus, of the Pawtucket engine, not only emphasize those made of the marine engine as to the value of the steam-jacket, but have also demonstrated that an engine having unjacketed cylinders can be run as slow as 48 revolutions per minute without impairing its economy.

The careful work of these gentlemen is valuable in throwing additional light upon a subject upon which, for many years, there has been much diversity of opinion, and if properly considered should do much toward relieving the steam-engine of a costly and dangerous adjunct.

Mr. Kent.—I would just like to ask Mr. Jacobus, in regard to these last figures, if the steam used per hour and per horsepower is the equivalent of dry saturated steam, or is it the actual steam?

Prof. Jacobus.—It is the actual steam and all the water injected into the pipe.

Mr. Kent.—The 14.16 result is for the superheated steam?

Prof. Jacobus.—Yes; this is for steam superheated 17° Fahr.

Mr. Fk. Meriam Wheeler.—I would like to ask if there is any one present who can say whether the engines of the steamship *City of Paris* are jacketed or not. Those engines showed a remarkable economy, but accounts in the journals omit to state whether the cylinders were jacketed.

Mr. F. W. Dean.—The cylinders of the *City of Paris* and the *City of New York* are jacketed, but those of the *Majestic* and *Teutonic* are not.

Mr. Wheeler.—I desired to ask the question because I was told they were not jacketed; and I was surprised to learn that, because the English practice is to jacket marine work.

CCCCLXXIV.*

A COMBINATION IRON AND OAK PAVEMENT.

BY J. WENDELL COLE, COLUMBUS, O.
(Member of the Society.)

In his introduction to his "Manual of Road-making," Gillespie says very truly: "The roads of a country are accurate and certain tests of the degree of its civilization . . . their improvement keeps pace with the advances of the nation in numbers, wealth, industry, and science, of all which it is at once an element and an evidence." If you substitute street-pavement for roads, and city for nation, the same language is, if possible, a greater truth than when published in 1847. The genesis of pavements has been like the development of naval warfare—a strife between the attack and the defence. As in the latter it has been between the increasing power of ordnance and the resisting qualities of armor-plate, so in pavements the first aim was to make vehicles which bad pavements would not break, and then later to make a pavement that the traffic could not destroy; but no matter if the pavement did destroy the traffic—*i. e.*, the vehicles and the horses.

Mechanics have from time to time entered the arena with varied forms of metal pavement, but they soon wore smooth so as to become more dangerous than stone, or if roughened for a foot-hold were liable to tear off the shoes of horses or to fracture and give way and be short-lived. One after another metallic pavements have been rejected, and road-makers to-day favor for pleasure-driving the "macadam" and asphalt, where the famous shell-roads are unobtainable. For the ordinary traffic of commerce they favor hard-burned clay blocks, asphalt, and granite block, while for the congested traffic of the streets of our metropolitan cities Belgian block or granite block have been by common consent adopted. But are any of these pavements above criticism? Macadam will not survive a heavy commercial

* Presented at the New York Meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

traffic. Hard-burned clay blocks or brick, if sufficiently vitrified to stand the wear, do quite well on a level street. On the least grade, however, they afford but slight foot-hold for the horse, and are too brittle to endure the heaviest traffic. Asphalt will not stand severe usage, wearing into depressions and hollows which need constant attention. It is also when wet very dangerous to horses, because so slippery that no safe foot-hold can be obtained. Granite and Belgian block do stand the traffic, but at what a cost of wear and tear to vehicles. They are very noisy, and injure the feet and limbs of horses by their rigidity, also when somewhat worn and wet they give a dangerous foothold.

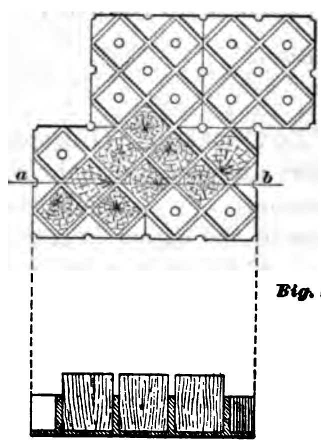

Fig. 46.

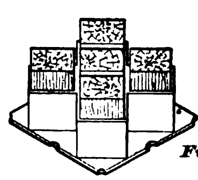

Fig. 47.

The desideratum for the best pavement appears to be one which would give a good and safe foothold to the horse, which would afford a good surface for traffic and which would be durable. One which could combine lack of noise with these other advantages would appear to be well in the front rank for meeting the requirements of to-day.

There is a small piece of pavement, 9 feet by 27 feet 4 inches, in the city of my residence, at a place where the traffic is very heavy, a crossing over the sidewalk from our principal business street into the freight yards of the " Big Four Line," which has attracted considerable attention. It is a combination pavement of cast iron pockets and bottom plate with oak blocks, making an even bottom surface and thus avoiding settling into holes. These iron pockets are filled with green oak blocks $3\frac{7}{16}$ inches square on the top and about 5 inches long. These blocks are driven into the iron pockets some 3 inches, and receive the traffic on the end of the grain of the wood. Each full-sized bottom plate has 5 pockets, 4 half-pockets and 4 quarter-pockets. See detail drawings, Fig. 46 and Fig. 47. Thus at

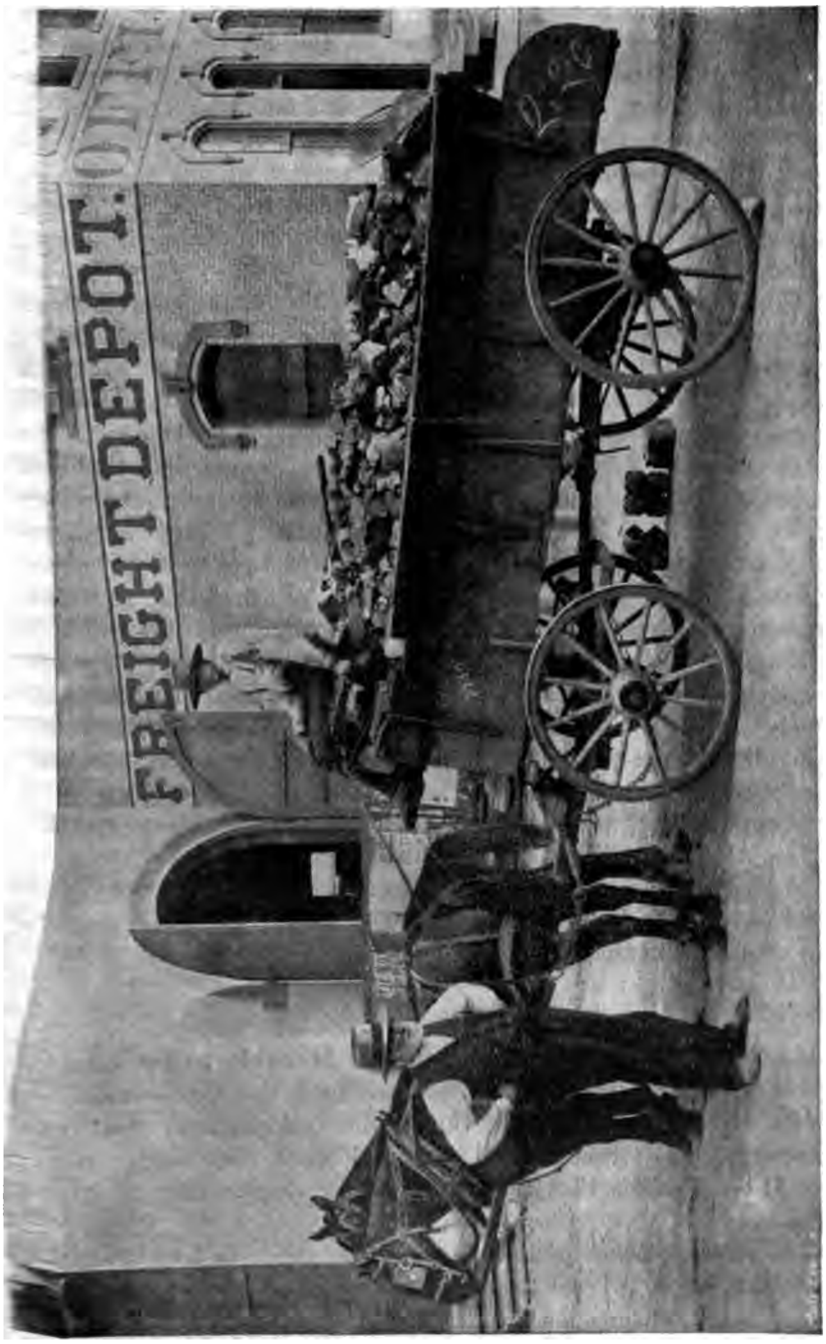

Fig. 48.

the joints of the bottom plates in some instances, 2, 3, or even 4 bottom plates may support one of the hard-wood blocks, dividing the strain and holding an even surface on the upper side of pavement. Each iron pocket has a small hole for draining away any moisture. The "Mosstype" view of the pavement (Fig. 48) shows how evenly this surface stands the wear, and the load of coal taken in view—the first team passing after camera was adjusted—shows the kind of traffic using it; *i. e.*, "loaded wagons."

This block pavement was put down May 30, 1890, with the newly sawed oak blocks projecting 2 inches. In August, 1890, I inspected it and found them hammered and worn to $1\frac{7}{8}$ inch projection; and in September, 1891, another inspection showed them projecting about $1\frac{3}{4}$ inch above the metal pockets. The traffic crossing it includes largely loads of roofing slate and of rough curb-stones, in addition to the general commercial traffic of the "Big Four Line," and loads of from 3 to 5 tons each are quite frequent. The railway yard surface is broken-rock "macadam." The street is asphalt. I am informed that at this crossing new road metal has been deposited "in the yard" within 90 days, while the asphalt was renewed for some space from this block pavement about four months ago. By looking under the wagon on the "Mosstype" view is seen an iron bottom plate before being filled; also two views of filled bottom plates ready for setting.

It appears that this pavement so far meets the requirements of

1st. A substantial roadway to stand heavy traffic.

2d. A good firm foot-hold for the horse.

3d. A practically noiseless roadway suited to heavy traffic, but also comfortable for pleasure driving or a wheelman.

DISCUSSION

Prof. John E. Sweet.—The question is liable to be raised, whether the construction of street paving is one that comes directly within the province of the mechanical engineer; but one comparing our streets as they *are* with what they ought *to be*, will at once admit that there is room for engineering of some kind.

The street pavement described in the paper is a mechanical structure, no doubt one that will endure an excessive amount of punishment, and in places where the pavement will wear out before it will decay, though very expensive, might still be justifiable.

The function performed by the cast iron is that of holding the blocks in place, which they will do effectually provided the castings are bedded on unyielding foundation.

In my own city, which is built upon a reclaimed cedar swamp, this composite pavement, unless built upon a foot or two of solid masonry, would not stand much longer than it would take to build it. With a solid masonry foundation it would seem that the castings might be dispensed with. It is true the castings (until the blocks are worn down to their level) hold the blocks apart, making, what the author of the paper deems a necessity, a foot-hold for the horses. This the writer believes to be a mistake, at least in all places except on hillsides of considerable slope. Believing that a team will draw more upon ice, and with less danger of injury than on any other track, the nearer a street approaches a dead plane the better. Horses slip on all kinds of pavement, not because they are flat, but because the wheels of the vehicles have to be drawn out of the holes, and making holes for the calks of horse-shoes, makes holes for wagon wheels; and so, if the street pavement is a perfect plane, made of some material not liable to become slippery (and the end grain of wood is about the best of all things), a team will draw more with much less exertion and less danger of injury than if channels are left for the foot-hold. These channels always grow larger, and of course the top of the blocks or stones more crowning, so that channels become the starting-point for the street to grow worse and worse. Diagonally across the street is the direction to lay the joints, but as shown in the paper, the evil effects of the channels between the blocks are made doubly worse by the arrangement. The corners of the blocks are the parts which give way first, and, as shown, four corners come together, and the hole will be twice as large and deep as if the blocks were placed with their diagonals lengthwise of the roadway and their edges breaking joints in successive lines.

Good roadways can be built, using various materials for what John Richards calls the carpet, provided the foundation upon which the carpet is laid is an absolutely true and unyielding foundation. The trouble is not with the top of our streets, which we are always looking at and talking about, but the bottom, the foundation, which we never pay any attention to, because we cannot see it. And this all grows out of the "great American idea," which will not be content with anything less than 40 or 50 feet of road-bed, however shabby, whereas, 14 or 16 feet of thoroughly good roadway would more than answer every purpose.

We make our roadways wide and poor, with unsightly gutters, which never serve any purpose except to hold rubbish and filth and stagnant water. We pay lots of money to sprinkle down the dust into mud, to dry up into the same old dust, to be again sprinkled down into the same old mud, rather than to make a narrow, good, flat road-bed, easily kept clean by sweeping and carting the dust away. With a narrow road-bed there will not be one half the dust to blow about, and with the doubly wide lawn between the road-bed and walk, half more of the dust is caught in the grass before it reaches the walks or houses at all.

Our people in the provincial towns cannot afford to build wide road-beds good, even could they be convinced what the word good means, but they could afford to build narrow ones good, and if they would only spend one half day to watch any residence street in any city in the country, they could convince themselves that 14 or 16 feet is ample, and with a perfect foundation of that width, which they could well afford, carpeted with seasoned wood-blocks steeped in bitumen, or vitrified brick set diagonally, thoroughly bedded in asphalt or cement, or a complete asphalt pavement laid flat, the beauty added to our cities would be a revelation to our people.

The place for the electric car tracks throughout the residence portions of our cities is along the lawns between curbstone and sidewalks; neater, cleaner, more convenient and cheaper for the companies, as they would have no pavement to maintain, and a great nuisance removed from every vehicle and team which travels the street.

Mr. George L. Fowler.—This pavement seems to be practically a wooden pavement. I had considerable experience with wooden pavement in Michigan, where they used round cedar blocks. As to the pavement mentioned in the paper the author does not say anything in regard to its duration, simply because it has been down such a short time, I suppose. The specifications which were used for the cedar blocks were that no block should be more than six inches in diameter after the bark was trimmed off. The street was graded with a turnpike crowning and laid with about six inches of sand. We had an opportunity of getting good, sharp sand to put in the bottom. On top of that we placed a planking of about one and one-eighth inch pine boards. Where the better class of pavement was to be laid we put down shipping culls, and where the poorer sort was to be laid we laid mill culls. That pavement with the ordinary traffic which will obtain in the principal streets of a manufacturing town of 25,000 or 30,000 inhabitants, would last about seven years before it would have to be relaid. Very little repairing would have to be done in the meanwhile if the work were properly done in the first place. There is one difficulty I should mention. Where I happened to be located we were liable to floods, and when one was imminent the principal streets looked like the scrap heap of a poorly conducted foundry, because the authorities went all over and borrowed all the heavy pieces of scrap iron they could get hold of and laid them on the streets to keep the pavement from floating off. The blocks were cut off square on the ends. There were several machines in the market for taking off the bark. The best one was simply a cylinder with sharpened edges and the blocks were driven down to it. The man who operated would have, I suppose, half a dozen sizes of these cylinders and a press beam rising and falling by machinery; the smallest cylinder being three and a half inches in diameter. A block which would go through that size without touching was condemned. As I say, that pavement would last about seven years and keep in good condition all the time.

Mr. Wood.—I would like to inquire if there is any arrangement to remove the sap-wood as well as the bark, and what is the cost of this kind of pavement.

Mr. Fowler.—I cannot answer the question of cost exactly, but I know it is very much less, because there was no work on the blocks except sawing off and driving through these cylinders and placing in position. The bark only is removed.

Mr. Woolson.—What is the comparative cost between the pavement Mr. Cole writes about, and the Belgian blocks?

Mr. Cole.—I cannot answer as to Belgian blocks, because I do not know what they cost. But I asked the Western Paving Co., in Chicago, what granite block pavement cost in Chicago. They said it was about $4.35 per square yard. This combination pavement has, of course, not been made in large quantities. In the quantities yet made it would require for foundation about the same as granite block, which the above Paving Co. says is about eighty-five cents a square yard. The castings would cost about $3.00 and the oak blocks would cost about $1.00 per square yard, making the whole cost about $4.75 per square yard, making it a more expensive pavement than granite block. But I inquired of Mr. F. H. Taylor, of Chicago, and he estimates that in quantities made with moulding machines the price could be reduced for the castings to about $2.60 per square yard, and I think from inquiries I have made in several places that the oak blocks could be furnished in Chicago at about 85 or 90 cents per square yard, thus reducing the price to about the same as granite blocks in Chicago. This cedar pavement which has been mentioned by the gentleman is largely used in small towns through the Northwest and in the timber districts, and for small towns and residence streets in cities—medium sized cities of the West, even to the suburbs of Chicago, is cheaper than granite block or this pavement which I have presented, and just as good in every respect, and better because cheaper. But I speak of this iron and oak pavement only for the heavy severe traffic such as you get in this city on Broadway; and on Chestnut Street and parts of Broad Street in Philadelphia; and in the central part of Chicago. There it would probably cost no more than granite block, except possibly a little more for creosoting, which would make the blocks stand a good deal longer by removing the sap and preventing decay; and would be cheaper to renew, as the metal sockets could be used over again. But for residence streets and streets where the traffic is light, and the parties are satisfied with cedar blocks, I would recommend cedar blocks. This iron and oak pavement is to take the place of the stone pavement, which does not give a foothold to the horse, and which is very expensive when we look at it as reducing the life of horses used in traffic—I think I would be safe in saying twenty per cent, and that is something which very few persons have ever estimated in taking pavements into consideration.

Mr. Parsons.—I would like to say that at present experiments are being made in France, and I understand with success, on a wooden pavement, laid as follows: Splitwood is used, something like the wood used for burning in a grate. It is stood on end, the spaces between the pieces filled with fine gravel and sand, and the whole kept wet; the wood swells and makes a tight, firm pavement.

Mr. Cole.—We are all familiar with the manner in which wood is used in the metal socket, of a carpenter's framing chisel. You take the same wood and submit it to the same uságe, not held in a metal socket, and its life would not be one half what it is held in a metal socket. I believe that this holding of these oak blocks in this metal socket doubles their life.

Mr. Oberlin Smith.—I think there is one feature of this paving business to which too little attention is paid in all discussions regarding it, and that is the infinite superiority of asphalt and wooden pavements over *any* kind of stone pavement, in the matter of avoiding noise and jar. If any pavement is reasonably smooth and durable, it is generally considered good enough. A great many people think there can be nothing more perfect than a Belgian pavement. To my mind it is an abomination, and any pavement which does not make such horrible noise and jar would be cheap at double the price, comparing it with the best stone pavement that can be made. I do not believe very many people realize the enormous nerve wear-and-tear constantly going on in our cities, due to the truly infernal noise and the constant jar of all business places and residences alongside of these stone pavements. I believe that the time is coming when the world will wake up to a sense of this, and when the best stone pavement, to say nothing of cobble-stones, will be considered but a relic of barbarism.

Mr. Woolson.—In the last paragraph of Mr. Cole's paper, the statement is made that his pavement wears one-eighth of an inch in three months; that makes half an inch a year. I understand from this private explanation to me, that it is hammering the blocks into their sockets and not wear, except in a small degree.

Mr. Le Van.—In the city of Hanover, Germany, "India-rubber" pavements have been in use for some three years, and have given such good satisfaction, that the city of Berlin has also put some down near the Lutzow Bank. Hamburg is also considering the advisability of adopting the same.

This pavement is the invention of a German engineer, named Busse; it is said to consist of 80% of ground stone, and 20% of rubber. It is said to combine great elasticity with the hardness of stone, to be completely noiseless, and to suffer neither from cold nor hot weather. Moreover, it is not slippery, like asphalt, and is more durable.

Mr. Ashworth.—The experience of the city of Pittsburgh as regards wooden block pavements with which a large area of the streets were fitted up, has been so unfortunate that it would impress even a stranger going out among the suburbs to see the amount of wear and tear. The point I want to make is this: there is a pavement in the Ohio Valley; I think it was first tried in the city of Wheeling, and it has now reached Steubenville and those smaller cities, and has reached a number of other prominent towns in the Ohio Valley. It consists of a hard brick, approaching fire-brick, placed on edge; and having been in all these places, I have received exceedingly high and favorable reports of the durability of this style of pavement, and it has been seriously considered in some of our metropolitan cities, and it is spreading through our Western country. That seems out West to be the coming pavement.

The President.—The city of Detroit has put down a great deal in the last year.

Mr. Crane.—I would like to ask if they can make these block pavements non-absorbent. I have heard that advanced in London as one objection to them that their block pavements were absorbent, and after a shower an odor arose from them which was unpleasant.

Mr. Cole.—I am familiar with Wheeling fire-brick. In the paper, I have said: "For the ordinary traffic of commerce they favor hard burned clay blocks." Then a few lines further on: "Hard burned clay blocks or brick, if sufficiently vitrified to stand the wear, do quite well on a level street. On the least grade, however, they afford but small foothold for the horse, and are too brittle to endure the heaviest traffic." I paid about three or four years ago for 130 feet lineal on the curb of the street of this Wheeling fire-brick laid, at what was at that time my residence, and that pavement is in good shape to-day. But on State street, Columbus, O., fire-brick, where it gets twice or three times the wear, is deteriorating quite fast.

In reference to Mr. Crane's inquiry about making the oak blocks non-absorbent, "creosoting" or "kyanizing" the wood blocks before placing them in the metal sockets would remove that objection.

Referring again* to Mr. Woolson's remarks, I do not think he clearly understood my explanation.

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* Author's closure.

In the portion of the paper he refers to, it mentions as he says, "a wear of one-eighth inch in three months," but that was the first three months the pavement was in use, and as an inspection made over a year later, referred to in the same paragraph, shows one-eighth inch wear per annum, we can safely infer that the apparent greater wear the first three months in use was largely caused by a hammering of the blocks into the metal sockets, and a battering of the tops, which possibly became a "buffer" to assist in cushioning the wear of the following year.

Referring to the discussion of Prof. Sweet, would say, to prevent decay of the wood, "kyanize" or "creosote" the blocks.

In reference to "the reclaimed cedar swamp foundation" (or lack of foundation), to make it suitable for any pavement, except "ice" or corduroy, it would be necessary to first drain it, and then deck it over with masonry or concrete, and any foundation which would be satisfactory with Belgian or granite block, fire-brick or cedar block, would be just as satisfactory with this "Iron and Oak" * pavement, or even more so, as the iron plates assist in forming a deck, while in no place does a surface joint appear, as all the joints in the metal are covered by the wood blocks, that is if the wood blocks extend through the metal deck. When inspecting the roadway surface it is impossible to identify the joints in the metal by any difference in the looks of the pavement.

In reference to whether a foot-hold for the horse is a necessity or not, allowing "that a team will draw more upon ice * * * than any other track," in order to make it "with less danger of injury" also, it *must be* a sharp-shod team.

Had Prof. Sweet led a smooth-shod horse one and a half miles to a smith-shop on a sleety, icy morning, as the writer did once when a lad, he would realize the necessity of a foot-hold. On the return trip I was in the saddle, and both horse and rider had more confidence.

The old Russ pavement on Broadway, New York, became "a smooth, flat pavement," but because horses could not stand upon it and draw loads it was finally taken up and split into sizes giving a foot-hold.

A good foot-hold for the horse is both a commercial and a humane necessity.

True that smooth lake or river ice forms a good pavement for

* Note.—G. Schreger's U. S. Patent No. 420,020.

drawing a heavy load, but it is practically without grade, and a sharp-shod horse cuts his own foot-holds, just where he needs them, which he cannot do except in a perishable material, too much so for a durable pavement.

If the Professor will closely scan the photo-engraving of street surface (page 197), he will see that the lines separating the wooden blocks are diagonal, but the lines separating the metal plates or deck underneath are at right angles to, and parallel with, the curb. And should Prof. Sweet be passing through Columbus, so he could see the pavement, it would be an easy matter to show him that the channels do not grow enough larger, and the tops of the blocks are so slightly crowning, after 17 months' heavy wear, in the heaviest traffic to be found in our city, that the pavement is practically as good as new, the fibres of the wood being held from spreading or splitting by the metal sockets.

If channels were cut into a wooden pavement not held in metal sockets, they would grow larger, as we have seen in the Nicholson.

In reference to foundations all our paving specifications in Columbus have required for some four years or more:

1st. An excavation of the soil.

2d. Five (or more) inches of broken rock, of sizes used for "Macadam road-metal," rolled with a steam roller.

3d. Two inches of sand on top of this broken rock, in addition to that which fills into the crevices of the rock, and again steam rolled. Then the pavement.

Prof. Sweet's point of narrowing the roadway is well taken, and is especially satisfactory to one who pays the street assessment, but extremes are to be avoided. I have found a 30-foot roadway very convenient, a 24-foot street might do, but I fear that in a city of 60,000, "and growing," a street with but 16-foot roadway, would not become popular, and real estate upon it would depreciate. The "arteries of travel," whether for business or pleasure, would probably demand some 40 to 60 foot roadway, and return the cost in an increased valuation of the real estate.

I thank the gentlemen who have so kindly discussed this subject, and believe we have all gathered some information about paving since the discussion opened.

CCCCLXXV.*

TEST OF A PULSOMETER.

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

THE pulsometer tested was one taken from the ordinary stock of machines of the Pulsometer Steam Pump Company, and known by them as the "New Pulsometer No. 6." It had a $3\frac{1}{2}$-inch suction pipe and discharge pipe, stood 40 inches high, and weighed 695 pounds. The object of the test was to determine the duty and efficiency as a lifting pump under varying conditions. It was run as nearly as could be under the actual conditions in practice in regard to throttling. The test was made at Stevens Institute, by C. G. Atwater and Charles B. Hodges, for their thesis work for 1891.

The suction and delivery pipes were each 4 inches diameter. Holes were drilled into the pipes just before entering and just after leaving the pump for attachments of pressure gauges and the insertion of mercury wells for determining the temperature.

The water pumped was measured by a meter kindly loaned by Mr. John Thomson, member of the Society. It had eight-inch diameter outlet and inlet, and was a rare piece of apparatus to be so easily secured for such a purpose.

The return pipe for receiving the discharged water was of such construction as to avoid siphoning. The water was returned to a tank from which it passed through a meter to the large tank from which the water was originally pumped. An even temperature was maintained in the large tank by admitting cold water and providing for an overflow of the surplus amount.

The steam pipe was 1 inch in diameter, and a throttle placed about 2 feet from the pump, and pressure gauges placed on both sides of the throttle, and a mercury well and thermometer placed beyond the throttle. Thus a definite knowledge of the steam was obtained. The wire-drawing due to throttling caused superheating.

* Presented at the New York meeting (1891) of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

The pounds of steam used were computed from the increase of temperature of the water in passing through the pump. We have

Pounds of steam $\times$ *loss of heat* $=$ *pounds of water sucked in* $\times$ *increase of temperature,*

from which the pounds of steam may be computed. The loss of heat in a pound of steam with the total heat in a pound of saturated steam as found from "steam tables" for the given pressure, plus the heat of superheating, minus the temperature of the discharged water; or

$$\text{Pounds of steam} = \frac{\textit{lbs. water} \times \textit{increase of temp.}}{H - 0.48t - T}.$$

The work was computed by means of the pressure gauges and the quantity of water as measured by the meter. The gauges were 3 feet apart vertically, and this amount was added to the head of the suction. This method determined the work done by the pump, as it eliminated the effect of friction and losses due to bends, or, in short, any losses in and along the pipes.

The efficiency is the ratio of the work done to the heat absorbed, both measured in foot-pounds.

RECORD TAKEN DURING TEST 1.

TIME.	STEAM PRESSURE LBS. IN SQ. INS.		TEMPERATURE IN DEGREES FAHR.			Reading of meter in cu. ft.	GAUGE READINGS.		Strokes per min.	HEIGHT OF WATER, INS.		REMARKS.
	Initial.	Final.	Water. Initial.	Water. Final.	Steam.		Lift lbs. sq. in.	Suction ins. of Head'g.		Tank No. 1.	Tank No. 2.	
4:20						11,785	13	8¼	70	60	70.0	
4:30	128	18	78.7	82.8	291.5	12,280	13	8¼	72	61	70.0	
4:40	110	20	75.7	79.9	290.0	12,692	13	8¼	72	65	68.5	
4:50	120	18	74.4	77.8	276.0	13,002	13	8¼	66	57	64.5	
5:00	125	18	74.0	77.6	275.0	13,382	13	8¼	66	57	63.6	
5:10	115	20	73.4	78.1	289.0	13,761	13	8¼	72	58	63.6	
5:20	114	20	74.1	78.7	260.0	14,162	13	8¼	72	62	63.6	
5:30	109	21	74.6	79.8	263.0	14,572	13	8¼	74	65	63.6	
5:40	109	18	75.8	80.2	258.0	14,975	13	8¼	72	64	63.6	
5:50	111	19	75.0	79.3	277.0	15,377	13	8¼	71	64	63.6	
6:00	124	19	74.6	79.0	257.5	15,775	13	8¼	71	63	63.6	
6:10	104	20	74.7	79.7	263.0	16,175	13	8¼	73	63	59.2	
6:20	116	20	75.3	80.0	276.0	16,585	13	8¼	73	62	57.7	
6:30	108	20	75.6	80.8	264.0	16,992	13	8¼	73	62	57.3	
6:40	110	20	76.3	81.2	265.0	17,400	13	8¼	76	62	57.3	
6:50	115	19	76.4	80.8	278.0	17,805	13	8¼	72	64	57.3	
7:00	112	19	74.7	79.0	263.0	18,205	13	8¼	71	65	57.3	
7:10	112	18	74.2	78.8	256.0	18,603	13	8¼	70	64	57.3	
7:20		19				19,004	13	8¼		66	55.4	
		19	75.15	79.62	270.7		13	8¼	71			

Test 1 gives:

Average rise of temperature of water	4.47°
Average degrees of superheating	13.1°
Total heat of steam above 79.62°	1118.67°
Water pumped, lbs	404786
Heat absorbed B. T. U	1815127
Mech. equivalent of heat absorbed, foot-lbs	1412169047
Steam used by temp. test, lbs	1617
Height of lift, actual, in feet	25.3
Dynamic head by gauge on lift, ft	29.9
Height of suction, actual, ft	7.5
Reading of vacuum gauge on suction, ft	12.26
Total work as determined by gauges, *u* ft. lbs	16996886
Efficiency, *u* ÷ heat absorbed	.012
Duty = foot-lbs. per 10 lbs. steam	105114

The other three tests were conducted in a similar manner. The results for the four tests are given in the following table:

TEST OF A PULSOMETER BY C. G. ATWATER AND CHARLES B. HODGES.

DATA AND RESULTS.	NUMBER OF TEST.			
	1	2	3	4
Strokes per minute	71	60	57	64
Steam pressure in pipe before throttling	114	110	127	104.3
Steam pressure in pipe after throttling	19	30	43.8	26.1
Steam temp. after throttling, deg. Fah	270.4	277	309.0	270.1
Steam amount of superheating, deg. Fah	3.1	3.4	17.4	1.4
Steam total heat above that of water, B.T.U.	1118.67	1112.44	1127	1121.2
Steam used as determined from temp., lbs.	1617	931	1518	1019.9
Water pumped, lbs	404786	186362	228425	248053
Water temp. before entering pump, deg. F.	75.15	80.6	76.3	70.25
Water temp. after leaving pump, deg. Fah.	79.62	86.1	83.79	74.80
Water temp. rise of	4.47	5.5	7.49	4.55
Water heat absorbed, B. T. U	1815127	1024993	1704498	1143527
Water head by gauge on lift, ft	29.90	54.05	54.05	29.90
Water head by gauge on suction	12.26	12.26	19.67	19.67
Water head by gauge, total (H)	42.16	66.31	73.72	49.57
Water head by measure on lift	25.30	50.30	50.3	25.30
Water head by measure on suction	7.5	7 50	16.3	16.30
Water head by measure, total (h)	32.8	57.80	66.6	41.60
Coefficient of friction of plant (h) ÷ (H)	0.777	0.877	0.911	0.839
Total work as per gauges	16966886	12235940	16795192	12275923
Efficiency of pulsometer	0.012	0.0155	0.0126	0.0138
Efficiency of plant exclusive of border	0.0093	0.0136	0.0115	0.0116
Efficiency of plant if that of border be 0.7	0.0065	0.0095	0.0080	0.0081
Duty of pump per 100 lbs. coal if 1 lb. evaporates 8 lbs. water	8409120	10712800	8847200	9629040
Duty if 1 lb. evaporates 10 lbs. water	10511400	13391000	11059000	12036300

Of the two tests having the highest lift (54.05 ft.), that was more efficient which had the smaller suction (12.26 ft.), and this was also the most efficient of the four tests. But, on the other hand, the other two tests having the same lift (29.9 ft.), that was the more efficient which had the *greater* suction (19.67), so that no law in this regard was established. The pressures used, 19, 30, 43.8, 26.1, follow the order of magnitude of the total heads, but are not proportional thereto. No attempt was made to determine what pressure would give the best efficiency for any particular head, and there is a field for experiment in this particular. The pressure used was intrusted to a practical runner, and he judged that when the pump was running regularly and well, the pressure then existing was the proper one. It is peculiar that, in the first test, a pressure of 19 lbs. of steam should produce a greater number of strokes and pump over 50 per cent. more water than 26.1 lbs., the lift being the same, as in the fourth experiment.

APPENDIX.

STANDARDIZATION OF INSTRUMENTS.

The meter was intended for 6-inch piping, and was rated at 40 cubic feet per minute under a head of 10 feet. The calibrations were made first to find out just what correction should be applied under the conditions existing in the tests; secondly, to find the absolute accuracy of its reading, passing water at the temperature used. The first test was all that was absolutely necessary for the calculations; the second was made merely as an investigation of interest. The method employed was as follows: The discharge pipe was carried over into tank No. 2, in which the calibration was made by means of the water-glass and scale attached. Tank No. 1 was filled with water to about 70 inches depth and heated to 75° by steam, this being approximately the temperature of the water entering the pulsometer during tests. Tank No. 2 was emptied till about 8 inches of water remained. This was then pumped from tank No. 1 to tank No. 2 passing through the meter. The meter readings were taken at 0, 25, 50, 75, 100 cubic feet, the tank being read simultaneously by an assistant. The pump was always started a little before the first reading was taken to prevent the registry of the air in the pipe as water. The strokes per minute were made to coincide as nearly as might be with those during the test. It will be seen that the absolute accuracy of the meter itself was of secondary importance, the main thing being to determine the value of the meter readings under the conditions of the test. The dial of the meter was graduated to read 25 cubic feet, and the readings were always taken from a graduation mark. Tank No. 2 was measured and contained 1.86 cubic feet per inch of depth. The results are appended.

CALIBRATION OF METER.

	Meter Readings in cubic feet.	0	25	50	75	100	Strokes per minute.	Temp. in deg. Fahr.
Cu. ft. by tank.	Test I...........	0	24.2	45.4	68.8	91.1	68	74
	Test II...........	0	25.7	48.3	69.9	93.9	68	74
	Test III...........	0	21.4	42.8	66.5	88.3	60	75
	Test IV...........	0	24.2	46.5	69.7	91.3	60	75

The first two readings in each test are not as accurate as the latter ones, owing to the vibration of the water in the tank, which ceased as the water grew deeper. The correcting factor used was .9.

Having learned from Mr. Thomson that the calibration was intended to be perfect, another test was made, in which the head in the tank from which the water was taken for passing through the meter was considerably increased, and it registered correctly. From this it was inferred that during the regular tests, in which the head was about one foot above the end of pipe, air found its way into the pipe and passed through the meter.

GAUGES AND THERMOMETER.

The gauges were compared with a standard kept at the Institute, which in its turn had been compared with a mercury column, and the errors were not sufficient to affect accuracy of results.

The thermometers used were made by J. H. Green. Those used for the water temperatures were 20 inches long and graduated to one-fifth of a degree. Those used in the first two tests were known to be correct. These were broken and replaced by others, which were carefully compared throughout the range of temperature of the test and found accurate. The thermometer used in the steam was also correct.

DISCUSSION.

Mr. Chas. E. Emery.—I have been interested in comparing the performance shown by the pump referred to in the paper with the results obtained with vacuum and other pumps tested by the writer and others, the results of which are given in the *Report of the Judges of Group XX., Centennial Exhibition*, at page 20. In that report the speaker calls attention to the fact that with a good boiler a pound of coal will impart 10,000 British thermal units to the water, or 1,000,000 heat units for 100 pounds of coal. This is in proportion to the number of millions of foot-pounds per 1,000,000 thermal units or 100 pounds of coal, so the number of foot-pounds per thermal unit equals also the duty, or the number of millions of pounds of water lifted one foot high per 100 pounds of coal. A vacuum pump tested by the speaker, in 1871, gave a duty on the above basis of 4.7 millions; one tested by J. F. Flagg, C.E., at the Cincinnati Exposition, in 1875, reduced to the same basis, gave a maximum duty of 3.25 millions. Several vacuum and small steam pumps, compared later on the same basis, at the suggestion of the speaker, were reported to have given duties of 10 to 11 millions, the steam pumps doing no better than the vacuum pumps, for reasons there stated. The duty on the same basis shown by the second experiment recorded in the paper is 10.5 millions. This is on the basis of the actual delivery to a given height per thermal unit. If the friction be taken into consideration, the duty is somewhat greater. On page 121 of the report previously referred to, will be found a comparative statement of the performance of various kinds of engines compared on this same basis of duty. Evidently the maximum duty possible is fixed by the mechanical equivalent of heat at 772 millions. Injectors, when used for lifting water not required to be heated, have an efficiency of 2 to 5 millions; vacuum pumps vary generally between 3 and 10; small steam pumps between 8 and 15; larger steam pumps between 15 and 30, and pumping engines between 30 and 120 millions.

CCCCLXXVI.*

THE IDIOSYNCRASIES OF CHIMNEY DRAUGHT.

BY W. E. CRANE, WATERBURY, CONN.

(Member of the Society.)

ONE of the favorite themes of the compiler of formulæ is that of chimney draught, and it becomes an easy matter to calculate just what draught a certain chimney ought to give. The engineer, however, who must design and use these chimneys, finds it a complex problem, as there are little things which enter into it which the formulæ do not take into account.

We are told that it makes no difference if the inside flue stops somewhere in the chimney or is continued to the top; that it may be either larger or smaller at the top without detriment.

The question will naturally arise as to the effect on the velocity of a current of air or gas if it is suddenly enlarged, as it must enlarge when the inside flue is shorter than the chimney itself, as it must be in a measure affected the same as when issuing from the top. We are also told that the draught cannot be increased after the temperature arrives at 600°; and if we wish a draught of .85 inch of water, we must have a chimney or stack about 116 feet high, although this is attained in a stack 90 feet high; while with that height (90 feet) .75 to .80 inch is a common thing where the temperature of gases entering the chimney is 800°.

By some claiming to be authorities it is urged that a chimney must grow gradually larger from bottom to top.

With a view of getting some light on the subject, the writer has taken the draught from a number of chimneys working under various conditions, and presents the results for the consideration of the Society. He regrets that he has not the time at his disposal to give the subject the attention it deserves, and also regrets the lack of a complete set of accurate instruments. For this season the author does not claim that the observations are

* Presented at the New York meeting (1891), of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

more accurate than any before made, but are rather comparative with this set of observations.

CHIMNEY No. 1.—One hundred and thirty feet high; flue, 100 feet high; round, 4 feet diameter; taking gases from Babcock & Wilcox boiler, 59½ feet grate surface; and Harrison boiler, 17 feet grate; consumption of coal, 14 lbs. per square foot of grate, and temperature of gases, 340°. Draught in inches of water, .55 inch.

CHIMNEY No. 2.—Ninety feet high; flue square, 28 inches, continued to top of chimney, taking gases from 15 casting furnaces, each furnace 1.3 feet of grate and burning 25 lbs. of coal per foot; temperature of gases, 775°. Draught, .77 inch.

CHIMNEY No. 3.—Seventy-eight feet high; flue, 28 inches square, continued to top of chimney; 15 casting furnaces, same as No. 2, but burning but 19 lbs. coal; temperature not taken, but is sufficiently high to burn out the rounds that are placed in the chimney for a ladder for some 50 feet from the bottom. Draught, .64 inch.

CHIMNEY No. 4.—Ninety feet high; flue, 27 inches square, 75 feet high; 20 casting furnaces, same as No. 2, and burning about 17 lbs. coal; temperature not taken, but bottom of chimney at a dull red heat. Draught, .46 inch. With 10 furnaces and damper on others closed tight, .50 inch. With damper open 3 x 18 inches, and 10 furnaces, draught, .40 inch.

It is a mystery yet why this chimney falls behind the others doing the same work, as there is but one thing which is different and that is the stopped-off flue. This chimney was built by the lowest bidder, and when the draught was taken cracks had appeared, which may account in a measure for the poor results.

CHIMNEY No. 5.—One hundred and forty-eight feet high; flue, 48 inches square, carried to top of chimney; 2 Babcock & Wilcox boilers, with total grate area of 99½ feet; and 15 casting furnaces, discharging gases into chimney through a flue 3 x 4 feet. It was in this flue close to chimney that the draught gauge was placed. Through another flue gases from 44 casting furnaces are delivered into the same chimney; consumption of coal under boilers, about 14 lbs. per hour per square foot of grate, and casting furnaces about the same as No. 2; flue temperature from boilers, 350°, and the average combined temperature is not known, but would probably be 500° to 550°. Draught, .70 to .75 inch, and irregular, and the opening of a furnace door or pulling off a furnace cover is noticeable, as the chimney is overloaded.

CHIMNEY No. 6.—An iron stack $3\frac{1}{2}$ feet in diameter at bottom and 4 feet at top, 68 feet high; temperature of flue gases, 375°; 3 Harrison boilers, with grate surface total of 51 square feet; coal consumption, 12 lbs. per hour per square foot of grate. Draught, .35 inch.

Having occasion to change the location of these boilers, the stack was set large end down. At time of writing but two of the boilers are set; the other will be before the meeting of the Society. Draught practically the same, although gauge indicates from .01 to .02 inch better since the change.

From another set of chimneys are the following results:

CHIMNEY No. 1.—Forty-eight inches square flue, carried to top of chimney, 95 feet high, taking gases from 56 casting furnaces, each 16 inches square, and burning about 32 lbs. coal per hour in each furnace. Draught, .75 inch.

CHIMNEY No. 2.—Sixty feet high; flue, 18 inches square, carried to top, taking gases from annealing furnaces; 2 set grates, 9 x $1\frac{1}{2}$ feet; fuel, wood—about 1 cord chestnut wood in 9 hours. Draught, .40 inch.

CHIMNEY No. 3.—Ninety feet high, iron stack 30 inches diameter, with 16 casting furnaces same as No. 1. Draught, .85 inch.

CHIMNEY No. 4.—Eighty feet high, dimensions of flue not known; used for annealing. Draught, .50 inch.

CHIMNEY No. 5.—One hundred and thirty-five feet high; flue, 48 x 50 inches, carried to top of chimney, and used for boilers and annealing. The temperature of the gases from the boilers is reduced to 240° by an economizer, and in the chimney is again raised by the discharge from the annealing furnaces to some unknown temperature, but not to exceed 400°. Draught, .55 inch.

CHIMNEY No. 6.—Sixty feet high; dimensions of flue not known; used for annealing. Draught, .40 inch.

The draught on all these was taken on a clear morning, with temperature of air at 60° to 65°.

The writer hoped to have a much larger number of observations, and to have had more details, such as barometer readings, temperatures, etc., but has not had the time. From those taken, however, it shows the effect of temperatures in connection with draught in chimneys. He had hoped to find more examples with flues shorter than the chimney, but such cases are rare in his locality.

The temperature from casting furnaces may be taken as fr m 700° to 800°, and annealing furnaces at 700°. It would seem from the above that a manufacturer who was building chimneys for high temperatures would not need to build as high as he would for his boiler chimneys.

If it shall be proven that forced draught for boilers is the more economical, there would seem to be no good reason for building chimneys over 100 feet high.

This is important, as one chimney for casting was built 200 feet high, as it was claimed that better draught for casting furnaces was needed than for boilers, and it was supposed that the height governed the intensity of the draught.

The length, shape, and size of flue has much to do with draught in the furnace, but all the above were taken from the base of the chimneys.

The appended table summarizes these observations in a convenient form for examination and comparison:

OBSERVED DRAUGHT OF VARIOUS CHIMNEYS.

No. of chimney.	Height of chimney, ft.	Height of flue, ft.	Size and shape of flue.	Kind of furnaces and number.	Area of grate surface, sq. ft.	Coal burned per sq. ft. of grate per hour, lbs.	Temperature of gases.	Draught, ins. of water.	Remarks.
1	100	100	round, 4 ft. diam.	2 boilers*	76¼	14	840°	0.55	*1 Babcock & Wilcox, 1 Harrison.
2	90	90	28 ins. square	15 casting furnaces	19.5	25	775°	0.77	
3	78	78	28 ins. square	15 casting furnaces	19.5	19	.............*	0.64	*Not taken, but inside ladder burned out.
4	90	75	27 ins. square	20 casting furnaces	26	17	....*	0.46	*Not taken, bottom of chimney at a dull red heat.
4		...	....	....	......	......	....	0.50	Same chimney with 10 furnaces and damper on others closed tight.
4	...	...	....	....	......	......	....	0.40	Same chimney with damper open 8 x 18 ins. and 10 furnaces.
5	148	148	48 ins. square	2 boilers* / 50 casting furnaces	90.5 / 76.7	14 / 25	850° / ....*	0.70 to 0.75	*Babcock & Wilcox boilers. *Average combined temp., probably 500° to 500.
6	68	an iron stack, round*		3 Harrison boilers	51	12	....	0.35	*3½ ft. diam. at bottom, 4 ft. at top.
	The location of these boilers being changed, the stack has been reset, large end down. With two of the boilers running, the draught is practically the same as before, though the gage shows 0.01 in. to 0.02 better since the change.								
7	125	125	48 ins. square	56 casting furnaces	90.50	18	....	0.75	
8	60	60	18 ins. square	2 annealing furnaces	27.		....	0.40	1 cord chestnut wood in 9 hrs.
9*	100	100	30 ins. diam.	16 casting furnaces	28.8	18	....	0.85	*An iron stack.
10	80		Not known	annealing	......	......	....	0.50	
11	135	135	48 x 30 ins.	boilers and annealing	......	......	Not over 400°	0.55	
12	60		Not known	annealing	......	......	....	0.40	

DISCUSSION.

Prof. J. E. Denton.—Calculation of the water pressure for the cases 1, 2 and 6, where the author gives temperatures, shows that the observed pressure agrees with the difference of density due to temperatures. The calculated pressures do not allow for friction in chimney, but for the greatest velocity. This friction would only make about $\frac{1}{32}$ of an inch difference—too little to observe with an ordinary U tube.

If we knew the details of the flues leading to his stacks, we might show that the chimney heights agreed fairly well with the formula for height in *Rankine:*

$$n = \frac{n^2}{29} \left[G + 1 + \frac{fl}{m} \right]$$

In making 1 equal his chimney height, only [instead of all the flues leading to the stack as it should be] the calculated heights are as per last column, Table II.—all on the right side of the actual height.

In the case of the "Orange," I show, page 395, Vol. XI., that the pressure observed was 0.33 of water, and the calculated 0.37 which must be discounted for friction in *chimney.* Also, that where we *know* the air supply and the nature of the flues, the calculated chimney height was 61 feet against 54. In the "Bergen" the calculated height, page 423, is 59 versus 53 for actual height. If the facts were *all* observed in the case of Mr. Crane's chimneys, I think theory would acquit itself quite well enough to deserve respect.

TABLE I.

Calculation of the "head in feet in height of a column of hot gas in the chimney" by Rankine's Formula.* Also corresponding draught in inches of water.

No. of chimney.	Height in ft.	Dimensions of cross section.		Grate surface in sq. ft.	Rate of combustion per sq. ft. of grate per hour.	Temp. of gases.	Draught in ins. of water.		Calculated head in ft. of hot gas.
		Dimensions.	Area in sq. ft. = A.				Observed.	Calculated.	
1	130	4 ft. drain	12.57	76.5	14	340	.55	.62	58
2	90	28″ × 28″	5.44	19.5	25	775	.77	.72	112
6	68	3½ × 4 ft. diam.	Minimum 9.6	51.0	12	375	.35	.36	35

$$* \; h = H \left(0.96 \frac{\tau_1}{\tau_2} - 1 \right)$$

Corresponding head in inches of water $= h \times .161 \frac{\tau_0}{\tau_1}$

TABLE II.

Calculated Height of Chimneys, if the Grate Resistances are those common in ordinary boiler practice.

No. of chimney.	Height, Cranes.	Velocity* in chimney in ft. per sec.	Hydraulic mean depth of chimney.	Factor of resistance for friction of the flue $= \frac{Fl}{m}$	Heat of hot gas to produce velocity and overcome friction, if factor of resistance for grate is taken at 12.	$.96 \frac{\tau_1}{\tau_2} - 1$	Height of chimney = Col. 6 + Col. 7.
1	2	3	4	5	6	7	8
1	130	10.7	1.00	1.56	25.9	.45	58
2	90	17.3	.58	1.86	69.1	1.24	56
6	68	8.4	.88	.92	15.3	.51	30
Bergen	53	15.25	1.00	.64	49.2	1.10	45

* The volume of air per pound of coal is taken as 300 for all cases except the Bergen, when it is estimated to be 190.

Mr. Wm. Kent.—The force of draught is equal to the difference between the weight of the column of hot gas inside the chimney and the weight of a column of cool air of the same height outside the chimney.

If D is the density of the air outside, d the density of the hot gas inside, in pounds per cubic feet, h the height of the chimney in feet, and .192 the factor for converting pressure in pounds per square foot into inches of water column, then the formula for the force of draught expressed in inches of water is, $F = .192\, h\, (D-d)$

The density varies with the temperature. (See *Rankine. Steam Engine*, p. 287.) $d = \frac{\tau_0}{\tau_1}\, 0.084$; $D = 0.0807 \frac{\tau_0}{\tau_2}$ (for 30 inches barometer), where τ_0 is the absolute temperature at 332° F, $= 493.2$; τ_1 the absolute temperature of the hot gas, and τ_2 that of the external air. Substituting these values, the formula for force of draught becomes $F = .192\, h \left(\frac{39.8}{\tau_2} - \frac{41.43}{\tau_1}\right)$.

Mr. Crane makes one statement of a general case, viz.: that a stack 90 feet high will give a draught of .75 to .80 inch when the temperature of the gases is 800°. He also gives observations of the draught of twelve different chimneys, but in only three of these does he know the temperature of the gases, viz.: Nos. 1 and 2 on second page, and No. 5 on third page of his paper. In these three cases only are there enough data given to know whether they present any idiosyncrasies or not, or whether or not they conform to any formula. Tabulating Mr. Crane's general case and his three chimneys, we have the following, assuming the external air to be at 70° F.:

Chimney.	Height.	Temperature.	τ_1	τ_2	Draught Calculated by Formula.	Draught Observed by Mr. Crane.
1	130	340	801	531	.58	.55
2	90	775	1236	531	.72	.77
3	68	375	836	531	.33	.35
General	90	800	861	531	.73	.75 to 80

If we calculate the general case on the assumption that the external air is at 32° F., the calculated draught is .83 inch, and allowing for differences in temperature of the external air above or below 70°, it appears that his observations confirm the formula that his chimneys are not idiosyncratic, and that his criticism of the formula makers is not well taken.

Mr. W. B. Le Van.—I would like to make a few remarks on chimneys. A number of works have been written on this subject, the mathematics of which do not agree with the results obtained in my practice of the past thirty years.

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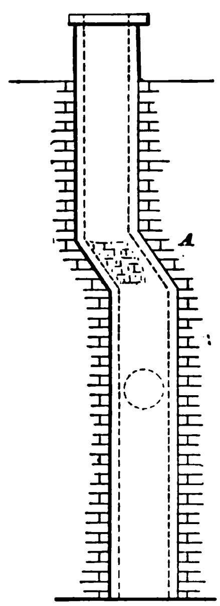
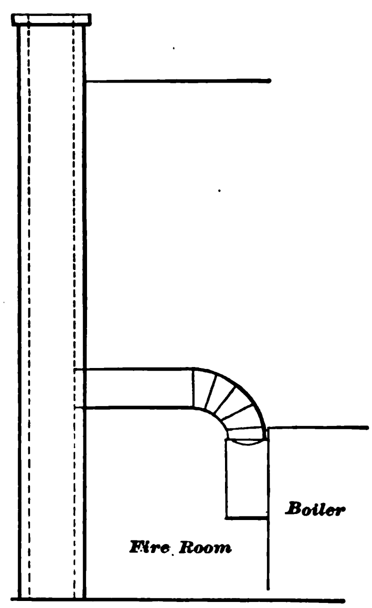


FIG. 49.



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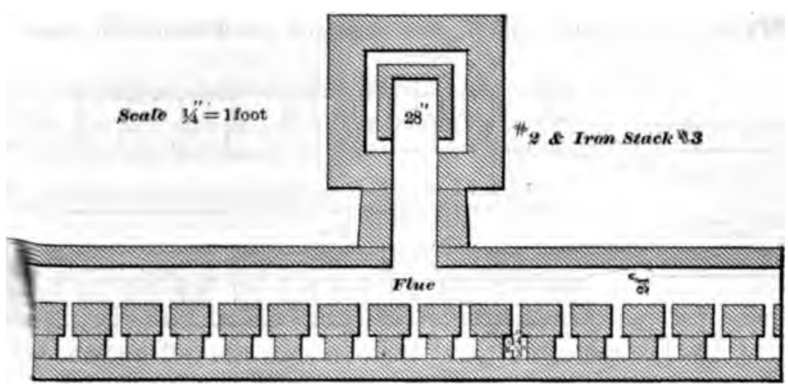

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the flue, we could not promise any satisfactory result from the boiler. The draught at the bottom, at the cleaning door, showed that by holding a match three or four inches away from it, it would blow it out instantly. The brick work was taken down, and it was found that the brick-mason had allowed a whole lot of bricks to accumulate in there, that almost entirely filled the offset up. There was not more than about six or eight inches of an opening. After the stack was cleaned out the boiler was found to be satisfactory in all respects.

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with draught in chimneys. He had hoped to find more mples with flues shorter than the chimney, but such cases rare in his locality.

The temperature from casting furnaces may be taken as fr m 0° to 800°, and annealing furnaces at 700°. It would seem om the above that a manufacturer who was building chimneys or high temperatures would not need to build as high as he ould for his boiler chimneys.

If it shall be proven that forced draught for boilers is the more conomical, there would seem to be no good reason for building himneys over 100 feet high.

This is important, as one chimney for casting was built 200 feet igh, as it was claimed that better draught for casting furnaces as needed than for boilers, and it was supposed that the height overned the intensity of the draught.

The length, shape, and size of flue has much to do with draught n the furnace, but all the above were taken from the base of the himneys.

The appended table summarizes these observations in a conenient form for examination and comparison:

OBSERVED DRAUGHT OF VARIOUS CHIMNEYS

No. of chimney.	Height of chimney, ft.	Height of flue, ft.	Size and shape of flue.	Kind of furnaces and number.	Area of grate surface, sq. ft.	Coal burned per sq. ft. of grate per hour, lbs.	Temperature of gases.	Draught, ins. of water.	Remarks.
1	130	100	round, 4 ft. diam.	2 boilers*	76¼	14	340°	0.55	*1 Babcock & Wilcox, 1 Harrison.
2	90	90	28 ins. square	15 casting furnaces	19.5	25	775°	0.77	
3	78	78	28 ins. square	15 casting furnaces	19.5	19	............*	0.64	*Not taken, but inside ladder burned out.
4	90	75	27 ins. square	20 casting furnaces	26	17	............*	0.46	*Not taken, bottom of chimney at a dull red heat.
4	....	....	....	....	.....	.....	....	0.50	Same chimney with 10 furnaces and damper on others closed tight.
4	....	....	....	....	.....	.....	....	0.40	Same chimney with damper open 8 x 18 ins. and 10 furnaces.
5	148	148	48 ins. square	2 boilers*	99.5	14	350°	0.70 to 0.75	*Babcock & Wilcox boilers.
				59 casting furnaces	76.7	25	............*		*Average combined temp., probably 500° to 500.
6	68	an iron stack, round*		3 Harrison boilers	51	12	....	0.35	*3½ ft. diam. at bottom, 4 ft. at top.
	The location of these boilers being changed, the stack has been reset, large end down. With two of the boilers running, the draught is practically the same as before, though the gage shows 0.01 in. to 0.02 better since the change.								
7	95	95	48 ins. square	56 casting furnaces	99.50	18	....	0.75	
8	60	60	18 ins. square	2 annealing furnaces	27.			0.40	1 cord chestnut wood in 9 hrs.
9*	90	90	30 ins. diam.	16 casting furnaces	28.8	18	....	0.85	*An iron stack.
10	80		Not known	annealing	.....	.....	....	0.50	
11	135	135	48 x 50 ins.	boilers and annealing	.....	.....	Not over 400°	0.55	
12	60		Not known	annealing	.....	.....	....	0.40	

DISCUSSION.

Prof. J. E. Denton.—Calculation of the water pressure for the cases 1, 2 and 6, where the author gives temperatures, shows that the observed pressure agrees with the difference of density due to temperatures. The calculated pressures do not allow for friction in chimney, but for the greatest velocity. This friction would only make about $\frac{1}{32}$ of an inch difference—too little to observe with an ordinary U tube.

If we knew the details of the flues leading to his stacks, we might show that the chimney heights agreed fairly well with the formula for height in *Rankine*:

$$n = \frac{n^2}{29} \quad \left[G + 1 + \frac{fl}{m} \right]$$

In making 1 equal his chimney height, only [instead of all the flues leading to the stack as it should be] the calculated heights are as per last column, Table II.—all on the right side of the actual height.

In the case of the "Orange," I show, page 395, Vol. XI., that the pressure observed was 0.33 of water, and the calculated 0.37 which must be discounted for friction in *chimney*. Also, that where we *know* the air supply and the nature of the flues, the calculated chimney height was 61 feet against 54. In the "Bergen" the calculated height, page 423, is 59 versus 53 for actual height. If the facts were *all* observed in the case of Mr. Crane's chimneys, I think theory would acquit itself quite well enough to deserve respect.

TABLE I.

Calculation of the "head in feet in height of a column of hot gas in the chimney" by Rankine's Formula.* Also corresponding draught in inches of water.

No. of chimney.	Height in ft.	Dimensions of cross section.		Grate surface in sq. ft.	Rate of combustion per sq. ft. of grate per hour.	Temp. of gases.	Draught in ins. of water.		Calculated head in ft. of hot gas.
		Dimensions.	Area in sq. ft. = A.				Observed.	Calculated.	
1	130	4 ft. drain	12.57	76.5	14	340	.55	.62	58
2	90	28″ × 28″	5.44	19.5	25	775	.77	.72	112
6	68	3½ × 4 ft. diam.	Minimum 9.6	51.0	12	375	.35	.36	85

$$* \ h = H \left(0.96 \frac{\tau_1}{\tau_2} - 1 \right)$$

Corresponding head in inches of water $= h \times .161 \frac{\tau_0}{\tau_1}$

TABLE II.

Calculated Height of Chimneys, if the Grate Resistances are those common in ordinary boiler practice.

No. of chimney.	Height, Cranes.	Velocity* in chimney in ft. per sec.	Hydraulic mean depth of chimney.	Factor of resistance for friction of the flue $= \frac{Fl}{m}$	Heat of hot gas to produce velocity and overcome friction, if factor of resistance for grate is taken at 12.	$.96 \frac{\tau_1}{\tau_2} - 1$	Height of chimney = Col. 6 + Col. 7.
1	2	3	4	5	6	7	8
1	130	10.7	1.00	1.56	25.9	.45	58
2	90	17.3	.58	1.86	69.1	1.24	56
6	68	8.4	.88	.92	15.3	.51	30
Bergen	53	15.25	1.00	.64	49.2	1.10	45

* The volume of air per pound of coal is taken as 300 for all cases except the Bergen, when it is estimated to be 190.

Mr. Wm. Kent.—The force of draught is equal to the difference between the weight of the column of hot gas inside the chimney and the weight of a column of cool air of the same height outside the chimney.

If D is the density of the air outside, d the density of the hot gas inside, in pounds per cubic feet, h the height of the chimney in feet, and .192 the factor for converting pressure in pounds per square foot into inches of water column, then the formula for the force of draught expressed in inches of water is, $F = .192\ h\ (D\text{-}d)$ The density varies with the temperature. (See *Rankine, Steam Engine*, p. 287.) $d = \frac{\tau_0}{\tau_1} 0.084$; $D = 0.0807 \frac{\tau_0}{\tau_2}$ (for 30 inches barometer), where τ_0 is the absolute temperature at 332° F. $= 493.2$; τ_1 the absolute temperature of the hot gas, and τ_2 that of the external air. Substituting these values, the formula for force of draught becomes $F = .192\ h \left(\frac{39.8}{\tau_2} - \frac{41.43}{\tau_1}\right)$.

Mr. Crane makes one statement of a general case, viz.: that a stack 90 feet high will give a draught of .75 to .80 inch when the temperature of the gases is 800°. He also gives observations of the draught of twelve different chimneys, but in only three of these does he know the temperature of the gases, viz.: Nos. 1 and 2 on second page, and No. 5 on third page of his paper. In these three cases only are there enough data given to know whether they present any idiosyncrasies or not, or whether or not they conform to any formula. Tabulating Mr. Crane's general case and his three chimneys, we have the following, assuming the external air to be at 70° F.:

Chimney.	Height.	Temperature.	τ_1	τ_2	Draught Calculated by Formula.	Draught Observed by Mr. Crane.
1	130	340	801	531	.58	.55
2	90	775	1236	531	.72	.77
3	68	375	836	531	.33	.35
General	90	800	861	531	.73	.75 to 80

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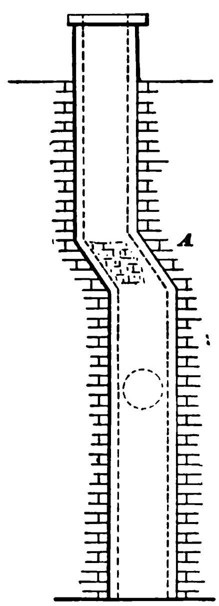

FIG. 49.

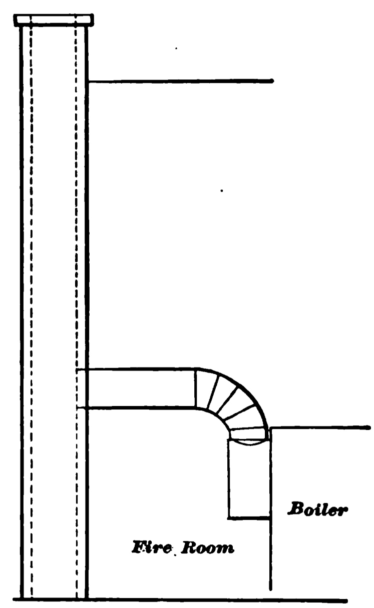

FIG. 50.

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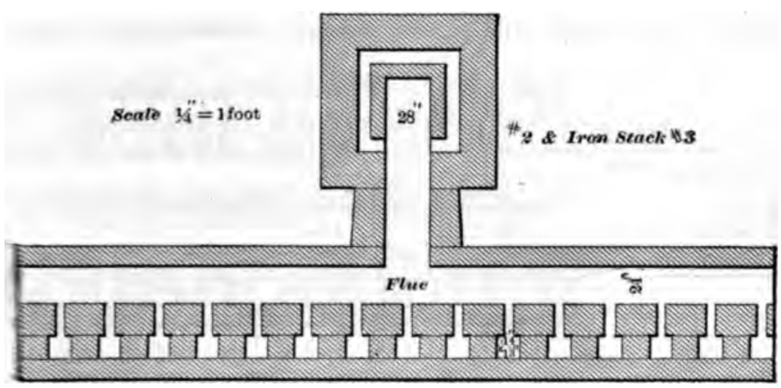

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Fig. 51 shows the arrangement in the case of chimney No. 2 on the first series, and on the second one—the iron stack No. 3, .85 of an inch. In the case No. 4, the flue comes in as shown in Fig. 52.

Replying to the criticisms on the paper, "Idiosyncracies of Chimney Draught," it appears to be generally conceded that 600° is not the limit of efficiency of temperature to produce draught in chimney, and this much being conceded, the title of the paper might be more appropriately termed, "Observations of temperatures and draught in chimneys."

Engineers should bear in mind that numerous diagrams and charts that are in existence showing the draught on a straight line after the temperature has reached 600°, are of no account and must be replotted.

If the limit was 600°, the highest draught which would be possi-

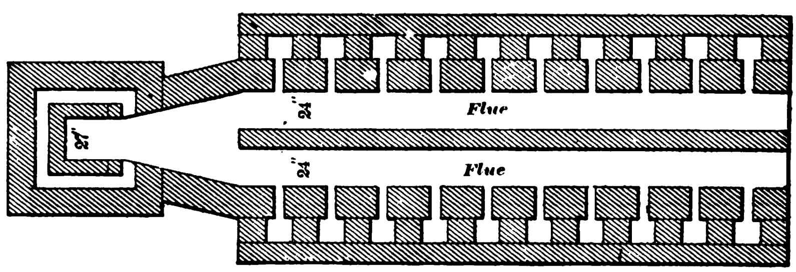

FIG. 52.

ble in a 90-foot chimney would be .64 of an inch, while in these chimneys with high temperatures the draught is from .75 to .85 of an inch, with external air at 65°.

These results correspond very closely with the experiments of Prof. Thurston, with the *Treatise on Heat* by Thomas Box, and the paper by Prof. Gale.

For some reasons I should have preferred not to have presented this paper until further investigation had been made. My object in starting it was to go through a whole series, looking not only at the chimneys, but also at the effect which a long flue, or a flue which is irregular in shape, had upon draught in furnaces; also, to find the temperatures at different points in the flue where we have long ones, and see whether straight ones had a different effect from those that were irregular, and I had hoped to have all of this together. For this I did not have time, and so put in what few data I had.

CCCCLXXVII.

TOPICAL QUERIES AND INTERCHANGE OF DATA.

XXIVth Meeting, November, 1891.

No. 477–89.

Has any one ever tried to standardize sizes for keys? If so, what are his sizes?

Mr. Wm. Kent.—I received a circular recently from the Sandwich Manufacturing Company, of Sandwich, Illinois, showing some of the standard sizes of keys that they are making. I do not recollect what the sizes were, or how they were standardized, but any one interested in the subject can by writing to that concern get the information.

Mr. E. G. Parkhurst.—In 1858 I adopted a standard for the size of keys, as follows: $\frac{1}{4}$ the diameter of the shaft for the width of key; $\frac{1}{8}$ of the diameter for the depth of key in shaft; taper of key, $\frac{1}{8}''$ to the foot. I was led to adopt this system after having had quite an extended experience in mill-gearing and hoisting apparatus for mining and quarrying purposes. Facts in the case have substantiated my conclusions in this matter, as I do not remember a single failure after having adopted the said system as a standard.

Several years ago, Mr. Joshua Rose was gathering shop statistics and I gave him my formula. He published the same, gave my name as author. A little time afterward, I was severely taken to task by some one—have forgotten who, or in what mechanical publication—and given quite a raking on the key question.

A few months later, Mr. Thomas Shanks, of the firm of Thomas Shanks & Co., of Johnstone, Scotland, makers of (extra heavy) machine tools, visited our works and we exchanged shop formulæ. Among them was their standard for keys, and much to my surprise it was the same as I had adopted years before.

I am free to say that I have not had much exchange of opinion with engineers on the subject, having for the last twenty years been almost out of a business which required any particular attention to the subject.

It was upon the cut and try plan that my knowledge was gained on which to base the formula; simply kept on increasing the width and depth until I reached a point where the key would hold enough to twist off a shaft, then stopped. I have used the said system ever since whenever the driving of machinery depended on keys.

Mr. Geo. L. Fowler.—In saw-mill work in Michigan that same standard is used almost entirely, except that there are no keys made except in sixteenths of an inch measurements. What I mean by that is that if we take a $\frac{15}{16}$-inch shaft we put a quarter inch key into it, and the key is square in every instance. The taper is somewhat less—I think one-sixteenth of an inch to the foot. But that is something which has not really been standardized, and every shafting maker there does about as he chooses. But it is generally understood, if a man sends in for a line of shafting, or for a pulley to be bored out and key-seated, that the key is to be cut one-quarter the diameter of the shaft, and that the key must be an even sixteenth of an inch square. All pulleys there are keyed to the shaft. I do not think I ever saw in any one of those Michigan saw-mills a pulley which was held to the shaft with set screws.

Mr. Jno. T. Hawkins.—While I do not know of any attempt having been made to establish a standard for keys, I am disposed to take exception to making such a standard—should a standard be adopted—as has been mentioned for the width of the key. I have had a good deal of experience in the use of keys and for a good many years, and I know of concerns which use them very largely, who have adopted as a standard of their own a width equal to one-third the diameter of the hole. It is pretty well known, I think, to engineers who use keys to any great extent that they generally hold better through good top and bottom fits than from side fitting; that a narrow key has a tendency to roll and imbed its corners into the shaft, and that a key flatted on and without being let into the shaft whose width equals one-third of the diameter of the hole, will hold better and do more severe work if well fitted top and bottom, than a key which is let into the shaft one-ninth of the diameter of the hole for its depth, and is only one-fourth of the diameter of the hole in width. For myself, in all ordinary cases where the work is severe, I would have keys made not less than one-third the diameter of the hole for the width.

I think that the amount which a key is let into the shaft is of little or no consequence, for the reason that if let in to such a depth as to provide a sufficient abutment of iron to withstand the crushing stress, the proportion of the key, if similarly let into the hub of the member which it is to drive, will have the same faulty proportion which characterizes a too narrow key; to say nothing of the fact that the nearer the bottom flat of the key approaches the centre of the shaft, the more it weakens the shaft, and the greater the leverage upon the key and shaft becomes. A side abutment in the key-seat in a shaft equal to one-ninth of the diameter of the hole is easily crushed, and certainly is not calculated to stand up against any very severe strains, particularly if the key be narrow. Very few, I think, realize the enormity of the stress brought upon such an abutment in ordinarily severe cases. This is aggravated in the tendency of a narrow key to roll, and thus to bring the stress upon the outer edge of the abutment only.

I believe that for the average case a better standard would be one-third the diameter of the hole for the width, and one-eighth of the diameter of the hole for the depth of the side abutment in the shaft.

Mr. Wm. O. Webber.—Herewith is a table and diagram of standard sizes of keys and key-ways, and our formula for obtaining the same. This we have had in practice for the past two years and find it to be very satisfactory in every way. (Fig. 54.)

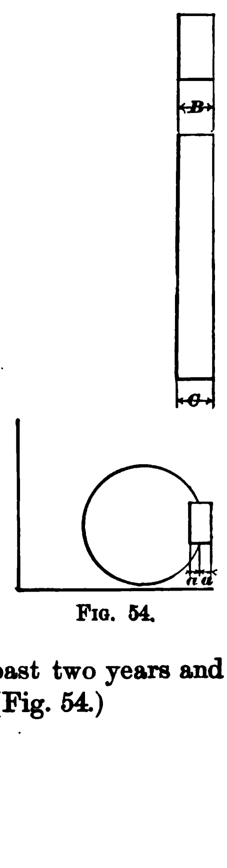

FIG. 54.

Size.	Style.	Width of key.	Small end depth at *A*.	Diameter of shaft.	Thickness at *B*.	Thickness at *C*.	Length of key.	Length of pulley hub.	Dia. of pulley for det. engines.	Face of pulleys for det. engines.	Diameter of hub.	Diameter of pulleys.
6×9	Det.	5/8	1/8	2½	1/4	11/32	7½	5½	36″ / 16	9½″ / 10½	4½″	
7×10	"	5/8	1/8	2⅝	1/4	3/8 + 3/64	8	6	44 / 20	10½ / 10½	5½	
8×10	"	3/4	5/32	2⅞	5/16	7/16 + 3/64	8	6	44 / 20	12½ / 10½	5½	
8×12	"	3/4	5/32	3	5/16	1/2	9	7	48 / 32	12½ / 10½	6	
9×12	"	7/8	3/16	3¼	3/8	9/16 + 3/64	9½	7½	54 / 32	12½ / 10½	6½	
10×12	"	7/8	3/16	3½	3/8	9/16 + 3/32	10½	8½	54 / 32	12½ / 10½	7¼	
10×15	D. & S.	1	7/32	3¾	7/16	5/8 + 3/32	10½	8½	60 / 36	14½ / 10½	7½	72 / 42
10×20	Sta.	1⅛	1/4	4½	1/2	11/16 + 3/64	11½	9½			9½	96 / 48
11×15	D. & S.	1⅛	1/4	4¾	1/2	11/16 + 3/64		9	72 / 36	14½ / 10½	8½	78 / 42
12×16	"	1¼	1/4	4⅞	1/2	3/4	12	10	72 / 36	16½ / 10½	9½	78 / 42
12×18	Sta.	1¼	1/4	4⅞	1/2	3/4 + 3/64	14	12			9½	96 / 48
12×24	"	1⅜	9/32	5½	9/16	13/16 + 3/64	14	12			10½	96 / 60
14×16	D. & S.	1⅜	9/32	5⅜	9/16	13/16 + 3/64	14	12			10½	
14×18	Sta.	1⅜	9/32	5½	9/16	13/16 + 3/64	14	12			10½	96 / 48
14×30	"	1½	11/32	6¼	11/16	15/16 + 3/64	14½	12½			13	120 / 60
16×18	D. & S.	1½	5/16	5⅞	5/8	15/16 + 1/64	16	14			12	
16×24	Sta.	1⅝	11/32	6½	11/16	1 + 1/64	15½	13½			13	96 / 60
18×24	"	1¾	3/8	7	3/4	1 1/16 + 1/64	15½	13½			14	96 / 60
20×24	"	2	7/16	8	7/8	1 3/16 + 3/64	17	15			16	96 / 60
20×30	"	2	7/16	8	7/8	1 3/16 + 3/64	17	15			16	120 / 60

WEIGHT AND DIAMETER OF FLY WHEELS.

6×9.	7×10.	8×10.	8×12.	9×12.	10×12.	10×15.	10×20.	11×15.	12×15.	12×18.	12×24.	14×16.	14×18.	14×30.	16×18.	16×24.	18×24.	20×24.	20×30.
....	....	....	....	....	....	....	...	....	....	2000	2420	2000	2500	....	....	3100	3600	4100	4600
....	....	....	....	....	....	...	...	....	....	108	110	108	110	....	...	120	120	120	120

Constant .0562 × diameter of shaft = depth at *A*.
.020833 = constant for 1″ length of taper of key.
Width of key = ¼″ for every diameter of shaft.
Length of pulley hub = length of journal.
Length of hub + 2″ = length of key.

	6×9.	7×10.	8×10	8×12.	9×12.	10×12.	10×15.	10×20.	11×15.	12×16.	12×18.	12×24.	14×16.	14×18.	14×30.	16×18.	16×24.	18×24.	20×24.	20×30.
Weight of small pulley for det. engine	303	342	342	..	650	..	..	..	..	..	....	....	1270	..	..	..	....	....	....	....
Weight of large pulley for det. engine	93	120	125	..	218	..	..	..	..	..	...	....	526	....	..	..	....	....	....	....
Weight of small pulley for sta. engine	...	...	...	..	...	..	..	..	..	..	....	....	....	....	..	..	....	....	...	....
Weight of large pulley for sta. engine	...	...	...	..	...	..	..	..	..	..	1720	2500	1600	2500	..	..	3400	3500	3500	5200

Mr. Frank H. Ball.—I would like to ask the members of the society if any of them ever tried using keys of the form illustrated in the sketch, Fig. 53. I have tried it with very good success.

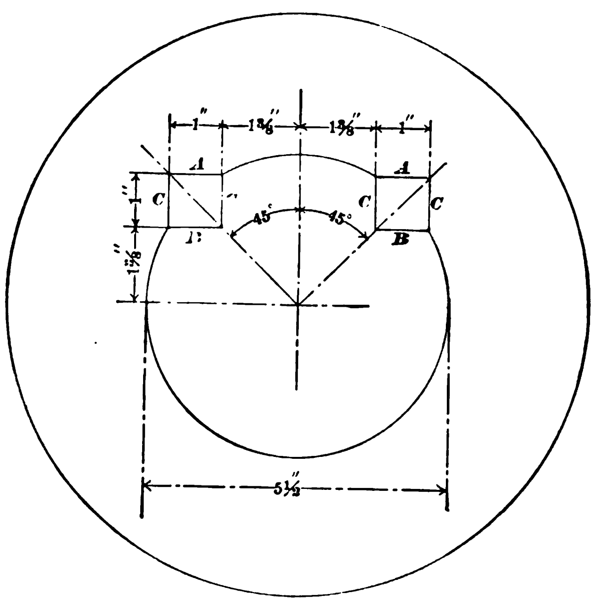

FIG. 53.

Mr. Geo. M. Bond.—I might say that there is a special system of small keys adopted at our works in which the depth consideration, so far as the shaft is concerned, is overcome by dropping a circular milling cut into the stem or shaft, and a key cut from a bar of semi-circular section fitted in tightly. In this way the side strain is resisted by a greater depth of contact in the shaft, while a considerably narrower width of key is safely used. These keys, by giving them a slight inclination longitudinally, by rotating in the receiving slot, are given the taper necessary to fit the top of the key-way in the hub of the pulley or gear, and in this way the key may be said to adjust itself to any position required of it.

This method is called the "Woodruff Keying System." I do not know that the makers have a positive or definite formula for the width, but they have obtained practically the data for the different widths and sizes by testing in a torsion machine, to the point at which the key would stand the maximum torsional strain of the shafting without shearing the key. It is curious to note that the milling cuts to receive the keys in this system are very much thinner than any of the older formulæ would seem to recommend as necessary. This is probably due to the absence of a tendency to "careen" under strain, and requires a direct shearing strain before any possible rupture can occur. I would say further that the form of key mentioned is mainly applicable to short hubs, or in cases where the key is not required to be long. Of course it can be doubled up or lengthened even more by dropping in these milling cuts consecutively, and inserting the half-round key sections in line. By placing them on opposite sides of the shaft or stud and used as a "feather key" extra strength of key is obtained, and for a sliding hub is unequalled by any of the older ways of keying. These keys are used for gears on the ends of studs or shafts for short hub pulleys and work of this class generally.

Mr. W. S. Rogers.—I like the standard given for this reason: I have been asked four times within the last month by men working in the shop what my standard was for keys, and, not having any, I made an effort to find out if there was a standard anywhere, and did not find it. I have made a memorandum here with the intention of writing to two or three of those boys in the shop in regard to this standard. There is one thing here I do not like about it, which is the diameter of the hole by the width. Take a $2\frac{7}{16}$ shaft or $2\frac{15}{16}$. Now the fellow in an ordinary job shop has got a rule laid off by eighths and sixteenths, and if he has got to figure out $2\frac{7}{16}$ and work it into ninths he will get very badly demoralized. Otherwise I think it a good proportion.

Mr. W. S. Huson.—In trying to standardize keys I followed very closely to the formula given, but I arranged the keys in sixteenths, thus averting the difficulty just mentioned.

Thus, $\frac{1}{4}$ inch key for 1 inch to $1\frac{1}{4}$ inch shafts; $\frac{5}{16}$ inch, $1\frac{1}{4}$

inches to $1\frac{1}{2}$ inches; $\frac{3}{8}$ inch, $1\frac{1}{2}$ to $1\frac{3}{4}$ inches, and so on until two keys become necessary. Taper $\frac{1}{8}$ inch to foot. The total thickness at large end of spline, $\frac{4}{5}$ of the width of key.

Mr. Geo. L. Fowler.—Our practice in the West in regard to this matter was to have the key-way in the hub of the pulley one-half the edge of the key measured at the edge, so that the shaft was cut out a trifle more on the centre line than it would be in the pulley, making it easier to the workman to measure.

The taper of the key is between the sides marked "A" and "B," with the sides "C" made parallel, and no attempt to get a bearing on these sides "C."

The loosening of a wheel on a shaft is generally the result of a slight turning on the shaft an innumerable number of times, each time of greater amplitude, due to the wearing away of the surfaces in contact.

With the arrangement shown, the likelihood of this first small beginning of motion is reduced, because it cannot occur without crushing the metal between "A" and "B," and the angle of resistance of these surfaces is much better than with a single key of ordinary width.

Mr. Roelker.—I used to make a great many things like that for heavy marine work, but there was a central key besides.

No. 477–90.

"Have you had any experience in out-door work at night in wind and storm, with portable apparatus for light in large quantities, other than electric light?"

Mr. W. W. Dingee.—The steamboats plying the Mississippi River and tributaries have always used at their landings a broom-shaped torch made of hemp and dipped in tar, and when set aflame it gave a great light and made a prodigious smoke. This invention was never patented nor is it known to whom the honor belongs. No one has been able in the past half-century to either improve or displace it. The songs of the negro boat hands, with many other things peculiar to the navigation of western rivers, are things of the past, but as a means of converting a limited area of darkness into day the flambeau stands without a rival.

Mr. Wm. Kent.—In the work of putting down the cable railway in Broadway, New York, the contractors are using the Wells light, which gives a great brilliance. The light is generated from some petroleum compound contained in a tank, into which air is injected from a receiver in the tank which contains the air at high pressure. The air is charged with the vapors of petroleum and it comes out with great force, making a brilliant light. It is very largely used, I believe, in England. The "Lucigen" is another form of light of a similar kind.

Mr. W. S. Rogers.—I have seen that light Mr. Kent speaks of, but I was always under the impression that naphtha was used in it with air pressure. I have seen it used for clearing up wrecks on railroads, and I do not know of anything which could be better adapted to the purpose, that is, for work requiring a great deal of light. I have seen torches used and flambeaux somewhat similar to the Mississippi River device, but I think the nicest and best I have ever seen is the Wells light for that purpose.

The President.—Of course in fixed plants the electric lights have become so common now that nothing else will satisfy us. But there are so many places where a temporary light is required that a good one is of great importance.

Mr. Robert Cartwright.—Of course the Wells light and the Lucigen light are good things. I have used this last year and a half, working night and day, the electric light and what we call the "Dago" light; that is, the light which you see these peanut venders have on the corner of the streets, which burn plain common kerosene. The mosquitoes and the Canada soldiers had such a bad effect on the arc light that we had to fall back on the Dago light, which burns them up.

The President.—You ought to tell us now what a Canada soldier is.

Mr. Cartwright.—A Canada soldier is a gad-fly; some call it the shad-fly. They actually come in such quantities that they will fill the globe of an arc light within two or three hours.

No. 477-91.

"What is the best design for line shafting transmitting over fifty horse power, permitting them to be stopped and started on any floor, without interfering with the motor or other shafting?"

Mr. W. W. Dingee.—Where loose pulleys are used for stopping shafting much of the annoyance incident to their use can be avoided by having the pair of pulleys which run idle less in diameter than those which drive the shaft, thus relieving the strain of belt and the pressure under which the loose pulley runs. The difference in diameter can be provided for by a gradual incline up which the belt can easily be shipped.

Friction pulleys when they can be large enough for the work required furnish a very satisfactory means of stopping and starting shafting. When at work the shafting is in line, and but little movement is required to throw them out of contact.

When shafting is driven on upper floors from below, a properly arranged tightener can be used for stopping and starting. The objection to this method is that it is running an idler all the time for the sake of stopping when occasion requires.

Mr. Wm. Kent—Those who are interested in this question can see an example of it in the Delavergne Refrigerating Works when they visit them to-morrow.

Mr. Geo. M. Bond.—I think a well-designed friction clutch would be as efficient as anything. In the building for the small-tool department lately added to our shop we have about seventy-five horse power transmitted through belts to the upper floors and arranged so as to be disengaged by the simple operation of a lever which can be detached and set up in the corner so as to be out of the way. Whenever the power is to be disconnected the friction is thrown out and there is no disturbance of any kind—no getting out of line and no undue friction while the belts are idle.

Mr. Jas. McBride.—I have a number of large machines of fifty horse power and upwards with which I use tighteners exclusively for stopping and starting. I think, of all the mechanical abominations of which I know, a loose pulley on a shaft with shifting belt is the worst. I use those belts eighteen or twenty inches wide. The distance between the centres of the shafts is about eighteen feet, one very nearly over the other.

A Member.—One trouble with friction clutches in connection with such matters oftentimes is the difficulty of getting the lubricant in where it ought to be. If the clutch is stopped much of the time, the portion of the pulley which revolves on the shaft is apt to be not lubricated. I found that that is very satisfactorily overcome in many cases in this way: Instead of using ordinary brass bushings use a graphite filled bushing. By that means it will work for a year or more practically without any oil. Otherwise I have had a great deal of trouble with their cutting. Where belts are wide it is of course almost impossible to shift them on loose pulleys. In that case the friction clutch works very satisfactorily. When it comes to the question of electric light stations, places of that sort where there are a great many pulleys to be taken care of, if the friction clutches hang on the shafts they are apt to make a very large percentage of friction. I had one case a short time ago where in the test of a plant tried on a day load where there was very light work the friction amounted to forty-four horse power and the actual work amounted to twelve. Of course in the reports relative to the test it was advised that the shafting be entirely stopped and a small engine put in to drive the day load direct. That has been done and undoubtedly the results are perfectly satisfactory. But that question of lubricating friction clutches has been quite an important one, which I found has been overcome very nicely by these graphite bushings. Mr. Cooper, I believe, has to do with a lubricant which is perhaps of the graphite order, although I do not know exactly what it is composed of, which he claims is very desirable indeed, and I think the Link Belt Engineering Company, of Philadelphia, have taken up the subject. I have not had experience with that at all, but I know that the graphite is good if properly put in.

Mr. M. W. Sewall.—I should like to ask if any one present knows of the use of a loose pulley capable of transmitting a thousand horse-power at sixty revolutions per minute.

I know of machinery under construction in which such a pulley is to be used. It is to stand idle on a running shaft continuously for a long time each day. I should like to know if other members think the same of such use of it as I do.

Mr. Robert Cartwright.—I am one who has received his diploma in the school of adversity. I want to understand whether or not we are trying to formulate everything so that it can be accomplished by rule, so that all you have got to do is to study those rules and then you are a perfect engineer. My idea of engineering is that every case requires a different prescription. Now clutches are very good in some places, and some clutches are better than others for some places. The Frisbie clutch is a good clutch for some cases and not worth a rush in others. The Hunter clutch is a good clutch in some places. The shaft running loose requires lubrication, but it requires much more lubrication in a dirty rolling mill than in an electric light station. I have just had to do with a clutch transmitting 500 horse-power which was very nice to throw in, but you couldn't throw it out when it got in there. Now if anybody has a friction clutch which will answer all purposes and run in any place they have got a good thing, and I would like to get hold of it. The Weston clutch is a good clutch. We know what it is capable of, but after it has been in operation a while you cannot throw it out. Now we have got to apply our knowledge to the circumstances in different places. What is good practice to-day is poor to-morrow, and *vice versa,* and no one rule applies to all of these instances. I have been forty years in the harness and have seen a good many things tried. But as to the idea of making a law with these questions to apply to everything—you cannot do it.

The President.—That is quite true. But do you not think that as the members give their experience under certain conditions they give us valuable information which may keep you and me from going through their disastrous experience? As I take it, that is the idea and the purport of these topical questions. What we want is to avoid those dangers and get our diplomas from the school of success. With your permission, I think that is the idea of these questions. You tell us of your clutch which will go in and not go out. Now we will not put that in, you know. You have given us a contribution.

Mr. Cartwright.—It would be the same with any kind of clutch. There is no clutch which will cover the ground in all cases. It asks here what is the best design for line shafting transmitting over fifty horse-power, permitting them to be stopped or started on any floor without interfering with the motor or other shafting. You might go and look at it and say one thing. I might go and look at it and say another. There is no one law which will apply to anything of that kind. Then, in transmitting over fifty horse-power, are we to have one for fifty and one for sixty, and so on?

Mr. McBride.—As Mr. Cartwright is not in love with the clutch, I would suggest to him to try the belt and tightening pulleys.

Mr. Cartwright.—Probably some have had experience with the belts and tighteners; I have had. Now for a power hammer with an intermittent motion, there is nothing which equals the idler dropping on to the belt, in my estimation, because you can put your hammer in motion slowly and develop your power as you need it. But with an electric light plant there are a great many considerations to be thought of. In one with which I have had a good deal to do there are some thirty odd dynamos. Every one of them is set up with an idler dropping on to the belt—that is, gradually put on to the belt. We thought we could improve on that, and we put in this 500 horse clutch which clutched but did not unclutch. We had to stop and take the whole thing apart.

Mr. Chas. E. Emery.—The question as written seems to answer itself, though the discussion has taken a wide range. There seems but one way to do what is asked, viz., to transmit 50 horse-power, and permit the line of shafting on each floor to be stopped and started without interference with the motor. The only desirable arrangement which the conditions permit is the very customary one of running belts from floor to floor connecting pulleys on short shafts, to be kept in motion by the motor all the time the latter is running. The line shafts will naturally be in line with the short shafts, and, to fulfil the conditions, be connected at will therewith by friction clutches, as such a clutch is the only known device by which one shaft may be connected to another in line when one is revolving rapidly. Each short shaft can, of course, be connected with the line shaft on each floor by belting with either fast or loose pulleys or friction pulleys, but such an arrangement will rarely be desirable.

I will say in regard to the general question that in 1869, when superintendent of the American Institute Fair, the problem was presented of connecting all the engines to the line shaft while but one was operating the same. This was accomplished by belting each engine to a large pulley ordinarily running loose on the line shaft, but which could be connected thereto at any time by a clutch. A different engine was connected up each day and a ticket was hung on the engine which was driving the machinery that day, thereby distinguishing the engine which was doing the work from those which seemed to be assisting in doing it.

No. 477–92.

In arranging chimney-stacks for a battery of boilers, is it best to use one for each boiler or pair of boilers, or to use one larger chimney for the entire battery?

Mr. Charles T. Porter.—Some years ago I had my attention called in a forcible manner to the frequent difference in efficiency between those boilers in a battery which are nearest to, and those which are farthest from, the stack.

An arrangement has occurred to me which, where the space required is available, seems calculated to mend matters in this respect. Each one of the eight boilers, in the accompanying sketch (Fig. 55), must have the same draught, and that the best draught that the chimney can give. Four or six boilers could be arranged in a similar manner.

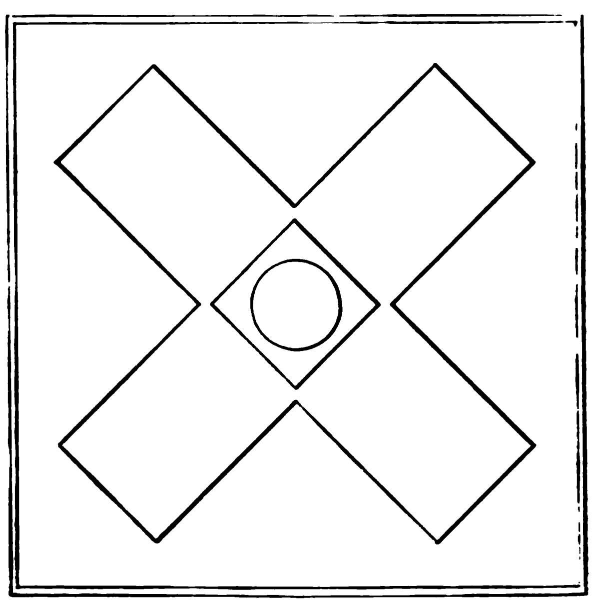

FIG. 55.

The greater cost of setting and of the building would be repaid in a few months in coal saved.

A large extent of wall surface is presented for the percolation of cold air. The brick surfaces, sides, and top ought, however, in all boilers, to be so treated that this percolation is impossible.

Of course, that is a plan that has never been tested, and I only threw out the suggestion as one possibly worthy of consideration in the arrangement of boilers, to give to each boiler an equal as well as the shortest possible horizontal flue.

Mr. Wm. Kent.—My own experience confirms that stated by Mr. Porter, that where several boilers lead into one large chimney it is very apt to be the case that only one or two of the boilers have a proper draught Generally those boilers nearest been almost out of a business which required any particular attention to the subject.

It was upon the cut and try plan that my knowledge was gained on which to base the formula; simply kept on increasing the width and depth until I reached a point where the key would hold enough to twist off a shaft, then stopped. I have used the said system ever since whenever the driving of machinery depended on keys.

Mr. Geo. L. Fowler.—In saw-mill work in Michigan that same standard is used almost entirely, except that there are no keys made except in sixteenths of an inch measurements. What I mean by that is that if we take a $1\frac{5}{16}$-inch shaft we put a quarter inch key into it, and the key is square in every instance. The taper is somewhat less—I think one-sixteenth of an inch to the foot. But that is something which has not really been standardized, and every shafting maker there does about as he chooses. But it is generally understood, if a man sends in for a line of shafting, or for a pulley to be bored out and key-seated, that the key is to be cut one-quarter the diameter of the shaft, and that the key must be an even sixteenth of an inch square. All pulleys there are keyed to the shaft. I do not think I ever saw in any one of those Michigan saw-mills a pulley which was held to the shaft with set screws.

Mr. Jno. T. Hawkins.—While I do not know of any attempt having been made to establish a standard for keys, I am disposed to take exception to making such a standard—should a standard be adopted—as has been mentioned for the width of the key. I have had a good deal of experience in the use of keys and for a good many years, and I know of concerns which use them very largely, who have adopted as a standard of their own a width equal to one-third the diameter of the hole. It is pretty well known, I think, to engineers who use keys to any great extent that they generally hold better through good top and bottom fits than from side fitting; that a narrow key has a tendency to roll and imbed its corners into the shaft, and that a key flatted on and without being let into the shaft whose width equals one-third of the diameter of the hole, will hold better and do more severe work if well fitted top and bottom, than a key which is let into the shaft one-ninth of the diameter of the hole for its depth, and is only one-fourth of the diameter of the hole in width. For myself, in all ordinary cases where the work is severe, I would have keys made not less than one-third the diameter of the hole for the width.

I think that the amount which a key is let into the shaft is of little or no consequence, for the reason that if let in to such a depth as to provide a sufficient abutment of iron to withstand the crushing stress, the proportion of the key, if similarly let into the hub of the member which it is to drive, will have the same faulty proportion which characterizes a too narrow key; to say nothing of the fact that the nearer the bottom flat of the key approaches the centre of the shaft, the more it weakens the shaft, and the greater the leverage upon the key and shaft becomes. A side abutment in the key-seat in a shaft equal to one-ninth of the diameter of the hole is easily crushed, and certainly is not calculated to stand up against any very severe strains, particularly if the key be narrow. Very few, I think, realize the enormity of the stress brought upon such an abutment in ordinarily severe cases. This is aggravated in the tendency of a narrow key to roll, and thus to bring the stress upon the outer edge of the abutment only.

I believe that for the average case a better standard would be one-third the diameter of the hole for the width, and one-eighth of the diameter of the hole for the depth of the side abutment in the shaft.

Mr. Wm. O. Webber.—Herewith is a table and diagram of standard sizes of keys and key-ways, and our formula for obtaining the same. This we have had in practice for the past two years and find it to be very satisfactory in every way. (Fig. 54.)

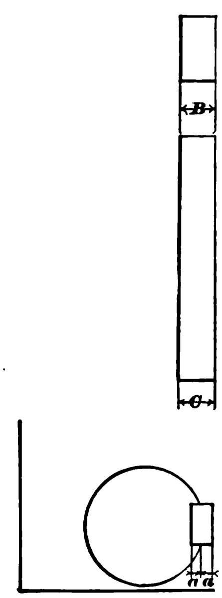

FIG. 54.

Size.	Style.	Width of key.	Small end depth at *A*.	Diameter of shaft.	Thickness at *B*.	Thickness at *C*.	Length of key.	Length of pulley hub.	Dia. of pulley for det. engines.	Face of pulleys for det. engines.	Diameter of hub.	Diameter of pulleys.
6 × 9	Det.	5/8	1/8	2 3/8	1/4	13/32	7 1/4	5 1/4	36″ / 16	9 1/2″ / 10 1/2	4 3/4″	
7 × 10	"	5/8	1/8	2 5/8	1/4	3/8 + 3/64	8	6	44 / 20	10 1/2 / 10 1/2	5 1/4	
8 × 10	"	3/4	5/32	2 7/8	5/16	7/16 + 3/64	8	6	44 / 20	12 1/2 / 10 1/2	5 1/2	
8 × 12	"	3/4	5/32	3	5/16	1/2	9	7	48 / 32	12 1/2 / 10 1/2	6	
9 × 12	"	7/8	3/16	3 3/8	3/8	9/16 + 3/64	9 1/2	7 1/2	54 / 32	12 1/2 / 10 1/2	6 3/4	
10 × 12	"	7/8	3/16	3 3/8	3/8	9/16 + 1/32	10 1/4	8 1/4	54 / 32	12 1/2 / 10 1/2	7 1/4	
10 × 15	D. & S.	1	7/32	3 7/8	7/16	5/8 + 1/32	10 1/4	8 1/4	60 / 36	14 1/2 / 10 1/2	7 3/4	72 / 42
10 × 20	Sta.	1 1/8	1/4	4 3/8	1/2	11/16 + 3/64	11 1/4	9 1/4			9 1/2	96 / 48
11 × 15	D. & S.	1 1/8	1/4	4 3/8	1/2	11/16 + 3/64	11	9	72 / 36	14 1/2 / 10 1/2	8 3/4	78 / 42
12 × 16	"	1 1/4	1/4	4 7/8	1/2	3/4	12	10	72 / 36	16 1/2 / 10 1/2	9 3/4	78 / 42
12 × 18	Sta.	1 1/4	1/4	4 7/8	1/2	3/4 + 3/64	14	12			9 3/4	96 / 48
12 × 24	"	1 3/8	9/32	5 3/8	9/16	13/16 + 3/64	14	12			10 3/4	96 / 60
14 × 16	D. & S.	1 3/8	9/32	5 3/8	9/16	13/16 + 3/64	14	12			10 3/4	
14 × 18	Sta.	1 3/8	9/32	5 3/8	9/16	13/16 + 3/64	14	12			10 3/4	96 / 48
14 × 30	"	1 1/2	11/32	6 1/4	11/16	15/16 + 3/64	14 1/2	12 1/2			13	120 / 60
16 × 18	D. & S.	1 1/2	5/16	5 7/8	5/8	15/16 + 1/64	16	14			12	
16 × 24	Sta.	1 5/8	11/32	6 3/4	11/16	1 + 1/64	15 1/2	13 1/2			13	96 / 60
18 × 24	"	1 3/4	3/8	7	3/4	1 1/16 + 1/64	15 1/2	13 1/2			14	96 / 60
20 × 24	"	2	7/16	8	7/8	1 3/16 + 3/64	17	15			16	96 / 60
20 × 30	"	2	7/16	8	7/8	1 3/16 + 3/64	17	15			16	120 / 60

WEIGHT AND DIAMETER OF FLY WHEELS.

6 × 9.	7 × 10.	8 × 10.	8 × 12.	9 × 12.	10 × 12.	10 × 15.	10 × 20.	11 × 15.	12 × 15.	12 × 18.	12 × 24.	14 × 16.	14 × 18.	14 × 30.	16 × 18.	16 × 24.	18 × 24.	20 × 24.	20 × 30.
....	....	....	....	....	....	....	....	....	....	2000	2420	2000	2500	....	....	3100	3600	4100	4600
....	....	....	....	....	....	....	....	....	....	108	110	108	110	....	....	120	120	120	120

Constant .0562 × diameter of shaft = depth at *A*.
.020833 = constant for 1″ length of taper of key.
Width of key = 1/4″ for every diameter of shaft.
Length of pulley hub = length of journal.
Length of hub + 2″ = length of key.

to one smoke-pipe, and as space on shipboard is valuable the farthest boilers had evidently the least draught. Now it is more common to put in more smoke-pipes, and connect each with a group of boilers at a reasonable distance therefrom. In the large steam station of the New York Steam Company, at Greenwich and Cortlandt streets, in this city, designed by the speaker, there was no difficulty of this kind, as the boilers are arranged in tiers, with only four on each side connected to one chimney. The flues from each group of four boilers enter on opposite sides of the chimney; but the latter is so large that no interference is detected, and it was not found necessary to put in deflectors. There is, of course, a difference in draught, due to the difference in elevation on the different floors.

Mr. Wm. O. Webber.—I think it is best to use one large chimney for any battery of boilers, for several reasons; among them being, first, the cheapness in cost of construction; the efficiency of the chimney, *i. e.*, less frictional resistance to the draught and consequently the better draught obtained in proportion to the cost; also the space occupied.

Mr. Daniel Ashworth.—In our system of steam plants at Beaver Falls, where we have twenty-eight boilers, we had adopted the system of a central stack located midway, being flanked on either side by the fourteen boilers. We do not find both at the Edgar Thomson and at this plant that the draught is decreased for the boiler farthest from the central point. We are drifting rapidly—in fact, it is becoming almost standard with us—to put up a large central stack and have the flue system leading to that. That is our intended practice.

No. 477-93.

"Why should any one cut a half-inch bolt with twelve threads to the inch? Is there any objection to the U. S. standard of thirteen threads?"

Mr. Charles T. Porter.—Some twenty-five years ago, at the Whitworth works in Manchester, I was told by Mr. Widdowson, who then was, and for many years had been, the superintendent of the tool department of those works—the tool department in contradistinction from the ordnance department—that giving to the ½-inch bolt only 12 threads was a mistake, and one which they had always regretted: that it cut away the bolt unnecessarily, and the bolt should have had 13 threads. He was frank

about that statement, and told me that it was a pity the pitch could not be changed. The way in which Mr. Whitworth fell into that error, which is the only error in the pitch of the threads in the Whitworth system, is obvious. The English do not use the $\frac{9}{16}$-inch bolt. The reduction in size is by eighths of an inch until you get down to half an inch, so that, the number of threads increasing one thread for each smaller bolt, they have thus: For the 1-inch bolt, 8 threads; for the $\frac{7}{8}$-inch bolt, 9 threads; for the $\frac{3}{4}$-inch bolt, 10 threads; for the $\frac{5}{8}$-inch bolt, 11 threads; for the $\frac{1}{2}$-inch bolt, 12 threads; for the $\frac{7}{16}$-inch bolt, 14 threads.

That seems to be a natural arrangement, although the last step is singularly abrupt, the diameter of the bolt being reduced only $\frac{1}{16}$ inch, and the number of threads increasing by two, 14 threads being given to the $\frac{7}{16}$-inch bolt. But in this country we use the $\frac{9}{16}$-inch bolt, and that cures the difficulty at once. We have for the 1-inch bolt, 8 threads; for the $\frac{7}{8}$-inch bolt, 9 threads; for the $\frac{3}{4}$-inch bolt, 10 threads; for the $\frac{5}{8}$-inch bolt, 11 threads; for the $\frac{9}{16}$-inch bolt, 12 threads; for the $\frac{1}{2}$-inch bolt, 13 threads; for the $\frac{7}{16}$-inch bolt, 14 threads.

There is no reason why this blunder of the English system should exist in this country, where we conform with what the designers of the English system wish they had done, but find too late they cannot do.

While we criticise that one defect in the Whitworth system, we should not overlook the fact that we are indebted to Mr. Whitworth for the uniform system. Mr. Whitworth found the system of screw threads in use in England in a state of chaos, every large engineering establishment having a thread peculiar to itself, and these threads differing widely, and each one maintaining its thread strenuously, so that nobody else could repair their machinery; and the problem with Mr. Whitworth was to introduce a uniform thread which would be adopted, and he told me himself that he proceeded in this way, which of course will strike every one as the common-sense method: He obtained from every leading manufacturer of machinery his system of threads. Then he averaged these, and adopted a system which was the mean of the existing practice, and so was presumably correct, and that recommended itself sufficiently to be generally adopted. But there is another thing about the English thread. How did it get such an impracticable and amazing feat-

ure as the angle of 55°? That is something which can be readily explained. Mr. Whitworth had no particular purpose of benefiting the world, but he had a particular purpose of benefiting himself. He wanted to introduce a system of threads which would be generally adopted, but which nobody else could make, and he was shrewd enough to solve that rather complicated problem quite satisfactorily for himself by adopting an angle for the thread which, he stated to the world, was also the mean of all the angles in use. But it was an angle which it was most difficult to originate, or to verify or to restore if it was lost, or to bisect with correctness so that the opposite faces of the threads would form the same angle with the normal, and so, by means of this angle of 55°, Mr. Whitworth was enabled for a long time to monopolize the manufacture of taps and dies in England. Cutting threads in lathes was little known at that time, stocks and dies were commonly used for cutting threads on bolts by hand, and the manufacture of taps and stocks and dies formed an important branch of the business of the Whitworth Company when I knew it. But we are fortunately delivered from that feature, also.

Mr. Geo. E. Whitehead.—It has always been a source of annoyance to me to have the 12 and 13 thread bolts manufactured at the same time. We have a great many customers who order ½-inch bolts, and we hardly know whether to send them 12 or 13 thread. Also, in tapping the nuts, the 12 thread and 13 thread are very apt to get mixed. I should like to see the 12 thread abolished entirely. I have asked a number of persons why they continue to use the 12 thread, but fail to get any good cause for sticking to it. It certainly makes the bolt very weak.

Mr. W. W. Dingee.—Thirteen threads are probably better than twelve for ½-inch bolts, but many old establishments cling to the latter because they started that way and do not like to change. It might be added that if nuts and bolts had always fitted each other properly, there would have been little demand for lock-nuts and washers, of which there are now a great variety.

Mr. Jno. T. Hawkins.—I would like to give a little contribution to the history of the sort of thing which Mr. Porter described as the original incentive to Mr. Whitworth to establish a uniform system of threads in England. The late Andrew Campbell, who was the inventor of the several varieties of the Campbell printing press, conceived the same unfortunate idea: that he would pre-

vent other manufacturers from supplying bolts and nuts for his machines, and he adopted a very curious system, which has, unfortunately, been allowed to exist until to-day. He thought he would adopt certain sizes of bolts which would be available to turn from the commercial sizes of iron, so that he adopted .28-inch for a bolt to be turned from $\frac{5}{16}$-inch iron, and .6-inch bolts from $\frac{5}{8}$-inch iron, .48-inch from $\frac{1}{2}$-inch iron, .72-inch from $\frac{3}{4}$-inch iron, and so on to .96-inch from 1-inch, and his machines are built to-day with all those sizes of bolts, with the number of threads in each case very near the U. S. standard.

Mr. W. O. Webber.—I do not know of any objection to cutting a $\frac{1}{2}''$ bolt with 13 threads to the inch, excepting that very few of the old engine lathes were provided with change gears which readily make 13 threads; and think this is probably the cause of adhering to the old 12-thread standard.

Mr. Oberlin Smith.—I think it was that great political engineer, the late Horace Greeley, who said that the best way to resume specie payments was "to resume;" and it seems to me that in a matter of this kind, when we see it is better to change from a bad thing to a good thing, the best way is to *do* it. I thought some years ago, when I was using 12 threads on $\frac{1}{2}''$ bolts, that it would be a very difficult thing to change, and would give a great deal of trouble. But I made up my mind that the longer I lived the worse it would grow, and the only thing to do was to change then—which I did. I believe it was the best policy, and that any one who does make such a change in his shops will certainly not regret it after a few months. After a year or so he will never think of it again. There is one thing worse than "getting the mean of the thread," and that is getting a mean thread, which a 12 per inch certainly is, on such small bolts.

No. 477–94.

Have you had any experience with systems for purification of bad feed water before it gets into a steam boiler, either by precipitation or otherwise?

Mr. F'k Merriam Wheeler.—I regret that the gentleman who suggested this topic is not here to bring out the points he desires. He speaks of "bad feed water." Now, these words can be construed mildly or otherwise. I would like to know what kind of "badness" he desires us to give consideration, and whether he has in view animal, mineral, or vegetable matter in

the feed water. I would also like to inquire if any gentleman here has had any extended experience in the matter of cleansing feed water from grease and oil. With the surface condenser system this is a trouble with which we have to contend, and it is sometimes quite a serious matter. It has been found that there is different chemical action on cylinder lubricants as used in different types of engines—that is to say, higher steam pressures and temperatures do not affect cylinder oils in the same way as is observed with lower pressures. It is comparatively an easy matter to filter feed water where there are lower steam pressures in use.

It is rather a difficult matter with the higher steam pressures, and in these modern days, of course, the higher pressures are the rule. Let me instance a case: On a certain stationary Corliss compound engine (using steam at about 120 pounds boiler pressure) they had trouble from the very beginning with the boilers, the tubes leaking badly, and it was found that the contractors in fitting up the engine had not provided a properly designed filter-box, or large enough for cleansing the feed water—that is to say, it was evident that there was not enough filtering material, and not as many chambers as there ought to be in a properly proportioned filter-box. An enlarged filter was substituted, but there was very little improvement with the boilers; the feed water as it came from the air pump seemed to be a mixture almost impossible to separate. There was a perfect emulsion of the water and oil. No matter how long it stood it would remain cloudy, with little or no grease rising to the surface. Finally, they tried a patent filter, having a chemical arrangement for coagulating the oil with alum; then they were able to cleanse the water perfectly from oil so that it was as clear as spring water, and the troubles immediately ceased with the boilers. I refer to this case, because it produced considerable discussion in the town where it occurred, and the first impulse was to condemn the use of surface condensers because of the injurious effect on the boilers. It would seem that the subject of cleansing feed water from oil and grease is one which has had very little consideration—at least in stationary engine practice. In marine engine practice we hear very little or nothing about trouble with oil in boilers—even with high pressures required for triple expansion engines. I am not a little surprised to find in stationary engine practice that there is so little known about it.

I was hoping that we might bring forth by this topical dis-

cussion the experiences of some of the members present. There has been a great deal written about the precipitation of lime and salts and other chemical impurities in feed water, but there has been very little said about grease in feed water as used in connection with surface condensers.

Mr. Daniel Ashworth.—This matter of bad water or water impregnated with very injurious properties with respect to the generation of steam has been a perplexing thing in our section of the country and in all tributaries entering into the Mississippi Valley. We have for a long time recognized this very simple fact, that the proper theory is to capture those impurities before they enter the boiler, and also that that can only be done by bringing it up to a temperature which is equivalent to the steam pressure in the boiler. In the Monongahela and the Youghiogheny and these various tributaries in our section of the country we have that trouble in an alarming degree, and hence to the people in the eastern section of the country and in the northwest it becomes a matter of surprise that we have a condition almost lapsing into a primitive barbarism as regards our steam boilers. I have no doubt that many excellent boilers well adapted to perform their functions in every other respect have been entirely ignored and condemned through a lack of facility for meeting this important requirement. This matter was presented to me, and after a thorough review at the Beaver Falls plant we introduced a system of purifiers of a cylindrical form having a series of shelves, and the circulation is conducted through these. I think it is known as the Hoppes system. While it is still but a brief period since its introduction it has worked with thorough satisfaction. I have also witnessed the operation of these purifiers where artesian wells have been sunk in our own city, in which the lime properties were such that the boilers had to be frequently taken out, and every plan was introduced which would obviate this without the outside influence of purifiers. In a period of eight days there would be a formation upon these shelves amounting to two and a half and three inches in thickness, and it was impossible to carry on their business successfully in the storage works without the application of this purifier system. That is the best system I have known of and it is one which is being closely watched. It is enabling us to use the modern boiler where heretofore we have simply stepped from the primitive cylinder boiler to the simple two-flue system

which we are endeavoring to get rid of, and we are in hopes that this purifier system will lead us out of this Slough of Despond.

Mr. John T. Hawkins.—I would like to ask Mr. Ashworth if he is familiar with the exact character of the impurities of the water in that section. Has any investigation been made to determine just what those impurities are—chemical, vegetable, or a combination?

Mr. Ashworth.—They are a combination. We have an analysis of all of them.

Mr. O. C. Woolson.—Mr. Ashworth said that there was no way of removing the solids or impurities except it be brought to the temperature carried in the boilers. Would that be absolutely necessary, or have any experiments been made to prove that the water had to be brought to a temperature or pressure equal to that carried by the boilers?

Mr. Ashworth.—It has been tried, and it was found absolutely necessary to bring it to the temperature of the boilers to get a thorough precipitation of those matters.

Mr. F'k Meriam Wheeler.—In the French "Belleville system" of boilers that is the great point that they make—precipitation at the highest temperatures. Belleville's purifying arrangement is really a part of the boiler, and the higher the temperature the more active the precipitation.

Mr. Wm. Kent.—A system was introduced some years ago, I think by Mr. Blessing of Albany, which consisted in taking water out of the boiler and pumping it through a filter to remove the scale, which was separated in the boiler, and then pumping it back again.

In regard to another point made by Mr. Wheeler respecting the emulsion of the oil in water, I would suggest that in very bad cases it might be well to apply the centrifugal system used for extracting cream from milk, although this would probably be a rather expensive method.

Mr. H. B. Roelker.—I would like to ask the gentleman if any other system is used but heating the feed water up to the temperature of the boiler.

Mr. Ashworth.—Nothing whatever, sir.

Mr. Roelker.—We were troubled a great deal with a deposit of Croton water scale in our boilers, and we finally thought we would try this system as it was brought before us by a well-known engineer. We took a cylinder 30 inches in diameter the chimney have so great a draught that they carry the gases into the chimney at too high temperature, and those farthest away have so low a draught that they do not work up to their rated capacity. The objections to one chimney to each boiler are chiefly æsthetical. I think that otherwise a chimney to each boiler is generally the best, although by careful designing of the horizontal flues and the dampers, and proper regulation of the latter, the evils of the single chimney may be avoided.

Mr. W. B. Le Van.—I would state that it does not make any difference where you place your boilers if the connecting flues to the boilers are properly proportioned and built. In one case where a well-designed horizontal connecting flue was used the boiler furthest off from the chimney had the best draught; in fact, the draught, according to the manner the flue is arranged, can be made about equal for all the boilers. If the connecting flue to the chimney is made of the proper capacity with double walls, and the inlets from each boiler to the flue are properly arranged, there will be no difficulty about each one having the correct amount of draught.

The manufacturing requirements of modern times necessitate the building of high chimneys.

All chimneys should be detached structures. The gravity of a well-built round exterior chimney will withstand a force of *fifty* (50) *pounds* on the square foot. Tall chimneys should diminish in thickness and in weight of material, from bottom to top, the top in all cases being as light as is compatible with security and endurance. A tall narrow stack is better than a low wide one.

The tops of chimneys should be concave toward the orifice, whereby the tendency will be rather to promote than to check the draught. If the top be straight across, a strong wind will act as a damper and tend to produce downward eddies, especially in very large chimneys.

The simplest and cheapest chimney is a brick one, the outside shaft built plumb—that is to say, without batter. It makes a simple structure, and as the amount of batter generally used does not add any to the stability, there is a great gain in time. The inner shaft is also carried up parallel with the outside shaft, with an air space between them. Grouting should never be used in chimneys or boiler setting. Grouting, being fluid mortar, becomes porous as the water evaporates, and possesses little adhesive power. Each course of bricks should be laid of the width or thickness of the wall and all joints well dashed with mortar and pointed before commencing the next course. The walls should be bonded every fourth course. In a sanitary point of view a high chimney is desirable. The city of Carlisle, England, has its sewers ventilated by about thirty chimneys belonging to the different factories. One of these chimneys is 300 feet high, having a velocity through it of fifty miles per hour, the pressure of air at the base being equal to a column of water 1.3 inches in height. The ventilation caused by this chimney was perceptible for a radius of 1,200 feet—equal to an area of over 103 acres.

Mr. O. C. Woolson.—My experience is, that it does not make much difference with the draught whether the boilers have a long or short flue provided the flue is properly and very carefully designed, and it is pretty difficult to design properly a flue of that kind. There are so many conditions, and yet it can be done. I for one should object seriously to setting boilers according to the sketch of Fig. 54, for some practical reasons; one objection is, that a man has got to travel too far to keep track of his gauges, his water, and so forth. Concerning the designing of flues and dampers I want to say that I came across some electric light boilers a few years ago which were not working uniformly or satisfactorily, due to a common error in swinging the butterfly dampers. There were four boilers in the plant, two each side of the brick stack, each pair connected with an ordinary hood or uptake, with a butterfly damper placed in the throat directly over the front end of boiler. The fireman remarked to me that he had trouble with the draught of the two left-hand boilers, and it puzzled him, because the boiler and settings were precisely the same for the four. I noticed that the dampers all swung in the same direction for the boilers at the left as well as for the two right-hand boilers, and this was obviously wrong, for it compelled the gases in the two left-hand boilers to travel a circuitous route to get to the stack. I told the fireman he would doubtless find all his difficulty in the dampers on the left, and he went to work and changed them to swing opposite to the right-hand battery, and has since had no further trouble.

I beg to add to my remarks at the meeting that at the Clark Thread Mills, East Newark, N. J., will be found a good example of boilers which have a better draught nearest the stack or furthest from the stack when set side by side and discharging into a single flue. At those mills there are four long batteries of ten to twelve boilers each in different buildings. Each battery has its own long independent flue (underground), and I should say the furthest boiler from the stack in each of these batteries was about 180 feet and the nearest boiler about 60 feet, and yet I have been told that there has been scarcely any difference detected in the draught of the several boilers, and such difference as was noticeable was that the furthest boilers in each of the batteries exhibited the best draught.

I can say this only, from my own observation, that with their 335 foot stack the draught at the fire doors is tremendous.

Mr. Charles E. Emery.—It seems to me that the office of an engineer is to settle questions like this as they occur. In the distribution of water in a city it is desirable that the pressure be substantially the same in different districts; a similar problem arises in electrical distribution in multiple arc, and the same question in a much more simple form must be considered in arranging to collect in a common stack the gases from boilers situated at different distances therefrom. It belittles the engineer to say that it is necessary to have a separate stack for each boiler, when it involves merely a question of engineering skill to arrange flues of substantially the same resistance from the boilers to the stack. The problem is a much more simple one when all the boilers are arranged as near the stack as possible. The arrangement suggested by Mr. Porter is a desirable one when the unit is a group of boilers instead of a single boiler. For single boilers the arrangement is entirely unnecessary, as there is no difficulty in arranging a flue so that the draught for each of a number of boilers connected therewith will be substantially the same. The same principle will, however, secure satisfactory results when some of the boilers are at a considerable distance from the stack. It simply requires that the farther boilers have ample flue area not interfered with by conflicting currents from the other boilers, and in some cases the flues of the nearer boilers must be reduced in area so as to give full draught to the farther ones. The latter is, of course, not desirable if it can be avoided, so that it is better to make the general design so that but few boilers are connected to the same flue. It was frequently quite common on board ship to connect all the boilers

	6×9.	7×10.	8×10	8×12.	9×12.	10×12.	10×15.	10×20.	11×15.	12×16.	12×18.	12×24.	14×16.	14×18.	14×30.	16×18.	16×24.	18×24.	20×24.	20×30.
Weight of small pulley for det. engine	303	342	342	..	650	..	..	..	..	..	....	....	1270	..	..	..	....	....	....	....
Weight of large pulley for det. engine	93	120	125	..	218	..	..	..	..	..	...	....	526	....	..	..	....	....	....	....
Weight of small pulley for sta. engine	...	...	...	..	...	..	..	..	..	..	....	....	....	....	..	..	....	....	...	....
Weight of large pulley for sta. engine	...	...	...	..	...	..	..	..	..	..	1720	2500	1600	2500	..	..	3400	3500	3500	5200

Mr. Frank H. Ball.—I would like to ask the members of the society if any of them ever tried using keys of the form illustrated in the sketch, Fig. 53. I have tried it with very good success.

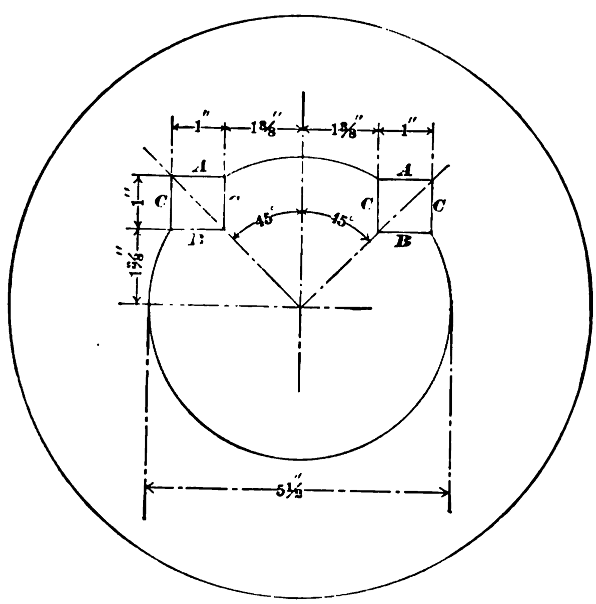

FIG. 53.

Mr. Geo. M. Bond.—I might say that there is a special system of small keys adopted at our works in which the depth consideration, so far as the shaft is concerned, is overcome by dropping a circular milling cut into the stem or shaft, and a key cut from a bar of semi-circular section fitted in tightly. In this way the side strain is resisted by a greater depth of contact in the shaft, while a considerably narrower width of key is safely used. These keys, by giving them a slight inclination longitudinally,

about that statement, and told me that it was a pity the pitch could not be changed. The way in which Mr. Whitworth fell into that error, which is the only error in the pitch of the threads in the Whitworth system, is obvious. The English do not use the $\frac{9}{16}$-inch bolt. The reduction in size is by eighths of an inch until you get down to half an inch, so that, the number of threads increasing one thread for each smaller bolt, they have thus: For the 1-inch bolt, 8 threads; for the $\frac{7}{8}$-inch bolt, 9 threads; for the $\frac{3}{4}$-inch bolt, 10 threads; for the $\frac{5}{8}$-inch bolt, 11 threads; for the $\frac{1}{2}$-inch bolt, 12 threads; for the $\frac{7}{16}$-inch bolt, 14 threads.

That seems to be a natural arrangement, although the last step is singularly abrupt, the diameter of the bolt being reduced only $\frac{1}{16}$ inch, and the number of threads increasing by two, 14 threads being given to the $\frac{7}{16}$-inch bolt. But in this country we use the $\frac{9}{16}$-inch bolt, and that cures the difficulty at once. We have for the 1-inch bolt, 8 threads; for the $\frac{7}{8}$-inch bolt, 9 threads; for the $\frac{3}{4}$-inch bolt, 10 threads; for the $\frac{5}{8}$-inch bolt, 11 threads; for the $\frac{9}{16}$-inch bolt, 12 threads; for the $\frac{1}{2}$-inch bolt, 13 threads; for the $\frac{7}{16}$-inch bolt, 14 threads.

There is no reason why this blunder of the English system should exist in this country, where we conform with what the designers of the English system wish they had done, but find too late they cannot do.

While we criticise that one defect in the Whitworth system, we should not overlook the fact that we are indebted to Mr. Whitworth for the uniform system. Mr. Whitworth found the system of screw threads in use in England in a state of chaos, every large engineering establishment having a thread peculiar to itself, and these threads differing widely, and each one maintaining its thread strenuously, so that nobody else could repair their machinery; and the problem with Mr. Whitworth was to introduce a uniform thread which would be adopted, and he told me himself that he proceeded in this way, which of course will strike every one as the common-sense method: He obtained from every leading manufacturer of machinery his system of threads. Then he averaged these, and adopted a system which was the mean of the existing practice, and so was presumably correct, and that recommended itself sufficiently to be generally adopted. But there is another thing about the English thread. How did it get such an impracticable and amazing feat-

inches to $1\frac{1}{2}$ inches; $\frac{3}{8}$ inch, $1\frac{1}{2}$ to $1\frac{3}{4}$ inches, and so on until two keys become necessary. Taper $\frac{1}{8}$ inch to foot. The total thickness at large end of spline, $\frac{4}{5}$ of the width of key.

Mr. Geo. L. Fowler.—Our practice in the West in regard to this matter was to have the key-way in the hub of the pulley one-half the edge of the key measured at the edge, so that the shaft was cut out a trifle more on the centre line than it would be in the pulley, making it easier to the workman to measure.

The taper of the key is between the sides marked "A" and "B," with the sides "C" made parallel, and no attempt to get a bearing on these sides "C."

The loosening of a wheel on a shaft is generally the result of a slight turning on the shaft an innumerable number of times, each time of greater amplitude, due to the wearing away of the surfaces in contact.

With the arrangement shown, the likelihood of this first small beginning of motion is reduced, because it cannot occur without crushing the metal between "A" and "B," and the angle of resistance of these surfaces is much better than with a single key of ordinary width.

Mr. Roelker.—I used to make a great many things like that for heavy marine work, but there was a central key besides.

No. 477-90.

"Have you had any experience in out-door work at night in wind and storm, with portable apparatus for light in large quantities, other than electric light?"

Mr. W. W. Dingee.—The steamboats plying the Mississippi River and tributaries have always used at their landings a broom-shaped torch made of hemp and dipped in tar, and when set aflame it gave a great light and made a prodigious smoke. This invention was never patented nor is it known to whom the honor belongs. No one has been able in the past half-century to either improve or displace it. The songs of the negro boat hands, with many other things peculiar to the navigation of western rivers, are things of the past, but as a means of converting a limited area of darkness into day the flambeau stands without a rival.

Mr. Wm. Kent.—In the work of putting down the cable railway in Broadway, New York, the contractors are using the Wells light, which gives a great brilliance. The light is generated from

vent other manufacturers from supplying bolts and nuts for his machines, and he adopted a very curious system, which has, unfortunately, been allowed to exist until to-day. He thought he would adopt certain sizes of bolts which would be available to turn from the commercial sizes of iron, so that he adopted .28-inch for a bolt to be turned from $\frac{5}{16}$-inch iron, and .6-inch bolts from $\frac{5}{8}$-inch iron, .48-inch from $\frac{1}{2}$-inch iron, .72-inch from $\frac{3}{4}$-inch iron, and so on to .96-inch from 1-inch, and his machines are built to-day with all those sizes of bolts, with the number of threads in each case very near the U. S. standard.

Mr. W. O. Webber.—I do not know of any objection to cutting a $\frac{1}{2}''$ bolt with 13 threads to the inch, excepting that very few of the old engine lathes were provided with change gears which readily make 13 threads; and think this is probably the cause of adhering to the old 12-thread standard.

Mr. Oberlin Smith.—I think it was that great political engineer, the late Horace Greeley, who said that the best way to resume specie payments was "to resume;" and it seems to me that in a matter of this kind, when we see it is better to change from a bad thing to a good thing, the best way is to *do* it. I thought some years ago, when I was using 12 threads on $\frac{1}{2}''$ bolts, that it would be a very difficult thing to change, and would give a great deal of trouble. But I made up my mind that the longer I lived the worse it would grow, and the only thing to do was to change then—which I did. I believe it was the best policy, and that any one who does make such a change in his shops will certainly not regret it after a few months. After a year or so he will never think of it again. There is one thing worse than "getting the mean of the thread," and that is getting a mean thread, which a 12 per inch certainly is, on such small bolts.

No. 477-94.

Have you had any experience with systems for purification of bad feed water before it gets into a steam boiler, either by precipitation or otherwise?

Mr. F'k Merriam Wheeler.—I regret that the gentleman who suggested this topic is not here to bring out the points he desires. He speaks of "bad feed water." Now, these words can be construed mildly or otherwise. I would like to know what kind of "badness" he desires us to give consideration, and whether he has in view animal, mineral, or vegetable matter in

the strain of belt and the pressure under which the loose pulley runs. The difference in diameter can be provided for by a gradual incline up which the belt can easily be shipped.

Friction pulleys when they can be large enough for the work required furnish a very satisfactory means of stopping and starting shafting. When at work the shafting is in line, and but little movement is required to throw them out of contact.

When shafting is driven on upper floors from below, a properly arranged tightener can be used for stopping and starting. The objection to this method is that it is running an idler all the time for the sake of stopping when occasion requires.

Mr. Wm. Kent—Those who are interested in this question can see an example of it in the Delavergne Refrigerating Works when they visit them to-morrow.

Mr. Geo. M. Bond.—I think a well-designed friction clutch would be as efficient as anything. In the building for the small-tool department lately added to our shop we have about seventy-five horse power transmitted through belts to the upper floors and arranged so as to be disengaged by the simple operation of a lever which can be detached and set up in the corner so as to be out of the way. Whenever the power is to be disconnected the friction is thrown out and there is no disturbance of any kind—no getting out of line and no undue friction while the belts are idle.

Mr. Jas. McBride.—I have a number of large machines of fifty horse power and upwards with which I use tighteners exclusively for stopping and starting. I think, of all the mechanical abominations of which I know, a loose pulley on a shaft with shifting belt is the worst. I use those belts eighteen or twenty inches wide. The distance between the centres of the shafts is about eighteen feet, one very nearly over the other.

A Member.—One trouble with friction clutches in connection with such matters oftentimes is the difficulty of getting the lubricant in where it ought to be. If the clutch is stopped much of the time, the portion of the pulley which revolves on the shaft is apt to be not lubricated. I found that that is very satisfactorily overcome in many cases in this way: Instead of using ordinary brass bushings use a graphite filled bushing. By that means it will work for a year or more practically without any oil. Otherwise I have had a great deal of trouble with their cutting. Where belts are wide it is of course almost im-

cussion the experiences of some of the members present. There has been a great deal written about the precipitation of lime and salts and other chemical impurities in feed water, but there has been very little said about grease in feed water as used in connection with surface condensers.

Mr. Daniel Ashworth.—This matter of bad water or water impregnated with very injurious properties with respect to the generation of steam has been a perplexing thing in our section of the country and in all tributaries entering into the Mississippi Valley. We have for a long time recognized this very simple fact, that the proper theory is to capture those impurities before they enter the boiler, and also that that can only be done by bringing it up to a temperature which is equivalent to the steam pressure in the boiler. In the Monongahela and the Youghiogheny and these various tributaries in our section of the country we have that trouble in an alarming degree, and hence to the people in the eastern section of the country and in the northwest it becomes a matter of surprise that we have a condition almost lapsing into a primitive barbarism as regards our steam boilers. I have no doubt that many excellent boilers well adapted to perform their functions in every other respect have been entirely ignored and condemned through a lack of facility for meeting this important requirement. This matter was presented to me, and after a thorough review at the Beaver Falls plant we introduced a system of purifiers of a cylindrical form having a series of shelves, and the circulation is conducted through these. I think it is known as the Hoppes system. While it is still but a brief period since its introduction it has worked with thorough satisfaction. I have also witnessed the operation of these purifiers where artesian wells have been sunk in our own city, in which the lime properties were such that the boilers had to be frequently taken out, and every plan was introduced which would obviate this without the outside influence of purifiers. In a period of eight days there would be a formation upon these shelves amounting to two and a half and three inches in thickness, and it was impossible to carry on their business successfully in the storage works without the application of this purifier system. That is the best system I have known of and it is one which is being closely watched. It is enabling us to use the modern boiler where heretofore we have simply stepped from the primitive cylinder boiler to the simple two-flue system

which was very nice to throw in, but you couldn't throw it out when it got in there. Now if anybody has a friction clutch which will answer all purposes and run in any place they have got a good thing, and I would like to get hold of it. The Weston clutch is a good clutch. We know what it is capable of, but after it has been in operation a while you cannot throw it out. Now we have got to apply our knowledge to the circumstances in different places. What is good practice to-day is poor to-morrow, and *vice versa,* and no one rule applies to all of these instances. I have been forty years in the harness and have seen a good many things tried. But as to the idea of making a law with these questions to apply to everything—you cannot do it.

The President.—That is quite true. But do you not think that as the members give their experience under certain conditions they give us valuable information which may keep you and me from going through their disastrous experience? As I take it, that is the idea and the purport of these topical questions. What we want is to avoid those dangers and get our diplomas from the school of success. With your permission, I think that is the idea of these questions. You tell us of your clutch which will go in and not go out. Now we will not put that in, you know. You have given us a contribution.

Mr. Cartwright.—It would be the same with any kind of clutch. There is no clutch which will cover the ground in all cases. It asks here what is the best design for line shafting transmitting over fifty horse-power, permitting them to be stopped or started on any floor without interfering with the motor or other shafting. You might go and look at it and say one thing. I might go and look at it and say another. There is no one law which will apply to anything of that kind. Then, in transmitting over fifty horse-power, are we to have one for fifty and one for sixty, and so on?

Mr. McBride.—As Mr. Cartwright is not in love with the clutch, I would suggest to him to try the belt and tightening pulleys.

Mr. Cartwright.—Probably some have had experience with the belts and tighteners; I have had. Now for a power hammer with an intermittent motion, there is nothing which equals the idler dropping on to the belt, in my estimation, because you can put your hammer in motion slowly and develop your power as you need it. But with an electric light plant there are a great and 10 or 12 feet high, and placed it so that from shortly above the bottom of it the water would run by natural gravity into the boiler. We carried about 80 pounds steam pressure. At the top of this cylinder we had a sprinkler arrangement perforated with very small holes, and we filled this cylinder pretty well up with coke and connected the boiler—the steam part as well as the water part—with it. Probably the water level was about 6 or 8 feet below the top of this upright cylinder. We ran our boilers in that way, feeding into the top of this cylinder, sprinkling into the steam space and making the water trickle over the coke, and it had absolutely no effect whatever, and there was not a particle of difference in the quantity of lime which we got out of the boiler nor in the trouble which we had from our boilers. Of course it is not a very large quantity of lime which is in Croton water, but we got about a pailful a week, and we had to abandon the system as absolutely worthless. After using it several months we could find no trace of lime on the coke or anywhere excepting in the boiler itself.

Mr. W. H. Odell.—I have had some experience with a device such as Mr. Kent asks about. Several years since I was engaged to superintend the erection and make test of an apparatus in which a branch from the blow-off was connected direct to a "Hyatt" filter and thence returned to the boiler. Circulation was induced with a "Blessing" trap. The very impure feed water was pumped directly into the boiler. Analysis of the feed water before entering the boiler showed the quantity used each day carried into the boiler about four pounds of impurities. These impurities were washed from the filter each twenty-four hours, and after thorough drying were weighed, and each day's collection weighed about four pounds. I cannot remember the exact weight of impurities or the analysis of the feed water, but I called at this place some four years later and was informed that the device had given most excellent satisfaction up to the time the buildings were destroyed by fire.

In the Sweet's rolling mill, at Syracuse, N. Y. (the same city in which the test I have just mentioned was made), they have a somewhat similar device, and Mr. William A. Sweet, proprietor, tells me it is giving most excellent results. In Mr. Sweet's mill I think the feed water is treated and filtered, in part, before entering the boilers, and if I remember rightly the small boiler which heats the feed water before filtering carries a higher steam pressure than the main boilers. Here are two cases from experience, and I might cite others.

Mr. Fred. Meriam Wheeler.—I would like to add one word about the water supply of Syracuse. As Mr. Odell says, they are very much troubled there—the people as well as the boilers —with the city water, and I am glad to learn that they are about securing a new and better water supply. I also know that Mr. Cogswell, of the Solvay Process Company, has spent many dollars in contending with bad water at his works. Mr. Kent's suggestion of getting rid of greasy matter from water by centrifugal action I am afraid would be rather ineffectual. I think this subject of purifying feed water for boilers really a very important one. While the filter which I spoke of (using the coagulating principle) is a very good one it might be improved upon. I am not connected with the manufacture of filters, but I am very much interested in securing a simple and perfect system for cleansing the grease from the feed water as used in connection with surface condensers, more particularly with the high steam pressures.

The Devoe Manufacturing Company have had a surface condenser in use eight or ten years and have had no trouble with their boilers. The engineer every day gives his boilers a dose of lime and soda; but then they use steam only at 60 lbs., which is an entirely different problem from using it at 100 or 120 lbs. pressure. In marine practice they have recourse to salt water very frequently, also the use of soda; but then their boilers, compared with the average stationary boilers, get very much more attention. As soon as a steamer comes into port the boilers get prompt and thorough care the same as the engines. You cannot say that for average stationary engine practice.

Mr. Arthur Falkenau.—I know of an experience in the Edison station in Philadelphia. In that case Professor Marks employed several different filters unsuccessfully. He at last put in some of the Hoppes filters that were just spoken of, but finally resorted to using cutch, a dye stuff, in the well. They have a well under the building. Cutch was put in the well and they used an injection of caustic soda in the heaters every day. That seemed to be amply successful for the purpose.

Mr. John J. Hoppes.—I do not like to blow my own horn, but the gentleman last on the floor made a statement which I think, when he investigates, he will agree with me was incorrect. Mr. Marks did put some purifiers in the building, but they were never connected up and never operated.

I would like, while I am on the floor, to show the principle of our purifier. They speak of it as a matter of precipitation. Our purifier, while it operates partially on the plan of precipitation, is not entirely based on that plan. That is, we do not depend on the plan of precipitation. All other purifiers which have been used depend upon the plan of precipitation or filtration, and some arrangements used in boilers, called skimmers, depend on skimming the top off the water—that is, those precipitates which float on top of the water, usually carbonate of magnesia. Generally the pans of our purifiers are made in a tray form. We have six pans in each tier in a purifier. These pans have extensions resting on ways. The ends of the pans are higher than the sides. The water flows over evenly on both sides of the pan the whole length of the purifier. It flows along the under side of the pan in a thin film, and nine-tenths of the solids caught in the purifier are caught on the under side of the pan, the same as a stalactite is caught on the roof of a cave. We use no filter whatever. We have formations showing as much as three inches taken from the under side of the second pan. They build up on the edge of the pan and run gradually to a thinner formation near the centre. The second pan has the heaviest formation, usually, when the purifier is ready to be cleaned, about an inch and one-half thick. The sixth or bottom pan will have a thirty-second to a sixteenth of an inch formation on the under side. Now waters which have different chemicals in them of course form different scales. The water which is the worst to purify shows a soft formation on the under side of the pans; that shows then that it is composed of sulphates. When soft and white we know then that the solids in the water are sulphates. Sulphates form a hard scale in boilers. Carbonates form a harder scale in the purifier, and we like to handle water which contains carbonates, because in operating purifiers where there are sulphates in large quantity, the least irregularity is likely to wash the sulphates off Where there are carbonates in the water, and also sulphates, you will find carbonates and sulphates mixed in the formation from the second pans down to the last. If silicates are present in the water the last pan will little adhesive power. Each course of bricks should be laid of the width or thickness of the wall and all joints well dashed with mortar and pointed before commencing the next course. The walls should be bonded every fourth course. In a sanitary point of view a high chimney is desirable. The city of Carlisle, England, has its sewers ventilated by about thirty chimneys belonging to the different factories. One of these chimneys is 300 feet high, having a velocity through it of fifty miles per hour, the pressure of air at the base being equal to a column of water 1.3 inches in height. The ventilation caused by this chimney was perceptible for a radius of 1,200 feet—equal to an area of over 103 acres.

Mr. O. C. Woolson.—My experience is, that it does not make much difference with the draught whether the boilers have a long or short flue provided the flue is properly and very carefully designed, and it is pretty difficult to design properly a flue of that kind. There are so many conditions, and yet it can be done. I for one should object seriously to setting boilers according to the sketch of Fig. 54, for some practical reasons; one objection is, that a man has got to travel too far to keep track of his gauges, his water, and so forth. Concerning the designing of flues and dampers I want to say that I came across some electric light boilers a few years ago which were not working uniformly or satisfactorily, due to a common error in swinging the butterfly dampers. There were four boilers in the plant, two each side of the brick stack, each pair connected with an ordinary hood or uptake, with a butterfly damper placed in the throat directly over the front end of boiler. The fireman remarked to me that he had trouble with the draught of the two left-hand boilers, and it puzzled him, because the boiler and settings were precisely the same for the four. I noticed that the dampers all swung in the same direction for the boilers at the left as well as for the two right-hand boilers, and this was obviously wrong, for it compelled the gases in the two left-hand boilers to travel a circuitous route to get to the stack. I told the fireman he would doubtless find all his difficulty in the dampers on the left, and he went to work and changed them to swing opposite to the right-hand battery, and has since had no further trouble.

I beg to add to my remarks at the meeting that at the Clark Thread Mills, East Newark, N. J., will be found a good example of boilers which have a better draught nearest the stack or furthest from the stack when set side by side and discharging into a single flue. At those mills there are four long batteries of ten to twelve boilers each in different buildings. Each battery has its own long independent flue (underground), and I should say the furthest boiler from the stack in each of these batteries was about 180 feet and the nearest boiler about 60 feet, and yet I have been told that there has been scarcely any difference detected in the draught of the several boilers, and such difference as was noticeable was that the furthest boilers in each of the batteries exhibited the best draught.

I can say this only, from my own observation, that with their 335 foot stack the draught at the fire doors is tremendous.

Mr. Charles E. Emery.—It seems to me that the office of an engineer is to settle questions like this as they occur. In the distribution of water in a city it is desirable that the pressure be substantially the same in different districts; a similar problem arises in electrical distribution in multiple arc, and the same question in a much more simple form must be considered in arranging to collect in a common stack the gases from boilers situated at different distances therefrom. It belittles the engineer to say that it is necessary to have a separate stack for each boiler, when it involves merely a question of engineering skill to arrange flues of substantially the same resistance from the boilers to the stack. The problem is a much more simple one when all the boilers are arranged as near the stack as possible. The arrangement suggested by Mr. Porter is a desirable one when the unit is a group of boilers instead of a single boiler. For single boilers the arrangement is entirely unnecessary, as there is no difficulty in arranging a flue so that the draught for each of a number of boilers connected therewith will be substantially the same. The same principle will, however, secure satisfactory results when some of the boilers are at a considerable distance from the stack. It simply requires that the farther boilers have ample flue area not interfered with by conflicting currents from the other boilers, and in some cases the flues of the nearer boilers must be reduced in area so as to give full draught to the farther ones. The latter is, of course, not desirable if it can be avoided, so that it is better to make the general design so that but few boilers are connected to the same flue. It was frequently quite common on board ship to connect all the boilers to one smoke-pipe, and as space on shipboard is valuable the farthest boilers had evidently the least draught. Now it is more common to put in more smoke-pipes, and connect each with a group of boilers at a reasonable distance therefrom. In the large steam station of the New York Steam Company, at Greenwich and Cortlandt streets, in this city, designed by the speaker, there was no difficulty of this kind, as the boilers are arranged in tiers, with only four on each side connected to one chimney. The flues from each group of four boilers enter on opposite sides of the chimney; but the latter is so large that no interference is detected, and it was not found necessary to put in deflectors. There is, of course, a difference in draught, due to the difference in elevation on the different floors.

Mr. Wm. O. Webber.—I think it is best to use one large chimney for any battery of boilers, for several reasons; among them being, first, the cheapness in cost of construction; the efficiency of the chimney, *i. e.*, less frictional resistance to the draught and consequently the better draught obtained in proportion to the cost; also the space occupied.

Mr. Daniel Ashworth.—In our system of steam plants at Beaver Falls, where we have twenty-eight boilers, we had adopted the system of a central stack located midway, being flanked on either side by the fourteen boilers. We do not find both at the Edgar Thomson and at this plant that the draught is decreased for the boiler farthest from the central point. We are drifting rapidly—in fact, it is becoming almost standard with us—to put up a large central stack and have the flue system leading to that. That is our intended practice.

No. 477-93.

"Why should any one cut a half-inch bolt with twelve threads to the inch? Is there any objection to the U. S. standard of thirteen threads?"

Mr. Charles T. Porter.—Some twenty-five years ago, at the Whitworth works in Manchester, I was told by Mr. Widdowson, who then was, and for many years had been, the superintendent of the tool department of those works—the tool department in contradistinction from the ordnance department—that giving to the ½-inch bolt only 12 threads was a mistake, and one which they had always regretted; that it cut away the bolt unnecessarily, and the bolt should have had 13 threads. He was frank

about that statement, and told me that it was a pity the pitch could not be changed. The way in which Mr. Whitworth fell into that error, which is the only error in the pitch of the threads in the Whitworth system, is obvious. The English do not use the $\frac{9}{16}$-inch bolt. The reduction in size is by eighths of an inch until you get down to half an inch, so that, the number of threads increasing one thread for each smaller bolt, they have thus: For the 1-inch bolt, 8 threads; for the $\frac{7}{8}$-inch bolt, 9 threads; for the $\frac{3}{4}$-inch bolt, 10 threads; for the $\frac{5}{8}$-inch bolt, 11 threads; for the $\frac{1}{2}$-inch bolt, 12 threads; for the $\frac{7}{16}$-inch bolt, 14 threads.

That seems to be a natural arrangement, although the last step is singularly abrupt, the diameter of the bolt being reduced only $\frac{1}{16}$ inch, and the number of threads increasing by two, 14 threads being given to the $\frac{7}{16}$-inch bolt. But in this country we use the $\frac{9}{16}$-inch bolt, and that cures the difficulty at once. We have for the 1-inch bolt, 8 threads; for the $\frac{7}{8}$-inch bolt, 9 threads; for the $\frac{3}{4}$-inch bolt, 10 threads; for the $\frac{5}{8}$-inch bolt, 11 threads; for the $\frac{9}{16}$-inch bolt, 12 threads; for the $\frac{1}{2}$-inch bolt, 13 threads; for the $\frac{7}{16}$-inch bolt, 14 threads.

There is no reason why this blunder of the English system should exist in this country, where we conform with what the designers of the English system wish they had done, but find too late they cannot do.

While we criticise that one defect in the Whitworth system, we should not overlook the fact that we are indebted to Mr. Whitworth for the uniform system. Mr. Whitworth found the system of screw threads in use in England in a state of chaos, every large engineering establishment having a thread peculiar to itself, and these threads differing widely, and each one maintaining its thread strenuously, so that nobody else could repair their machinery; and the problem with Mr. Whitworth was to introduce a uniform thread which would be adopted, and he told me himself that he proceeded in this way, which of course will strike every one as the common-sense method: He obtained from every leading manufacturer of machinery his system of threads. Then he averaged these, and adopted a system which was the mean of the existing practice, and so was presumably correct, and that recommended itself sufficiently to be generally adopted. But there is another thing about the English thread. How did it get such an impracticable and amazing feat-

re as the angle of 55°? That is something which can be readily explained. Mr. Whitworth had no particular purpose of benefiting the world, but he had a particular purpose of benefiting himself. He wanted to introduce a system of threads which would be generally adopted, but which nobody else could make, and he was shrewd enough to solve that rather complicated problem quite satisfactorily for himself by adopting an angle for the thread which, he stated to the world, was also the mean of all the angles in use. But it was an angle which it was most difficult to originate, or to verify or to restore if it was lost, or to bisect with correctness so that the opposite faces of the threads would form the same angle with the normal, and so, by means of this angle of 55°, Mr. Whitworth was enabled for a long time to monopolize the manufacture of taps and dies in England. Cutting threads in lathes was little known at that time, stocks and dies were commonly used for cutting threads on bolts by hand, and the manufacture of taps and stocks and dies formed an important branch of the business of the Whitworth Company when I knew it. But we are fortunately delivered from that feature, also.

Mr. Geo. E. Whitehead.—It has always been a source of annoyance to me to have the 12 and 13 thread bolts manufactured at the same time. We have a great many customers who order ½-inch bolts, and we hardly know whether to send them 12 or 13 thread. Also, in tapping the nuts, the 12 thread and 13 thread are very apt to get mixed. I should like to see the 12 thread abolished entirely. I have asked a number of persons why they continue to use the 12 thread, but fail to get any good cause for sticking to it. It certainly makes the bolt very weak.

Mr. W. W. Dingee.—Thirteen threads are probably better than twelve for ½-inch bolts, but many old establishments cling to the latter because they started that way and do not like to change. It might be added that if nuts and bolts had always fitted each other properly, there would have been little demand for lock-nuts and washers, of which there are now a great variety.

Mr. Jno. T. Hawkins.—I would like to give a little contribution to the history of the sort of thing which Mr. Porter described as the original incentive to Mr. Whitworth to establish a uniform system of threads in England. The late Andrew Campbell, who was the inventor of the several varieties of the Campbell printing press, conceived the same unfortunate idea: that he would pre-

vent other manufacturers from supplying bolts and nuts for his machines, and he adopted a very curious system, which has, unfortunately, been allowed to exist until to-day. He thought he would adopt certain sizes of bolts which would be available to turn from the commercial sizes of iron, so that he adopted .28-inch for a bolt to be turned from $\frac{5}{16}$-inch iron, and .6-inch bolts from $\frac{5}{8}$-inch iron, .48-inch from $\frac{1}{2}$-inch iron, .72-inch from $\frac{3}{4}$-inch iron, and so on to .96-inch from 1-inch, and his machines are built to-day with all those sizes of bolts, with the number of threads in each case very near the U. S. standard.

Mr. W. O. Webber.—I do not know of any objection to cutting a $\frac{1}{2}''$ bolt with 13 threads to the inch, excepting that very few of the old engine lathes were provided with change gears which readily make 13 threads; and think this is probably the cause of adhering to the old 12-thread standard.

Mr. Oberlin Smith.—I think it was that great political engineer, the late Horace Greeley, who said that the best way to resume specie payments was "to resume;" and it seems to me that in a matter of this kind, when we see it is better to change from a bad thing to a good thing, the best way is to *do* it. I thought some years ago, when I was using 12 threads on $\frac{1}{2}''$ bolts, that it would be a very difficult thing to change, and would give a great deal of trouble. But I made up my mind that the longer I lived the worse it would grow, and the only thing to do was to change then—which I did. I believe it was the best policy, and that any one who does make such a change in his shops will certainly not regret it after a few months. After a year or so he will never think of it again. There is one thing worse than "getting the mean of the thread," and that is getting a mean thread, which a 12 per inch certainly is, on such small bolts.

No. 477-94.

Have you had any experience with systems for purification of bad feed water before it gets into a steam boiler, either by precipitation or otherwise?

Mr. F'k Merriam Wheeler.—I regret that the gentleman who suggested this topic is not here to bring out the points he desires. He speaks of "bad feed water." Now, these words can be construed mildly or otherwise. I would like to know what kind of "badness" he desires us to give consideration, and whether he has in view animal, mineral, or vegetable matter in

the feed water. I would also like to inquire if any gentleman here has had any extended experience in the matter of cleansing feed water from grease and oil. With the surface condenser system this is a trouble with which we have to contend, and it is sometimes quite a serious matter. It has been found that there is different chemical action on cylinder lubricants as used in different types of engines—that is to say, higher steam pressures and temperatures do not affect cylinder oils in the same way as is observed with lower pressures. It is comparatively an easy matter to filter feed water where there are lower steam pressures in use. It is rather a difficult matter with the higher steam pressures, and in these modern days, of course, the higher pressures are the rule. Let me instance a case: On a certain stationary Corliss compound engine (using steam at about 120 pounds boiler pressure) they had trouble from the very beginning with the boilers, the tubes leaking badly, and it was found that the contractors in fitting up the engine had not provided a properly designed filter-box, or large enough for cleansing the feed water—that is to say, it was evident that there was not enough filtering material, and not as many chambers as there ought to be in a properly proportioned filter-box. An enlarged filter was substituted, but there was very little improvement with the boilers; the feed water as it came from the air pump seemed to be a mixture almost impossible to separate. There was a perfect emulsion of the water and oil. No matter how long it stood it would remain cloudy, with little or no grease rising to the surface. Finally, they tried a patent filter, having a chemical arrangement for coagulating the oil with alum; then they were able to cleanse the water perfectly from oil so that it was as clear as spring water, and the troubles immediately ceased with the boilers. I refer to this case, because it produced considerable discussion in the town where it occurred, and the first impulse was to condemn the use of surface condensers because of the injurious effect on the boilers. It would seem that the subject of cleansing feed water from oil and grease is one which has had very little consideration—at least in stationary engine practice. In marine engine practice we hear very little or nothing about trouble with oil in boilers—even with high pressures required for triple expansion engines. I am not a little surprised to find in stationary engine practice that there is so little known about it.

I was hoping that we might bring forth by this topical dis-

cussion the experiences of some of the members present. There has been a great deal written about the precipitation of lime and salts and other chemical impurities in feed water, but there has been very little said about grease in feed water as used in connection with surface condensers.

Mr. Daniel Ashworth.—This matter of bad water or water impregnated with very injurious properties with respect to the generation of steam has been a perplexing thing in our section of the country and in all tributaries entering into the Mississippi Valley. We have for a long time recognized this very simple fact, that the proper theory is to capture those impurities before they enter the boiler, and also that that can only be done by bringing it up to a temperature which is equivalent to the steam pressure in the boiler. In the Monongahela and the Youghiogheny and these various tributaries in our section of the country we have that trouble in an alarming degree, and hence to the people in the eastern section of the country and in the northwest it becomes a matter of surprise that we have a condition almost lapsing into a primitive barbarism as regards our steam boilers. I have no doubt that many excellent boilers well adapted to perform their functions in every other respect have been entirely ignored and condemned through a lack of facility for meeting this important requirement. This matter was presented to me, and after a thorough review at the Beaver Falls plant we introduced a system of purifiers of a cylindrical form having a series of shelves, and the circulation is conducted through these. I think it is known as the Hoppes system. While it is still but a brief period since its introduction it has worked with thorough satisfaction. I have also witnessed the operation of these purifiers where artesian wells have been sunk in our own city, in which the lime properties were such that the boilers had to be frequently taken out, and every plan was introduced which would obviate this without the outside influence of purifiers. In a period of eight days there would be a formation upon these shelves amounting to two and a half and three inches in thickness, and it was impossible to carry on their business successfully in the storage works without the application of this purifier system. That is the best system I have known of and it is one which is being closely watched. It is enabling us to use the modern boiler where heretofore we have simply stepped from the primitive cylinder boiler to the simple two-flue system

which we are endeavoring to get rid of, and we are in hopes that this purifier system will lead us out of this Slough of Despond.

Mr. John T. Hawkins.—I would like to ask Mr. Ashworth if he is familiar with the exact character of the impurities of the water in that section. Has any investigation been made to determine just what those impurities are—chemical, vegetable, or a combination?

Mr. Ashworth.—They are a combination. We have an analysis of all of them.

Mr. O. C. Woolson.—Mr. Ashworth said that there was no way of removing the solids or impurities except it be brought to the temperature carried in the boilers. Would that be absolutely necessary, or have any experiments been made to prove that the water had to be brought to a temperature or pressure equal to that carried by the boilers?

Mr. Ashworth.—It has been tried, and it was found absolutely necessary to bring it to the temperature of the boilers to get a thorough precipitation of those matters.

Mr. F'k Meriam Wheeler.—In the French "Belleville system" of boilers that is the great point that they make—precipitation at the highest temperatures. Belleville's purifying arrangement is really a part of the boiler, and the higher the temperature the more active the precipitation.

Mr. Wm. Kent.—A system was introduced some years ago, I think by Mr. Blessing of Albany, which consisted in taking water out of the boiler and pumping it through a filter to remove the scale, which was separated in the boiler, and then pumping it back again.

In regard to another point made by Mr. Wheeler respecting the emulsion of the oil in water, I would suggest that in very bad cases it might be well to apply the centrifugal system used for extracting cream from milk, although this would probably be a rather expensive method.

Mr. H. B. Roelker.—I would like to ask the gentleman if any other system is used but heating the feed water up to the temperature of the boiler.

Mr. Ashworth.—Nothing whatever, sir.

Mr. Roelker.—We were troubled a great deal with a deposit of Croton water scale in our boilers, and we finally thought we would try this system as it was brought before us by a well-known engineer. We took a cylinder 30 inches in diameter and 10 or 12 feet high, and placed it so that from shortly above the bottom of it the water would run by natural gravity into the boiler. We carried about 80 pounds steam pressure. At the top of this cylinder we had a sprinkler arrangement perforated with very small holes, and we filled this cylinder pretty well up with coke and connected the boiler—the steam part as well as the water part—with it. Probably the water level was about 6 or 8 feet below the top of this upright cylinder. We ran our boilers in that way, feeding into the top of this cylinder, sprinkling into the steam space and making the water trickle over the coke, and it had absolutely no effect whatever, and there was not a particle of difference in the quantity of lime which we got out of the boiler nor in the trouble which we had from our boilers. Of course it is not a very large quantity of lime which is in Croton water, but we got about a pailful a week, and we had to abandon the system as absolutely worthless. After using it several months we could find no trace of lime on the coke or anywhere excepting in the boiler itself.

Mr. W. H. Odell.—I have had some experience with a device such as Mr. Kent asks about. Several years since I was engaged to superintend the erection and make test of an apparatus in which a branch from the blow-off was connected direct to a "Hyatt" filter and thence returned to the boiler. Circulation was induced with a "Blessing" trap. The very impure feed water was pumped directly into the boiler. Analysis of the feed water before entering the boiler showed the quantity used each day carried into the boiler about four pounds of impurities. These impurities were washed from the filter each twenty-four hours, and after thorough drying were weighed, and each day's collection weighed about four pounds. I cannot remember the exact weight of impurities or the analysis of the feed water, but I called at this place some four years later and was informed that the device had given most excellent satisfaction up to the time the buildings were destroyed by fire.

In the Sweet's rolling mill, at Syracuse, N. Y. (the same city in which the test I have just mentioned was made), they have a somewhat similar device, and Mr. William A. Sweet, proprietor, tells me it is giving most excellent results. In Mr. Sweet's mill I think the feed water is treated and filtered, in part, before entering the boilers, and if I remember rightly the small boiler which heats the feed water before filtering carries a higher steam pressure than the main boilers. Here are two cases from experience, and I might cite others.

Mr. Fred. Meriam Wheeler.—I would like to add one word about the water supply of Syracuse. As Mr. Odell says, they are very much troubled there—the people as well as the boilers—with the city water, and I am glad to learn that they are about securing a new and better water supply. I also know that Mr. Cogswell, of the Solvay Process Company, has spent many dollars in contending with bad water at his works. Mr. Kent's suggestion of getting rid of greasy matter from water by centrifugal action I am afraid would be rather ineffectual. I think this subject of purifying feed water for boilers really a very important one. While the filter which I spoke of (using the coagulating principle) is a very good one it might be improved upon. I am not connected with the manufacture of filters, but I am very much interested in securing a simple and perfect system for cleansing the grease from the feed water as used in connection with surface condensers, more particularly with the high steam pressures.

The Devoe Manufacturing Company have had a surface condenser in use eight or ten years and have had no trouble with their boilers. The engineer every day gives his boilers a dose of lime and soda; but then they use steam only at 60 lbs., which is an entirely different problem from using it at 100 or 120 lbs. pressure. In marine practice they have recourse to salt water very frequently, also the use of soda; but then their boilers, compared with the average stationary boilers, get very much more attention. As soon as a steamer comes into port the boilers get prompt and thorough care the same as the engines. You cannot say that for average stationary engine practice.

Mr. Arthur Falkenau.—I know of an experience in the Edison station in Philadelphia. In that case Professor Marks employed several different filters unsuccessfully. He at last put in some of the Hoppes filters that were just spoken of, but finally resorted to using cutch, a dye stuff, in the well. They have a well under the building. Cutch was put in the well and they used an injection of caustic soda in the heaters every day. That seemed to be amply successful for the purpose.

Mr. John J. Hoppes.—I do not like to blow my own horn, but the gentleman last on the floor made a statement which I think, when he investigates, he will agree with me was incorrect. Mr. Marks did put some purifiers in the building, but they were never connected up and never operated.

I would like, while I am on the floor, to show the principle of our purifier. They speak of it as a matter of precipitation. Our purifier, while it operates partially on the plan of precipitation, is not entirely based on that plan. That is, we do not depend on the plan of precipitation. All other purifiers which have been used depend upon the plan of precipitation or filtration, and some arrangements used in boilers, called skimmers, depend on skimming the top off the water—that is, those precipitates which float on top of the water, usually carbonate of magnesia. Generally the pans of our purifiers are made in a tray form. We have six pans in each tier in a purifier. These pans have extensions resting on ways. The ends of the pans are higher than the sides. The water flows over evenly on both sides of the pan the whole length of the purifier. It flows along the under side of the pan in a thin film, and nine-tenths of the solids caught in the purifier are caught on the under side of the pan, the same as a stalactite is caught on the roof of a cave. We use no filter whatever. We have formations showing as much as three inches taken from the under side of the second pan. They build up on the edge of the pan and run gradually to a thinner formation near the centre. The second pan has the heaviest formation, usually, when the purifier is ready to be cleaned, about an inch and one-half thick. The sixth or bottom pan will have a thirty-second to a sixteenth of an inch formation on the under side. Now waters which have different chemicals in them of course form different scales. The water which is the worst to purify shows a soft formation on the under side of the pans; that shows then that it is composed of sulphates. When soft and white we know then that the solids in the water are sulphates. Sulphates form a hard scale in boilers. Carbonates form a harder scale in the purifier, and we like to handle water which contains carbonates, because in operating purifiers where there are sulphates in large quantity, the least irregularity is likely to wash the sulphates off Where there are carbonates in the water, and also sulphates, you will find carbonates and sulphates mixed in the formation from the second pans down to the last. If silicates are present in the water the last pan will usually show silicates. Sometimes a hard scale on the last pan will show silicate of lime and sulphate of lime combined together. The Beaver Falls plant spoken of I never saw cleaned, but I understood from one of my men who was there that it showed a good deal of vegetable matter, besides the solids in solution. At the Union Ice plant in Pittsburgh large filters were used and an attempt made to filter the water at the temperature of the water in the boiler. They had a cylinder with a spray to feed the water into a steam space connected directly with the boiler. They passed it down through a large filter, almost as large as the Hazleton boilers on which they were placed. It did no good whatever. I do not say that because we afterward handled the water successfully, but simply as a fact. It is not by precipitation but by formation by which we catch the scale in the purifier.

Mr. Jno. T. Hawkins.—Mr. Ashworth has given us a valuable instance of the successful operation of purifiers in certain parts of the country. I think he would add a good deal to the value of his discussion if he would give us in connection with it the analysis of the impurities in these localities. Of course, we can readily understand that the same system might not operate so successfully under other conditions where the impurities were of different character. If he is not prepared to do it now he might incorporate it in the discussion.

Mr. Kent.—I would like to make a suggestion, also, that Mr. Cogswell, of Syracuse, be asked to add to this discussion. He has had much experience in handling one of the worst waters in the country.

Mr. W. B. Cogswell.—In our works for purifying, we use a weak soda liquor, containing about 12 to 15 grams $Na_2 Co_3$ per litre. Say $1\frac{1}{2}$ to 2 M^3. (or 397 to 530 gals.) of this liquor is run into the precipitating tank. Hot water about 60° C. is then turned in, and the reaction of the precipitation goes on while the tank is filling, which requires about 15 minutes. When the tank is full the water is filtered through the Hyatt (4), 5 feet diameter, and the Jewell (1), 10 feet diameter, filters in 30 minutes. Forty tanks treated per 24 hours.

Charge of water purified at once, 35 M^3, 9,275 gallons.
Soda in purifying reagent, 15 kgs., Na_2Co_3.
Soda used per 1,000 gallons, 3.5 lbs.

Analysis of lake water January 1st, 1892:

Calcium sulphate	.261	grams per litre.
Calcium chloride	.186	" " "
Calcium bi-carbonate (as $CaCo_3$)	.091	" " "
Magnesium bi-carbonate (as $MgCo_3$)	.015	" " "
Magnesium chloride	.087	" " "
Salt	.63	" " "

Analysis of mud from Hyatt filter:

Silica	15.17	grams per litre.
Iron and aluminum oxide	8.75	" " "
Calcium sulphate	8.70	" " "
Magnesium carbonate	1.11	" " "
Calcium carbonate	63.87	" " "

Analysis of scale from boiler tube Nov. 14th, 1887:

Silica	2.29	grams per litre.
Iron and aluminum oxide	1.10	" " "
Calcium carbonate	19.76	" " "
Magnesium carbonate	25.21	" " "
Calcium sulphate	51.24	" " "
Na Cl	.14	" " "
	99.74	

Analysis of scale found in pump, pumping from tanks through filters:

Silica	.8	grams per litre.
Iron and aluminum oxide	1.2	" " "
Calcium carbonate	87.	" " "
Calcium sulphate	10.9	" " "
	99.9	

A sample is taken from each boiler every other day and tested for deg. Beamé soda and salt.

If the deg. B. is more than 2, that boiler is blown to reduce it below 2 deg. B.

Samples taken from 12 boilers on Feb. 10th, 1889, when canal water was used for steam, gave the following results on testing for deg. B., Na_2Co_3, Na_2So_4, and NaCl:

Boiler.	Deg. B.	Na_2Co_3.	Na_2So_4.	NaCl.
1.	1.	2.86	3.39	.94
2.	1.8	5.14	6.51	1.31
3.	.8	1.53	1.63	.585
4.	1.6	4.24	5.51	1.52
5.	2.4	6.62	8.97	2.34
6.	1.	2.49	2.92	.906
7.	2.	5.56	7.91	2.77
8.	2.8	8.42	10.36	1.98
9.	1.6	4.45	5.77	1.57
10.	1.2	2.86	3.47	1.02
11.	1.6	4.24	5.9	1.58
12	3.1	6.51	15.8	2.19

				Ca So_4.
Feed pump...............	371	.213	.043	.05

The analysis of the canal water at this time was :

$CaSo_4$..................................	.246	grams	per	litre.
$CaHCo_3$ as $CaCo_3$......................	.031	"	"	"
NaCl..................................	.043	"	"	"
$MgCl_2$..................................	.038	"	"	"

It will be seen that at this time the carbonate of soda and sulphate of soda were present in greatest quantity, and the boilers had to be blown to keep these down in saturation.

This was not the case on January 1st, 1892. The salt in the lake water is now very high. More than twenty times the amount is now present in the lake water, and hence the high deg. B. is caused by the salt more than by the sulphate and carbonate of soda.

The following is test of deg. B., Na_2Co_3, and salt on January 1, 1892:

	Deg. B.	Grammes per Litre.	
		NaCl.	Na_2Co_3.
No. 1..............................	1.0	7.87	.848
No. 2..............................	.8	8.56	.318
No. 3..............................	2.7	17.30	2.96
No. 4..............................	1.9	10.99	1.84
No. 5..............................	2.6	16.66	.42
No. 6..............................	.5	4.09	2.96
No. 7..............................	2.8	17.30	3.71
No. 8..............................	3.4	20.00	4.1
No. 9..............................	3.4	21.52	3.18
No. 10..............................	3.0	18.72	3.00
No. 11..............................	2.7	16.66	3.18
No. 12..............................	2.5	15.08	3.00

It would then be much better to use in the boilers canal water instead of lake water, to avoid this large percentage of salt.

The analysis of the canal water is:

CaSO	.223
CaCl	None.
CaCo	.088
MgCo	.06
NaCl	.04

One man at day and one at night attends to the work of all purification at $2 per man per day.

PURIFICATION OF WATER AT LAKE PUMP.

Amount purified per day (24 hours), 13,000 gallons.

Soda used, 40 lbs. in 24 hours.

Soda per 1,000 gallons, 3¼ lbs.

Filter used a Bunnell, 3 feet 6 inches in diameter, and 5 feet high. Washed twice in 24 hours.

The soda (about 20 lbs.) is dissolved in 90 gallons of water, and this solution is mixed in the top of the filter with water from the hot well and the circulating water from the boilers at 65° C., and is then filtered. Filter washed twice in 24 hours.

There is no scale now in these boilers.

No. 477–95.

Is it better to have the lead increase with the load in high-speed automatic engines, and if so, why?

Mr. Wm. O. Webber.—I think it is better to have the lead increased with the load, for the reason that the impulse necessary to keep up the momentum with the heavier load should be greater at the starting point than with the light load; and in support of this would submit indicator diagrams taken from engine No. 190 (Fig. 56), which had an increasing lead and gave very close regulation; also diagrams from engine No. 179 (Fig. 57) by Edson autograph recording apparatus, showing a regulation under different loads, with the lead increasing with the load; and also diagram from engine No. 189 (Fig. 58), from the same instrument, showing the lead remaining constant, and having no reference to the load. In other words, I have found the regulation to be more constant, and the general performance of the engine better, under careful tests, with the increasing lead in proportion to the load. These diagrams practically answer a query sometimes asked, as to the advantages to be claimed for

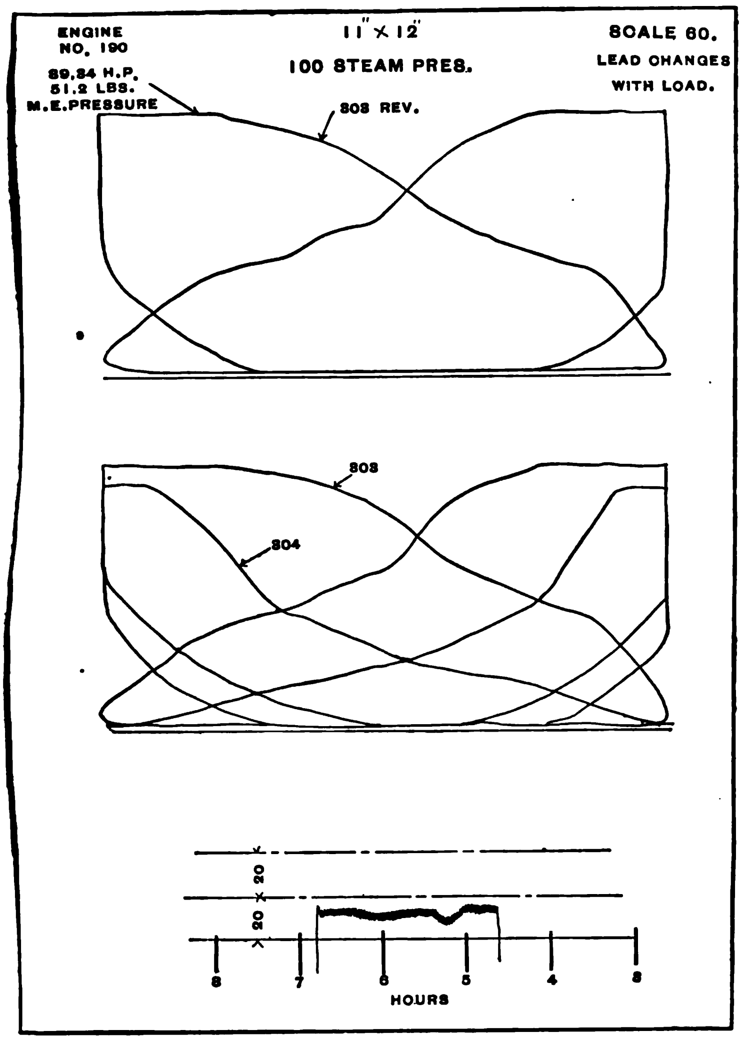

FIG. 56.

a minimum amount of back pressure and a high compression line, as you will see by the diagrams that the compression line is higher; and although no atmospheric line is shown on these cards, it will be noticed there is a slight move upward, showing a varying amount, say two to three pounds back pressure. We think all these points conduce to the proper running of engines, less wear on the reciprocating parts, and greater and better regulation.

Mr. E. J. Armstrong.—The writer would take the affirmative,

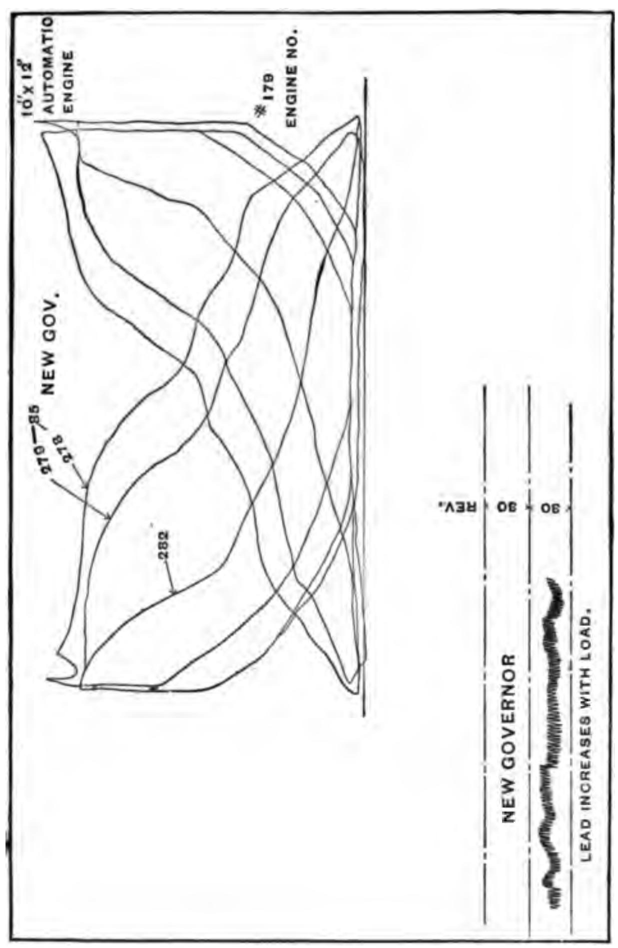

FIG. 57.

believing that, at least with single-valve engines, the advantages to be gained by such an arrangement are well worth the attention of designers. To summarize the most important considerations—

1st. It is an aid to quiet running. At the latest cut-off of a single-valve automatic, the compression is low, and is nearly neutralized by the high terminal pressure on the other side of the piston. With high speeds a sharp lead at this point helps to prevent pounding, and the best amount may be much in excess of that which would be permissible were it to remain constant throughout the governor range. At very early cut-off the compression approaches boiler pressure, and the pressure on the other side of the piston is low, so that lead has little effect, and may be negative without bad results, in some cases with positive gain in smooth running.

2d. It prevents racing. When constant lead is employed, it

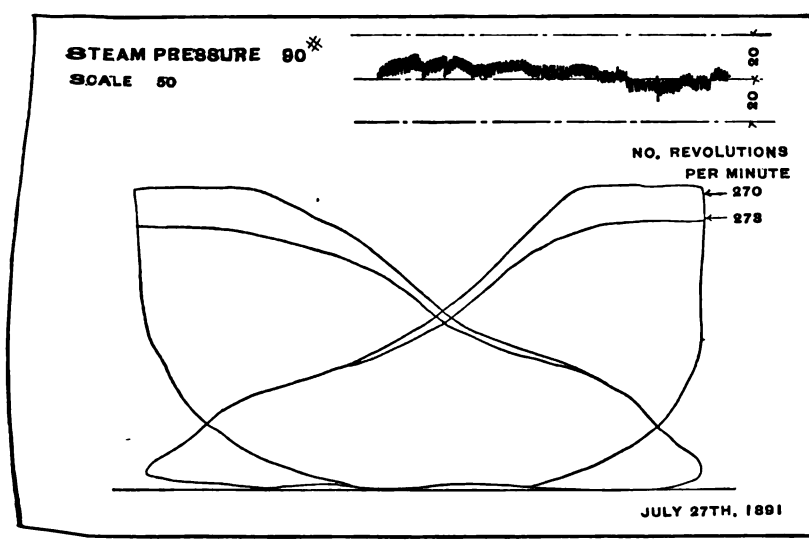

FIG. 58.

must be small in amount. As it becomes impossible for the governor to entirely shut off steam, the valve will at least open the amount of the lead, and if this is considerable, the engine will race when running unloaded. Any misadjustment of the valve, occurring by wear or otherwise, will also cause racing, by admitting steam to one end of the cylinder. A negative lead at early cut-off removes trouble from this source, and by enabling the governor to entirely shut off steam, gives it better control of the engine.

3d. It makes it easier to obtain good governing. Apart from giving the governor better control, there is an advantage in that the centre of the eccentric does not have to shift through so great a distance, consequently the governor may have greater leverage over the eccentric, and as a result, greater stability.

4th. It tends to equalize the compression, and the exhaust opening. The angular advance of the eccentric being slightly greater at late cut-off than with constant lead, the exhaust closes earlier, and the compression is a little higher. At early cut-off the angular advance is very much less than with constant lead, and the exhaust closing later, the compression does not run nearly so high. The exhaust does not open so prematurely at early cut-off as with constant lead, and opens earlier at late cut-off. The most marked effect on the cards is the more nearly equal compression, which by permitting less clearance, should be conducive to economy. At early cut-off the steam line is a little lower, and slopes off to the expansion curve a little more. Though this does not produce so pretty a card, it is an open question whether it is bad or not. At early cut-off there is so much difference between the water consumption as shown by the card and by the meter, that the cylinder condensation must be very great. Possibly the slight wire-drawing may reduce this enough to balance the loss in card area.

CCCCLXXVIII.

In Memoriam.

ALFRED CHARLES HOBBS.

Born, October 7, 1812; died, November 5, 1891.

IT is with profound sorrow and the grief of a personal loss that I announce to the American Society of Mechanical Engineers the death of one of its most distinguished members, Alfred Charles Hobbs.* This sad event occurred at Bridgeport, Conn., on the 5th day of November. This Society will honor itself by bestowing somewhat more than a passing glance upon his honorable character and eventful career, for his experiences and abundant success are peculiarly calculated to stimulate to greater enthusiasm and exertion the younger members of the profession, and cannot fail to be of encouraging and profitable interest to us all.

Our deceased friend was born in Boston, October 7, 1812. When he was about three years of age his father died, leaving his young family destitute. Young Hobbs lived with his mother until ten years of age, "going to school occasionally, playing truant quite often (as he himself states †), and, in many ways trying to earn a few pennies which went into the general fund for the family support."

In February, 1822, he left his home and entered the service of a farmer in Westfield, Mass., but at the end of four years "he had learned all he desired to know of farming," and returned to Boston to find another occupation. After a brief period spent

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* This memorial paper was in course of preparation during the XXIVth Meeting of the Society in New York, and I fully expected to present it, but was prevented by circumstances beyond my control.

This sketch of the life and character of a noble man must of necessity be incomplete. Nothing short of the ample scope of a biography, in which the varying fortunes and remarkable achievements of his career are set forth in detail, could do justice to his memory.—W. F. DURFEE, *West New Brighton, N. Y.*

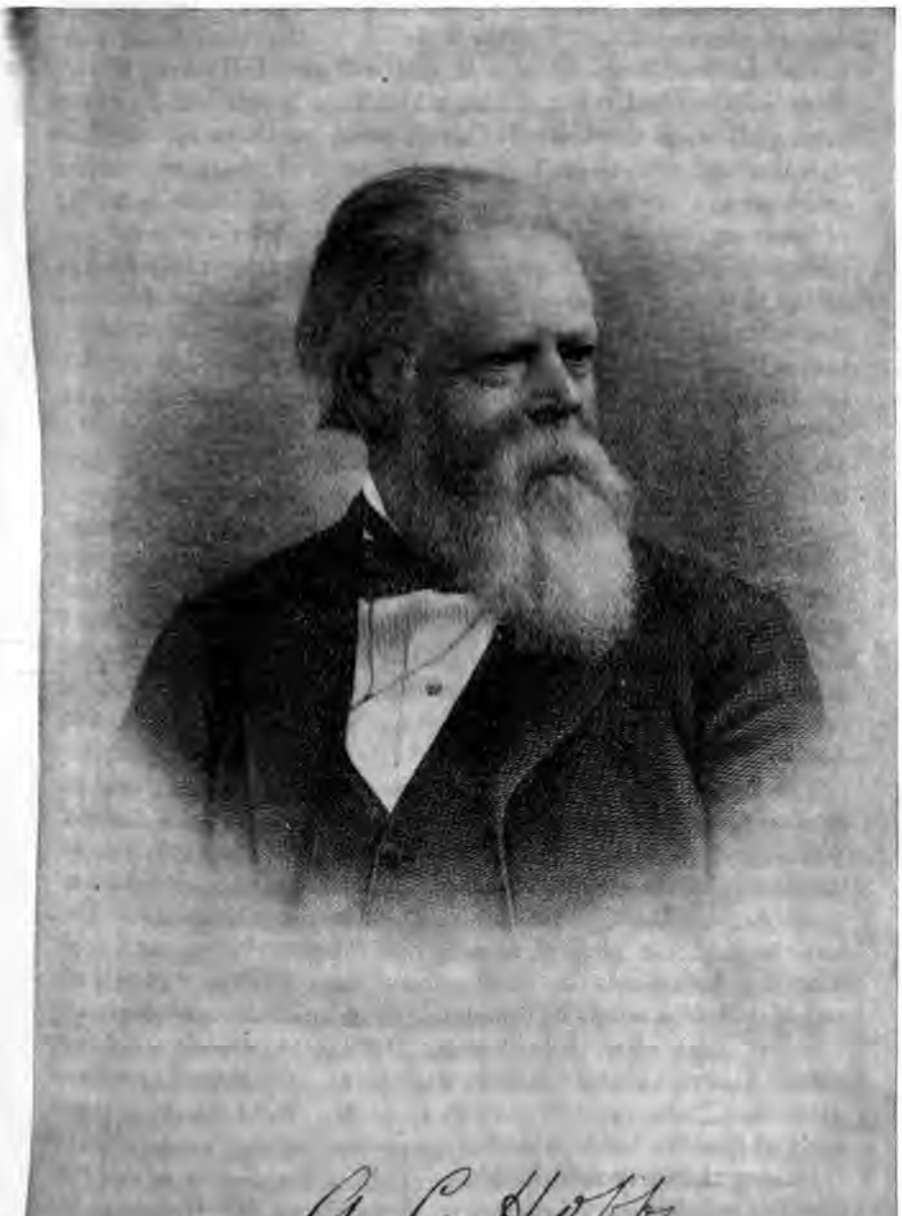
† *Transactions* of the Fairfield County Historical Society, 1887.

in a dry-goods store he went to learn the trade of wood-carving of an older brother; this did not suit him, and then he tried his hand at carriage-body making, but his employer failed, which caused him to change to the rudiments of carriage painting, which he describes as consisting of "rubbing off the paint from old carriages with pumice-stone and cold water." This employment not being to his mind, he next attempted to become a sailor, and secured a berth on the ship *Leonidas*, bound for Charleston, S. C., and from thence to Havre, France; but the voyage to Charleston and return to Boston satisfied him that he did not care to go to sea before the mast. He then tried "tin plate work," and later "coach trimming and harness making"; neither of which was agreeable to his fancy. At sixteen years of age he left Boston and went to Sandwich, Mass., and served an apprenticeship at glass-cutting in the works of the Boston and Sandwich Glass Co. When he was twenty-four years old he returned to Boston and established the glass-cutting business in a building on Bromfield Street. A part of his work was the cutting of glass door knobs, and he invented and patented a method of fastening them into the socket by which they were attached to the locks; this brought him for the first time in contact with some of the lock makers, and he was induced to enter that business as the junior partner of the firm of Jones & Hobbs, but this association was very brief and unsatisfactory.

During his residence in Boston at this time, which covered a period of five years, he was an active member of the fire department, the Washington Light Guard, the Massachusetts Charitable Mechanics' Association, and President of the Boston Musical Education Society.

On the dissolution of the firm of Jones & Hobbs, the junior went to New York, under an agreement with Edwards & Holman, to sell their locks and fire-proof safes.

This business did not prove agreeable to him, but during the time he was connected with it he carefully studied the construction of locks, and came to the conclusion that most of those in use were of very little value. His association with Edwards & Holman was not of long duration, and at its close he made arrangements with Messrs. Day & Newell, then prominent as bank lock makers in New York, to take the entire charge of selling their bank locks. In order to succeed in this business, it was necessary to prove to bankers that the locks which they had in use


A. C. Hobbs

were not secure. He scored his first success in this at Stamford, Conn., in January, 1847, by opening, in 23 minutes, the outside door of the bank, and the three locks on the vault. He continued in this business of selling the lock of Messrs. Day & Newell for several years, during which time he travelled over a large portion of the United States, going south as far as New Orleans, and west to St. Louis, and to most of the larger cities and towns in the country. While thus engaged he, as a matter of business, opened the locks of his competitors whenever opportunity offered, and he, in consequence, became widely known as the most accomplished lock expert in America; but his successes in his native land were destined to be far surpassed by his brilliant achievements abroad. With a view to exhibit the Day & Newell lock at the first International Exhibition in London, and, if favorable opportunity offered, testing the much-vaunted security of certain prominent English locks, Mr. Hobbs left New York in April, 1851; on his arrival at Southampton he was welcomed by the American consul, who, after hearing, under a pledge of secrecy, what Mr. Hobbs proposed to do, said "it was the first time he ever heard an American desire to have anything he knew kept quiet," and that he would "be delighted if anything could be done to raise the standard of the American department of the Exhibition, as it then consisted mainly of a few barrels of shoe-pegs, some bunches of brooms, and a few American carriages."

The locks of Day & Newell at once attracted a great deal of attention, and Mr. Hobbs received several calls from the Duke of Wellington. A short time after one of these visits, at one of the Queen's drawing-rooms, the Duke of Wellington (who was standing near the Queen) said to the American Minister: "Mr. Lawrence, I am pleased to see you. I have seen the great American lock—it is one of the finest things in the Exhibition, and Mr. Hobbs is one of the cleverest of men."

A few days after, Her Majesty the Queen, accompanied by Prince Albert, several ladies in waiting, and the Prince of Wales, with attendants, came to see the lock, Mr. Hobbs having been notified the day before that they were coming.

During the early months of the exhibition a variety of rumors had become active in regard to the skill of Mr. Hobbs as an opener of locks; but it was not until July 22d, three months after his arrival in England, that he gave the first public illustration of it, by opening a "Chubb lock," having three "bolts" and six "tumblers," which was affixed to the door of a "strong room" at 34 Great George Street, Westminster. This he accomplished with his instruments (never having seen the key) in 25 minutes, and in response to a request he relocked the door in 7 minutes, neither operation occasioning the slightest injury to door or lock.

This success was followed by the triumph of opening the famous "Bramah lock," which had been the despair of a generation of locksmiths, and receiving from Messrs. Bramah by order of the arbitrators (George Rennie, Edward Cowper, and J. R. Black) the 200 guineas reward offered for so doing.* On the same day that this lock was opened (August 23, 1851) the yacht *America* won the international race at Cowes. These two events very much increased the reputation of the United States at the World's Fair, and "shoe-pegs" and "brooms" ceased to be regarded as typical representatives of our ingenuity and manufactures.

The Hon. Abbot Lawrence, at that time United States Minister to England, was so much pleased with the opening of the Bramah lock that he gave Mr. Hobbs a dinner at his residence, inviting the three arbitrators, Mr. George Peabody the American banker, and a number of other American gentlemen. Mr. Hobbs always took great pleasure in speaking of the kind attention and courteous treatment which he received in the several contests he had with his many competitors during the time of the World's Fair in London, and of the many evidences of an appreciative consideration of his work. His success in showing the insecurity of the locks then in use in England, which were mostly made by hand labor, suggested the starting of a factory for the manufacture of locks by machinery in accordance with the most approved American methods. With this end in view, a partnership was formed, and the firm of Hobbs, Ashley & Co. commenced business in Cheapside, London. At first, progress was discouragingly slow, but the second year was more promising, and at the end of the third year a new building was required for the factory, which was built outside of the city, the building in Cheapside being retained as a store and salesroom.

* Some account of this period of Mr. Hobbs's life has been given in his own words before the Society, and will be found in Volume V., p. 123, and Volume VI., p. 283.

The firm of Hobbs, Ashley & Co. received an order for furnishing the locks required for the steamship *Great Eastern*. This gave Mr. Hobbs opportunity frequently to visit the ship during its construction, and he was invited to join the party who were to have been launched in the ship; but, as that event unexpectedly consumed several weeks, he was not present all of the time, but was on board during the trial trip, and witnessed the explosion of the feed-pipe casing of one of the boilers, that resulted in a total wreck of the dining-room and an amount of other destruction which would have sent any ordinary vessel to the bottom of the sea. As time passed, the business of Mr. Hobbs's firm increased, and the mechanical appliances at the factory were largely augmented, a new store was added, and the firm received a large portion of the business of the Bank of England, which added very much to their reputation and standing.

In August, 1860, Mr. Ashley died, and then Mr. Hobbs welcomed an opportunity to sell out and return to America. A very satisfactory arrangement was made, one of the terms being that the name of Hobbs should stand at the head of the new firm, which became Hobbs, Hart & Co. Twenty years after this firm was formed, I called at its place of business, 76 Cheapside, and inquired for Mr. Hobbs, and was told by a dignified young salesman that he was "not in"; and in response to my query as to when he would probably return, he replied: "It is quite uncertain, as he is out of the country at present." It was quite evident that there was a method in these answers, and that the name of Hobbs still exerted a potent influence as the basis of the business reputation of the firm.

Hobbs, Hart & Co. have (since the death of Mr. Hart in March, 1887) become a limited liability company; the word limited being added to the old firm name. They have the largest establishment for the manufacture of bank locks and safes in England, and there is no more prosperous manufacturing business in London than that of Hobbs, Hart & Co., Limited, 76 Cheapside.

During his residence abroad, Mr. Hobbs made many friends and formed numerous agreeable associations. He became a member of the Society of Arts, and by request delivered before it a lecture on ancient and modern locks, and he was frequently called upon to address similar local societies in the larger towns of England.

Prof. Airey, then Astronomer Royal, in speaking of Mr. Hobbs's peculiarly happy manner of expressing himself, said that he reminded him "more of Prof. Faraday than any one he had ever listened to."

Alfred Charles Hobbs. Assoc: Inst: C. E.

Telford Premium. 1854.

Mr. Hobbs was elected an associate of the Institution of Civil Engineers, to which he contributed a paper on the "Principles and Construction of Locks," and was awarded the Telford Pre-

It would then be much better to use in the boilers canal water instead of lake water, to avoid this large percentage of salt.

The analysis of the canal water is:

CaSO	.223
CaCl	None.
CaCo	.088
MgCo	.06
NaCl	.04

One man at day and one at night attends to the work of all purification at $2 per man per day.

PURIFICATION OF WATER AT LAKE PUMP.

Amount purified per day (24 hours), 13,000 gallons.

Soda used, 40 lbs. in 24 hours.

Soda per 1,000 gallons, 3¼ lbs.

Filter used a Bunnell, 3 feet 6 inches in diameter, and 5 feet high. Washed twice in 24 hours.

The soda (about 20 lbs.) is dissolved in 90 gallons of water, and this solution is mixed in the top of the filter with water from the hot well and the circulating water from the boilers at 65° C., and is then filtered. Filter washed twice in 24 hours.

There is no scale now in these boilers.

No. 477-95.

Is it better to have the lead increase with the load in high-speed automatic engines, and if so, why?

Mr. Wm. O. Webber.—I think it is better to have the lead increased with the load, for the reason that the impulse necessary to keep up the momentum with the heavier load should be greater at the starting point than with the light load; and in support of this would submit indicator diagrams taken from engine No. 190 (Fig. 56), which had an increasing lead and gave very close regulation; also diagrams from engine No. 179 (Fig. 57) by Edson autograph recording apparatus, showing a regulation under different loads, with the lead increasing with the load; and also diagram from engine No. 189 (Fig. 58), from the same instrument, showing the lead remaining constant, and having no reference to the load. In other words, I have found the regulation to be more constant, and the general performance of carriage to the city of Corinth; thence by steamboat down the Gulf of Lepanto to the Island of Zante, and from there to Cephalonia, where, in the harbor, he saw a wonderful stream of water running into a mountain and furnishing power for two flouring mills. From Cephalonia he went to the Island of Corfu, crossed the Adriatic to Brindisi, and from there he went to Naples, arriving just in time to witness an eruption of Mount Vesuvius which threw cinders and ashes into the streets of Naples. After visiting the ruins of Pompeii he went to Rome, Florence, Paris, London, and Liverpool, and thence home, where he arrived in the middle of June, 1872.

About ten years after this Mr. Hobbs made another journey to Europe, going again to Italy, and spending some months in that country and France. After his return he devoted himself with renewed energy to the affairs of the Union Metallic Cartridge Co., which yearly increased its powers of production, keeping well in advance of its business competitors; it has achieved a world-wide reputation for the volume and perfection of its products, and justly ranks among the most successful of the manufacturing establishments of America.

Mr. Hobbs remained in active charge of all its manufacturing interests until within about a year of his death.

Thus much for our friend as a mechanician, manufacturer, and traveller; I come now to speak of him as a neighbor, friend, and citizen.

Soon after taking up my residence in Bridgeport, Conn., in 1868, I found that A. C. Hobbs was my nearest neighbor, and it was not long before acquaintance ripened into an intimacy and friendship which lasted for over twenty years and which, I trust, has only been interrupted by his death.

I had not been his neighbor many months, when he suggested that "it would be a great deal handier if we had a gate in the board fence between our gardens"; and he proposed to "furnish the lock and hinges," if I would have the gate made. This proposal I gladly accepted—the gate was promptly opened, but during my residence there it was never closed. Shortly after my removal to Philadelphia, in October, 1871, he wrote me, "That gate is locked and I have thrown away the key"; and now he has passed the gate in the boundary of life and has thrown away its key, and no skill of the master of the mysteries of locks will enable him to repass that portal and visit us as of old.

a varying amount, say two to three pounds back pressure. We think all these points conduce to the proper running of engines, less wear on the reciprocating parts, and greater and better regulation.

Mr. E. J. Armstrong.—The writer would take the affirmative,

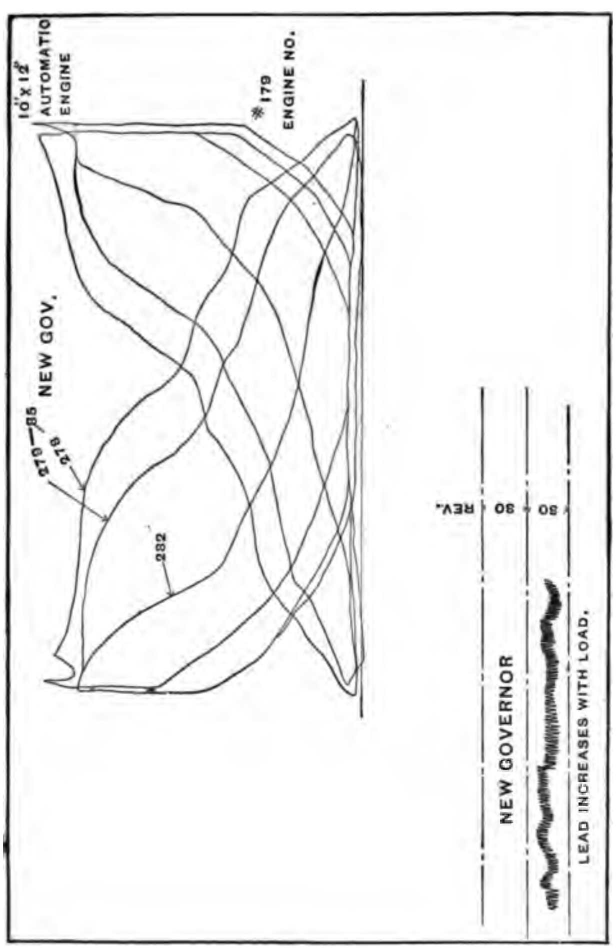


FIG. 57.

believing that, at least with single-valve engines, the advantages to be gained by such an arrangement are well worth the attention of designers. To summarize the most important considerations—

1st. It is an aid to quiet running. At the latest cut-off of a single-valve automatic, the compression is low, and is nearly realized in practice many times, every working moment of the day.

He was endowed with a ready wit, which on occasion he could use with a sarcastic effect rarely equalled. The following is illustrative of this: He was endeavoring to sell a lock to an officer of a bank who manifested a most exasperatingly penurious disposition, and as Mr. Hobbs was reassembling the lock, after having taken it apart, to show its construction, this official wished to know in detail the cost of each piece; and as a price was named he made a note of it. As soon as the lock was assembled he very querulously said: "Mr. Hobbs, these prices foot up so much, and even that seems a great deal to pay for so few pounds of brass and iron; now, how do you account for the large additional sum you ask for the lock?" The answer was both prompt and decisive: "We charge that for brains, sir, and we don't sell *them* by the pound." Of a more genial flavor were his words accompanying the presentation of a quarto dictionary to a friend: "I could not select and arrange words to express the kind feelings we have toward you and yours; I have therefore sent the whole vocabulary."

In looking backward in review of the life of our friend, we are most appropriately reminded that "beautiful upon the mountains are the feet of him who bringeth glad tidings." His quiet, unobtrusive charities, the natural outcome of his kindly nature, were innumerable; for into the lives of multitudes of strugglers against the slings and arrows of outrageous fortune, who were in sore need of such glad tidings as he could bring, his timely acts fell as gratefully as the shadow of a great rock in a weary land, and his kindly, encouraging words, those "apples of gold in pictures of silver," were the spontaneous benediction of a generous heart. Surely, these benefactions will find their just recognition in that record which, in the fulness of time, will read: "Inasmuch as ye did it unto the least of these my brethren, ye did it unto me."

For all those who counted him as a friend, the sweetest flowers of memory will ever bloom reminiscent of his life; faith in his future will always be firm, and belief in the certainty of a final reunion unshaken.

So, lovingly, reverently, and trustingly, they think of him as among those having conquered the ills of earth and time, who with enlarged perceptions and augmented powers are filling the place, and doing the work, assigned them in heaven and eternity.

"Alas for him who never sees
The stars shine through his cypress-trees:
Who, hopeless, lays his dead away,
Nor looks to see the breaking day
Across the mournful marbles play!
Who hath not learned, in hours of faith,
The truth to flesh and sense unknown;
That Life is ever Lord of Death,
And Love can never lose its own!"

WEST NEW BRIGHTON, N. Y., *November* 19, 1891.

CCCCLXXIX.

APPENDIX III.*

MINUTES OF THE THIRD CONFERENCE FOR THE UNIFICATION OF STANDARD METHODS OF TESTING MATERIALS OF CONSTRUCTION, HELD AT BERLIN, GERMANY, SEPT. 19–20, 1890.

TRANSLATED BY GUS. C. HENNING, M. E., REPORTER FOR COMMITTEE ON STANDARD TESTS AND METHODS OF TESTING MATERIALS, OF THE AMERICAN SOCIETY MECHANICAL ENGINEERS.

LIST OF PARTICIPANTS.

Achen, G., Cementtechniker, Braunschweig.
Aron, Dr. Julius, Berlin, Kruppstr. 6.
Bach, C., Professor, Stuttgart.
Bauschinger, Professor, München.
Belelubsky, Professor, St. Petersburg.
Bernoully, A., Director, Wildau.
Bienfait, L., Engineer, Amsterdam.
Boehme, Dr., Professor, Berlin.
Böttcher, Intendantur und Baurath, Berlin, W.
Büsing, Professor, Berlin.
Candlot, Ingenieur-Chimiste, Boulogne-sur-mer.
Caspar, C., Betriebs-Ingenieur der Münzstätte, Hamburg.
Cramer, R., Ingenieur, Berlin.
Debray, Professor, Ingenieur des Ponts et Chauss., Paris.
Delbrück, Commercienrath, Stettin.
Dümmler, K., Architekt, Vertreter der Deutschen Töpfer-u. Ziegler-Zeitung. Berlin N. 39, Fennstr. 40.
Dyckerhoff, R. Cementfabrikant, Amöneburg.
Eckerman, Ingenieur, Hamburg.
Erhard, L., Ingenieur, Nürnberg.
Feussner, Dr., Mitglied der Phys.-Techn. Reichsanstalt, Charlottenburg.
Von Forell, Director, Braunschweig.
Gaertner, Ingenieur, Wien.
Glein, C. O., Ingenieur, Hamburg.

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* This appendix to the Report of a Committee of the American Society of Mechanical Engineers, was presented for record at the New York Meeting, November, 1891. It attaches itself to the Report No. 380, and to Appendix II, No. 378 of the transactions of that Society, published at pages 527 and 604 of its Volume XI.

Greil, Alfred, Ingenieur, Leiter der städt. Prüfungsanstalt, Wien.
Guyot, Geh. Baurath, Wilhelmshaven.
Haack, Civil-Ingenieur, Berlin.
Haage, Cl. Ingenieur, Chemnitz.
Herfeldt, G., Trassgrubenbesitzer, Andernach.
Hesse, Fabrikant, Frankfurt-a.-M.
Hoehmann, Regierungsbaumeister, Berlin.
Hoffman, E. H., kgl. Kreisbaumeister, a. D., Berlin, N. Kesselstrasse 7.
Junk, C., Baurath, Charlottenburg.
Kayser, P., Baumeister und Vorstand der Prüfungsanstalt für Baumaterialien an der königl. Baugewerkschule, Dresden.
Kick, Fr., Professor, Prag.
Kirsch, Ingenieur, Wien.
Klebe, C., Assistent der k. techn. Hochschule, München.
Kuchinka, k. u. k. Schiffbau-Oberingenieur, Pola.
Kuhnow, A., Architekt, Berlin, Schäferstr. 14/II.
Kuntze, Eisenbahn Bau-Inspektor, Berlin.
Lämmerhirt, Bau-Inspector, Hamburg.
Leyde, O., Ingenieur, Berlin.
Liedtke, H., Vorsteher des Laboratoriums der Deutschen Töpfer- und Ziegler-Zeitung, Berlin N, Kesselstr. 7.
Martens, A., Prof. Vorsteher der kgl. mech.-techn., Versuchsanstalt, Berlin.
Meyer, C. W., Ingenieur, Berlin.
Meyer, Gustav, Eisenbahn-Bauinspector, a. D., Berlin.
Michaëlis, Dr. W., Cementtechniker, Berlin.
Middendorf, Director des Germ. Lloyd, Berlin.
Mohr, H., Maschinenfabrikant, Mannheim.
Molien, Ingenieur der kgl. Geschützgiesserei, Spandau.
Mouths, Lieut. bei der kgl. Geschützgiesserei, Spandau.
Naske, Ingenieur, Hamburg.
Olschewsky, W., Ingenieur, Berlin, Kesselstr. 31.
Otto, H., Oberingenieur, Essen a. d. Ruhr.
Pohlmeyer, V., Eisenbahn-Director, Dortmund.
Von Prondzynski, Generaldirector, Groschowitz.
Prüssing, Dr., Director der Portland-Cementfabrik, Rüdersdorf.
Pummer, G. A., Ingenieur, Neuberg.
Rautschka, Ober-Ingenieur der Nordbahn, Wien.
Richard, H., Professor, Karlsruhe.
Rudeloff, M., Ingenieur, Charlottenburg.
Scharowsky, Civilingenieur, Berlin.
Schertel, F., Ingenieur, Hamburg.
Schott, F., Director, Heidelberg.
Seger, Dr. H., Charlottenburg.
Skarbinski, Fabrikdirector, Grodriec, Russland.
Von Tetmajer, L., Professor, Zürich.
Toepffer, Cementfabrikant, Stettin.
Toller, Regierungsbaumeister, Dresden.
Tomey, Dr. A., Director, Finkenwalde.
Wallé, P., Architect, Berlin SW. 48.
Weyrich, Baumeister, Hamburg.
Wijkander, Aug., Professor, Göteborg, Schweden.

MINUTES OF THE FIRST DAY, SEPT. 19, 1890.

Called to order at 9:30 A.M.

Prof. Bauschinger, as Chairman of the second Standing Committee, takes the floor to welcome the members present and to explain the reasons why the third Conference had only now been called, instead of in September, 1888.

He furthermore points out that the order of business for the Convention would be the discussion of the reports of the eighteen problems which had been submitted to the second Standing Committee for further consideration, and which were then before the meeting.

In addition to these Mr. Tetmajer has prepared a Report on Changes in the Recommendations for Testing Cast-iron, and another on Wire and Wire Ropes, which had been previously printed and submitted to the members.

Mr. Roussel (Belgium) has presented a paper on Tests of Tires, and Mr. Rotter (Vienna) one on Drop Tests, which were, perhaps, to be eventually published by the Committee.

After making these announcements, Prof. Bauschinger requests that a chairman be elected, and the following were elected unanimously:

Chairman for both days of the convention:
PROF. BAUSCHINGER, Munich.

Vice-Chairmen:
PROF. KICK, Prague;
PROF. TETMAJER, Zürich.

As Secretaries:
MR. B. KIRSCH, Vienna;
MR. C. KLEBE, Munich.

The Chairman then proceeds to state the First Problem for discussion before the convention:

Examination of Pummer's shackles, as to their applicability in horizontal machines, and of other forms applicable for rounds and flats, which may be presented.

The following motion was carried:

Pummer's holders will answer only when provided with spherical bearings which allow initial adjustment under small loads. An axial position of the knife-edges is not secured with absolute certainty, as the relative positions of the bearings and knife edges may change, because the position of the former may change in a direction normal to the latter. For this reason the recommendation of the use of Pummer's holders shall be rescinded.

THE SECOND PROBLEM.

Determination of the manner in which the effect of time on tests is to be allowed for, is considered and the following resolutions are passed, upon presentation by Prof. Martens of a comparison of instructions for testing as laid down by the Society of German Metallurgists and the International Association of Boiler Inspection Societies :

1. To take account of the effect of the rate of testing by special instruction or rules, which are to be determined by the new standing committee.

2. To agree to the method adopted by the metallurgists of measuring extensions between two fixed gauge marks, excluding those tests in which rupture does not occur within the central third.

3. To measure the elongation of flat test pieces by using the "symmetrical method" (dividing in c. m. divisions), and then on one wide as well as both narrow sides, recording the elongation as obtained from the flat side and the average obtained from the two others separately.

4. To determine the necessary and sufficient accuracy of the machines and apparatus and results of tests, and to intrust this work to the new standing committee.

After a recess of one hour for lunch the Chairman announced that the Directors of the Royal Testing Laboratories at Charlottenburg had kindly invited the members present to inspect the institutions on Sunday morning.

THE THIRD PROBLEM.

To keep track of and test new designs of drop-test apparatus which may be proposed, and to collate further development of drop tests with a view of establishing a standard method of test.

The Conference, upon motion of the sub-committee, agrees :

1. To desist from prescribing a standard form of drop-test apparatus.

2. To recommend that the latter be constructed of iron.

3. To adopt a drop-weight (ball) of 1,000 kilos as the standard, using 500 k. in exceptional cases only, being in accordance with the directions of the railway managements, and because it is advisable to erect the apparatus in-doors.

4. That the drop-test apparatus should be provided with a device to insure accurate adjustment of the ball to any desired height of drop.

5. That the Chairman of the Standing Committee requests the association of German railway managers, as well as foreign boards, to submit to the Standing Committee the results of drop tests obtained by their experiments.

FOURTH PROBLEM.

Investigation of proper methods for abrasion tests (hardness and toughness).

1. One test is not sufficient to determine abrasion; and

2. Abrasion tests should be made in such a way that the manner of actual use of the material under investigation be reproduced as closely as possible.

The difficulty of this problem makes it appear desirable to abstain from its further consideration by the Conference and that it be intrusted to the Standing Committee.

FIFTH PROBLEM.

Collection of all available matter to elaborate a standard proceeding for piece tests, not of axles alone, but also of all products of iron and steel. Considerations of the availability of drop and other testing machines in establishing standard forms, for the purpose of making piece tests.

This problem, on motion of the Chairman, is recommitted to a new committee which takes proper interest in the subject.

SIXTH PROBLEM.

Determination of the conditions which are to be fulfilled by a slowly operating mechanical contrivance, which is to be used for making bending tests, is left unsolved, as, according to the explanations of the reporter, opinions have not yet been sufficiently settled. The problem is recommitted to the new committee, which is to make comparative tests with known or possibly new devices.

Discussion of Problem Seven.

Propositions of standard methods for testing copper, bronze, and other metals, led to the propositions given below after the Conference had adopted the following resolutions:

1. The importance of the crushing test is generally admitted; but as sufficient data have not, as yet, been obtained, this test is not now recommended, and the matter is referred to a committee.

2. Similarly, the bending test of a threaded piece is not recommended, as this test may be strongly influenced by the manner of execution.

3. The tension test-pieces are to be draw-filed and then polished with fine emery paper.

The propositions for the method of testing are as follows:

I. COPPER.

To determine the qualities of copper plates, sheets, bars, and wire, the following tests are to be made:

A. Copper in the form of plates, sheets, and bars:
1. Tension test.
2. Bending test, cold.
3. Bending test, hot.

B. Copper wire:
1. Tension test.
2. Bending test.
3. Winding or torsional test.

Condition of Material.

Tests are to be made of the material as delivered, or, if desired for *A*, in softened condition. In order to determine the qualities of the material in its natural condition, test-pieces must be annealed. To do this the strips out of which test-pieces are to be shaped are to be subjected, before finishing, to a heat of from 600° to 700° C., but not above this, then allowed to cool off to blackness (disappearance of glowing) in the air, and then cooled off in water of a temperature of 15° C.

Selection of Test-piece.

Test-pieces are always to be sawed out or else filed, or cut from the piece by a machine tool, particular attention to be paid to do as little straightening afterward as possible. When this becomes necessary it must be done with greatest care, and, if possible, by means of wooden or copper mallets. When the material is to be tested after annealing, the test-pieces may be straightened hot between roughing and finishing. But, in all events, they must again be annealed before testing.

Form and Preparation of Test-piece.

The Conference reserves determining the shape and dimensions of test-pieces for the present, as it is desirable to reduce the cost of testing such an expensive material as much as possible, and as comparative tests to determine them have not yet been completed. Until such time, however, the standard shapes adopted at former meetings shall be adhered to for tension tests. Inasmuch as the manner of preparing test-pieces of copper may have extraordinary influence on the results of tests, shaping the rough sample must be done with greatest care, and all cuts must be run out without interruption beyond the gauge marks, and the finishing cuts must be very light. The specimens are to be draw-filed and then ground or rubbed down with emery paper. All corners of samples for hot and cold bending tests should be carefully filed round at the bending point.

It is advisable to round the edges of flat test pieces for tension tests to a radius of 1 mm. ($\frac{1}{25}$ inch).

Method of Testing.

Tension test: After passing the elastic limit ("yield point") the rate of straining is to be so regulated as to produce 2% elongation per minute.

Elongation is to be measured as determined by previous resolutions.

In the cold bending test a stud whose diameter is equal to the thickness of the sheet, plate, bar, or wire is to be used, and the thermometer must not be below 50° Fahr. Test-pieces taken from plates, sheets, and bars which can be bent around the stud 180° without breaking, are to be closed down until the inner surfaces touch. Bending tests of wire are to be made as heretofore prescribed for wires.

Hot bending tests shall be made with samples heated to a cherry red, and to be carried until cracks appear or until the inner surfaces touch. Winding or torsional tests are to be made as previously prescribed for wire tests.

II. OTHER METALS AND ALLOYS.

To determine the qualities of other metals and alloys used for machinery, superstructures, railways, and ship-building, the following tests are considered necessary:

1. Tension test.
2. Compression test.
3. Transverse test.
4. Bending tests, hot and cold.

According to the qualities of the material to be tested the tests are to be similar to those prescribed by the previous Conferences for cast-iron or copper. When resembling cast-iron, tests 1–3 are to be made; when copper, tests 1, 2, and 4 are to be selected.

Hereupon Prof. Baushinger submits for discussion new propositions by Prof. Tetmajer relative to testing cast-iron; the necessity for these has been made apparent by the fact that the propositions 2 and 3, page 21 of the "Resolutions of Conferences of Munich and Dresden," have proven themselves to be unsatisfactory for casting test-pieces.

This method of casting has demonstrated that blow-holes, rough surfaces, and pieces of slag frequently remain at such points of the test-piece which are of the greatest importance in transverse tests. Several foundries have, indeed, absolutely refused to cast according to the prescribed method. The following method for casting was therefore proposed and accepted, viz.:

2. Test bars are to be cast in a flask slightly inclined, rising about 1 in 10.

3. The pressure under which the bars are cast, as measured by the gate, is to be 20 cm. (7.85 inches).

Upon proposition by Mr. Belelubsky it is agreed that test bars are to be provided at each end with a prolongation having a cross-section 25x25 mm. (1 inch square), out of which, in case of necessity, cubes 25 mm. (1 inch) high are to be obtained for compression tests.

It is further recommended that hereafter tension test-pieces be turned down to 18 mm. diameter (.711 inch) instead of 20 mm. (.790 inch).

In reference to

PROBLEM XVIII.

Comparison of standard shapes of tension test-pieces, the Chairman states that examinations relative thereto had been presented by Messrs. Berndt, Pohlmeyer, Kirsch, and Tetmajer. These are to be published with others now in course of completion.

In connection herewith Mr. Belelubsky (St. Petersburg) delivered a lecture on the enunciation of formulas for the influence of the shape of tension test pieces on their elongation and contraction placed on the determinations of two Russian engineers. This lecture appears as Appendix IV. to these minutes.

Finally the Chairman advocates the inauguration of a publication devoted to the announcements of the Standing Committee, and in general to such technological subjects in direct relation to testing in general, and the Chairman promises to bring up this subject for discussion at the meeting to be held the following day.

Meeting adjourns at 4 P.M.

Signed:

BAUSCHINGER.
KICK.
TETMAJER.
KIRSCH.
KLEBE.

MINUTES OF THE SESSION HELD ON SEPT. 20, 1890.

Chairman:
PROF. BAUSCHINGER, Munich.
Vice-Chairmen:
PROF. BELELUBSKY, St. Petersburg;
PROF. P. DEBRAY, Paris.
Secretaries:
ENGINEER GREIL, Vienna;
ENGINEER OLSCHEWSKY.

Before taking up the regular order of business, Dr. Haack makes the motion that the Conference find ways and means by which to determine the abnormal behavior of cast malleable iron (*flusseisen*, low steel), which is of frequent occurrence and cause of failure, in spite of the fact that test-pieces taken from the ends had given entirely satisfactory results.

inner surfaces touch. Winding or torsional tests are to be made as previously prescribed for wire tests.

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In connection herewith Mr. Belelubsky (St. Petersburg) delivered a lecture on the enunciation of formulas for the influence of the shape of tension test pieces on their elongation and contraction placed on the determinations of two Russian engineers. This lecture appears as Appendix IV. to these minutes.

Finally the Chairman advocates the inauguration of a publication devoted to the announcements of the Standing Committee, and in general to such technological subjects in direct relation to testing in general, and the Chairman promises to bring up this subject for discussion at the meeting to be held the following day.

Meeting adjourns at 4 P.M.

Signed:

BAUSCHINGER.
KICK.
TETMAJER.
KIRSCH.
KLEBE.

MINUTES OF THE SESSION HELD ON SEPT. 20, 1890.

Chairman:
PROF. BAUSCHINGER, Munich.

Vice-Chairmen:
PROF. BELELUBSKY, St. Petersburg;
PROF. P. DEBRAY, Paris.

Secretaries:
ENGINEER GREIL, Vienna;
ENGINEER OLSCHEWSKY.

Before taking up the regular order of business, Dr. Haack makes the motion that the Conference find ways and means by which to determine the abnormal behavior of cast malleable iron (*flussseisen*, low steel), which is of frequent occurrence and cause of failure, in spite of the fact that test-pieces taken from the ends had given entirely satisfactory results.

This problem is intrusted to the Standing Committee with an amendment by Prof. Martens, to the effect that Governments, directories, etc., etc., be requested to send material for test purposes whenever such cases occur, in order that thorough investigations, including chemical analysis, may be carried out to determine any possible existing relations.

PROBLEM VIII.

Development of processes for testing the relation between the porosity of the surface and the body of bricks, and for determining the permeability of tiles.

It is resolved that:

1. In description of tiles both maximum and minimum measurements are to be stated.

2. The *Sp. Gr.* shall be obtained by using powder which passes a sieve of 900 meshes per sq. c. m. (5,732 meshes per sq. in.) and from which all dust passing a sieve of 31,170 meshes per sq. in. (4,900 per sq. c. m.) has been removed. This determination is to be made by means of the volumenometer.

3. The volumetric weight is to be ascertained by the hydrostatic method, *i. e.*, by measuring the displacement of water of a given volume of material. In such cases where losses may be occasioned by solution or absorption of soluble materials, the volumenometer is to be used for measuring the weight of a given volume after the pieces have been coated with paraffine wax.

4. Test of absorption of water.

5. Determination of soluble salts.

6. The determination of injurious admixtures, as particles of unslaked lime, etc., are to be made in a manner analogous to that used in the case of artificial building stones. (Compare paragraph 4, page 31, also 6 and 7, page 32, of "Resolutions, etc., etc.")

7. The determination of the superficial absorption, as also of the permeability of slabs, is to be as follows:

Pieces of slabs are selected of such size that they can absorb from 20 to 25 c. c. m. (1.22–1.52 cub. in.) of water. These pieces are dried and edges coated with paraffine wax. Now cylindric pipes of 10 c. c. m. contents are sealed on them by the same material. The following observations are made.

to do as little straightening afterward as possible. When this becomes necessary it must be done with greatest care, and, if possible, by means of wooden or copper mallets. When the material is to be tested after annealing, the test-pieces may be straightened hot between roughing and finishing. But, in all events, they must again be annealed before testing.

Form and Preparation of Test-piece.

The Conference reserves determining the shape and dimensions of test-pieces for the present, as it is desirable to reduce the cost of testing such an expensive material as much as possible, and as comparative tests to determine them have not yet been completed. Until such time, however, the standard shapes adopted at former meetings shall be adhered to for tension tests. Inasmuch as the manner of preparing test-pieces of copper may have extraordinary influence on the results of tests, shaping the rough sample must be done with greatest care, and all cuts must be run out without interruption beyond the gauge marks, and the finishing cuts must be very light. The specimens are to be draw-filed and then ground or rubbed down with emery paper. All corners of samples for hot and cold bending tests should be carefully filed round at the bending point.

It is advisable to round the edges of flat test pieces for tension tests to a radius of 1 mm. ($\frac{1}{25}$ inch).

Method of Testing.

Tension test: After passing the elastic limit ("yield point") the rate of straining is to be so regulated as to produce 2% elongation per minute.

Elongation is to be measured as determined by previous resolutions.

In the cold bending test a stud whose diameter is equal to the thickness of the sheet, plate, bar, or wire is to be used, and the thermometer must not be below 50° Fahr. Test-pieces taken from plates, sheets, and bars which can be bent around the stud 180° without breaking, are to be closed down until the inner surfaces touch. Bending tests of wire are to be made as heretofore prescribed for wires.

Hot bending tests shall be made with samples heated to a cherry red, and to be carried until cracks appear or until the complete penetration of the needle, using a box not more than 4 c. m. deep (1.58 inch). For determining constancy of volume, a metal cylinder, slightly tapering toward the bottom, shall be used instead of that of hard rubber.

The details of procedure are adopted as suggested by the reporter, viz.:

"The test for determination of constancy of volume of puzzuolana (trass) mortar is to be made as follows: A mixture of 2 parts, by weight, of puzzuolana, 1 part powdered hydrate of lime, and 1 part of water is put into a vessel of strong sides (galvanized iron), 3–4 c. m. (1.18–1.57 inch) depth and 6–8 c. m. (2.36–3.15 inches) upper diameter, stroked off smooth and immediately submerged in a vessel of water to a depth of 2 c. m. (.78 inch) above the upper edge of vessel. The mixture must not stand above the edge of the vessel, nor must it be hollow. The vessel must have a rigid bottom in order that all expansion occurs in an upward direction."

Foundation: The ring of ebonite on the chamber of the needle apparatus is not sufficiently constant in shape when submerged, and the mortar can expand in this open ring upward as easily as downward. Expansion becomes more readily noticeable when the mortar can expand upwardly only.

Problem XII.

Discovery of rapid methods of determining constancy of volume of Portland cements in air as well as that of other hydraulic bond-materials, and especially examination and valuation of the boiling test as well as the influence of warm baths, is turned over to the Standing Committee with the request that cement manufacturers in general participate in their labors.

The ensuing discussion is summarized by the Chairman by stating that the "Verein Deutscher Cementfabrikanten" (Society of German Cement Manufacturers) agrees to assist in this work, considering the modification of existing methods not impossible, as strong reasons for such changes, based on new methods which have been proved superior, must exist before such changes would be advocated.

Problem XIII.

Examination of the possibility of substituting sieves with perforated plates instead of wire sieves, and determination of the size

PROBLEM XVIII.

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MINUTES OF THE SESSION HELD ON SEPT. 20, 1890.

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PROBLEM XVII.

Examination and valuation of the propositions given on page 46 of the "Report," for the determination of the adhesion of hydraulic bond-materials (mortars).

It is also agreed that the next Conference shall take place in Vienna in September, 1892. The Standing Committee and its chairman are re-elected, with authority to secure further co-operators.

Adjourned at 4.30 P. M.

[Signed.] BAUSCHINGER,
BELELUBSKY,
DEBRAY,
GREIL,
OLSCHEWSKY.

a. The time during which 10 c. c. m. water are absorbed, which are measured into the pipes by means of a pipette.

b. The lapse of time after which dew-like precipitation appears on the lower surface after another 10 to 15 c. c. m. water have been added.

c. The time which elapses until drops may possibly be formed upon the under surface after another 10 c. c. m. water have been added, or else the quantity of water which percolates into a dish below in case of permeability of the tiles.

8. Transverse strength is to be determined by applying two ridges of Portland cement 1 c. m. wide on the under side, 20 c. m. between inner edges. Then centrally between their ends on the opposite side another similar ridge is applied. The pressure is applied on this latter ridge, and bearing is given on the two others.

PROBLEM IX.

Report on the advisable speed of the drums which are used in testing materials for paving and macadam, on the quantities and dimensions of these, and also the determination of their abrasion or wear as determined by practical tests, is again turned over to the standing committee, not having been completed.

PROBLEM X.

More definite determination of the method and manner of determining the volumetric weight of a hard material or sand when (a) *sifted*, or (b) *shaken into a cylindrical standard liter measure* 10 *c. m. deep.*

It is resolved that three methods are to be used in determining the volumetric weight:

Sifting is to be done by the mechanism designed by Tetmajer.

Shaking-in is to be done by the mechanism designed by Tetmajer.

Filling by hand by using the filling funnel and the standard liter measure.

PROBLEM XI.

To test and appraise the propositions for determining setting, and examination of constancy of volume, of puzzuolana (trass) *as given on pages* 40 *and* 42 *of* "*Resolutions, etc., etc.,*" is solved by adhering, in general, to the propositions previously made. For determining time of setting, such weight is to be used which causes to these tests, elongation diminishes with thickness, as was to be expected, and that it amounted to 24% to 25% in plates of 10mm. thickness (0.394 inch), and decreased to 13% to 14% in plates of 5mm. thickness (0.190 inch).

These results agree generally with the specifications of the French Ministry of the Navy.

Concomitant with that problem the above-named engineers also approached the problem of comparison of standard shapes of test-pieces as proposed by the Conference at Dresden, in order to assist in its solution by using the investigations of Barba, Bennet, and others, as well as their own made in our laboratory.

Bearing on the relation existing between elongation and dimensions of test-pieces Barba's law * alone is thus far known, and is as follows: "Test-pieces having sections geometrically similar show actual elongations proportional to their dimensions, and equal when stated in per cent." This law is applied in France, where the reporter, on his visit to the shops of the Paris-Lyons-Medit. R. R., during the summer of 1889 (at the time of the Paris Exposition and the conventions called in connection therewith), saw the preparation of test-pieces which were geometrically similar for the purpose of inspection of iron.

Although this method makes it possible to prescribe a certain fixed per cent. of elongation for all shapes of material, still, it has several great objections; it is very difficult and inconvenient to adhere to the geometrical similarity of test-pieces in a large order of material. For practical purposes it is easiest to maintain a fixed length of test-piece, as is usually done, with a uniform width in the case of flat pieces, prescribing for rounds a fixed series of diameters, as has been done in the "Resolutions of the Conferences of Munich and Dresden."

Barba's law undoubtedly points to a further development, particularly a greater generalization, and the inquiry:

1. From a scientific point of view, what relations exist between the elongations of two test-pieces whose three dimensions—length, breadth, and thickness—vary irrespectively; and

2. From a practical point of view, what relation exists between the elongations of two test-pieces of equal length and different cross-section dimensions.

* Études sur les allongements des métaux après ruptures, par Barba (Mémoires de la Société ingenieurs civils, 1880).

of holes which correspond to the meshing of the sand sieves as given on page 43 *of the "Report,"* is discussed by the Conference, and it is agreed that perforated plates are decidedly preferable to wire sieves. The adoption of perforated sieves is resolved upon, and the exact determination of the sizes and spacing of holes and thickness of metal to produce a sand which assures a resistance equal to that obtained by the use of the standard sand heretofore produced by wire sieves is to be the work of the Standing Committee.

PROBLEM XIV.

Discovery of a standard sand which is not only uniform in grain but also in weight, as well as other properties which must be considered, is solved by the Conference agreeing that the sand found at Freienwalde be used as the standard sand—in a restricted sense —to which all tests are to be referred. This sand shall have passed through perforated sieves of such quality that it be between those obtained by using wire sieves of 60–120 meshes and sieves of 64 and 124 meshes per sq. c. m.

It is left to countries other than Prussia to obtain a standard sand, and, if possible, of such character that it be similar in its effects upon resistance to that found at Freienwalde. Should this be impossible, then the standing committee is to gain experience as to proper coefficients for comparison.

Problems XV., XVI., and XVII. are again left to the Standing Committee for further examination. They are:

PROBLEM XV.

Determination of the consistency of a standard mortar and discovery of a practicable mechanical method of producing briquettes, especially of the conditions by which similar density is obtained for tension and compression briquettes.

PROBLEM XVI.

Examination and valuation of tests of resistance of pure Portland cement (neat), prepared in standard consistency and on a non-absorbent surface, as well as those with admixture of standard sand after 3 *days' setting; development of propositions of a more rapid method of testing other hydraulic bond-materials.*

specimen. The total per cent. of elongation consists of two corresponding parts; it is smaller as the specimen increases in original length with the same cross-section, or as the cross-section diminishes with similar lengths.

There is no difficulty in separating the local from the proportional elongation by measuring the elongations of each centimeter of original length over the entire specimen, and plotting them.

Let us consider two test pieces, of which l and λ are length, e and ε total, e_1 and ε_1 proportional or uniform, and therefore, $e_{11} = e - e_1$ and $\varepsilon_{11} = \varepsilon - \varepsilon_1$ represent the local absolute elongations after rupture, and $e\%$, $\varepsilon\%$, $e_1\%$, $\varepsilon_1\%$, $e_{11}\%$ and $\varepsilon_{11}\%$ are the corresponding relative or per cent. of elongation. Let us assume that the first specimen has been tested, and the values e, e_1, and e_{11} have been determined, then the problem is the determination of the values of ε, ε_1 and ε_{11} of the second specimen, as functions of e, e_1, and e_{11}.

First Case: Test pieces have different lengths l and λ, but equal cross section $\bar{\omega}$.

It is clear that the absolute total elongation of the second specimen will be

$$\varepsilon = e_1 \frac{\lambda}{l} + e_{11} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1a)$$

and elongation

$$\% = \varepsilon\% = e \frac{100}{\lambda} = e_1 \frac{100}{l} + e_{11} \frac{100\, l}{l\ \lambda} = e_1\% + e_{11}\% \frac{l}{\lambda} \quad . \quad . \quad . \quad . \quad (1b)$$

Plotting the lengths as abscissæ on Fig. 1, and the total absolute on per cent. of elongations as ordinates, we obtain in the first case a straight line cutting the axis of Y at a distance of e_{11} from the origin; in the second case a hyperbola, one of whose asymptotes becomes the axis of Y, while the other is a line parallel to the axis of X, and at a distance equal to $e_1\%$.

Tests made by Barba of specimens, all cut from one bar of low steel (*flusseisen*), having uniformly a diameter of 17.2 mm. (.681 inch) and 50, 100, and 500 mm. long (1.97 inch, 3.94 and 19.70 inches), gave total actual elongations ε of 21.0 – 124.1 mm. (.827 – 4.889 inch) and total per cent. of elongation of 42.0 – 24.8%. The contraction of area was 68.3%; tenacity of 52529 lbs. per square inch; limit of elasticity (yield point?), 33,700 lbs. per square inch. These results substituted in formula $1a$ and $1b$, and plotted at $a\ b$ and $a_1\ b_1$ in Figs. 1 and 2. By pro-

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APPENDIX IV.

LECTURE BY PROFESSOR N. BELELUBSKY OF ST. PETERSBURG; CONTRIBUTION TO THE SOLUTION OF THE PROBLEM: "COMPARISON OF STANDARD SHAPES OF TENSION TEST-PIECES."

TRANSLATED BY GUS. C. HENNING, M. E., REPORTER FOR THE COMMITTEE ON STANDARD TESTS AND METHODS OF TESTING MATERIALS, OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

IN establishing technical conditions (specifications) for the delivery of low steels (*flusseisen*) for construction of boilers and ships, the Commission appointed by the Minister of Roads met with considerable difficulty in determining the desired elongation for plates of different thicknesses, for, as is well known, test-pieces of different thickness but otherwise similar dimensions (length and breadth) give different elongations, and the investigations heretofore made had not yet determined the laws governing these variations. For this reason, two Russian engineers, Brandt and Liachnitzki, both members of the above Commission and assistants at the Institution for the Construction of Roads (Department of Road Construction), made a series of tests of low steels (*flusseisen*) in the mechanical laboratory of this institution in 1887 and 1888, for which the material was furnished by three steel-works; the test-pieces used were obtained partly from plates of different thicknesses and partly by planing down strips cut from one plate to obtain the desired shape. In accordance with previous resolutions of the Conferences, all test-pieces had the same width and length, as is usually prescribed in specifications; and the problem of the investigation was to discover the relation between elongation and thickness of the test pieces.

Without going further into detail I wish to say that, according

* This Appendix to the Report of a Committee on Standard Tests and Methods of Testing Materials appointed by the American Society of Mechanical Engineers attaches itself to the Report of the committee, No. 380 of the Transactions and published at page 604 of Volume XI. It was presented for record at the New York Meeting 1891.

Tenacity was 68250—72500 lbs per sq. in. and reduction of 4—50%.

The third set of curves ef and e_1f_1 in Figs. 1 and 2 represent the test made by the Naval Advisory Board (see Rudeloff, *Festigkeits-untersuchungen von flusseisen in Zeitsch. d. Ver. deutsch. Ingen.* 1887) from test-pieces of oblong section of 25 mm. width (1 inch) and about 16.5 mm. thickness (.650 inch) were obtained with lengths of

38.1 (1 50 in.), 127.0 (5.0 in.), and 254.0 mm. (10.0 in.).
ε = 18.7 (.736 in.), 41.9 (1.65 in.), and 69.6 mm. (2.74 in.).
$\varepsilon\%$ = 49.0 33.0, and 27.4%,

with a reduction of 57% to 63%.

The curves obtained from Barba's tests are the most beautiful, being undoubtedly due to the homogeneity of the material, and probably also to the shape and uniformity of cross-section.

Formulæ (1a) and (1b) also enable us to determine the relation between the elongations of different parts of the same test-piece; it must, however, be understood that in all cases the fracture (neck, part having local elongation) be included between the ends of such part.

Second Case: Test-pieces have a circular section (simplest) and equal lengths, but different diameters (and all are, of course, cut from one piece of metal).

Let the specimens be described as follows:

l = length of both test-pieces,

d and δ = diameters, and

e, ε, $e_1\%$, $\varepsilon_1\%$, $e_{11}\%$, ε_{11} the same quantities as heretofore, and assuming that values for e, e_1, and e_{11} are obtained by direct measurements of first test-piece, then we shall obtain the following formulæ for the other specimen:

$$\varepsilon = e_1 + e_{11}\frac{\delta}{d} \quad \ldots\ldots\ldots \quad (2a)$$

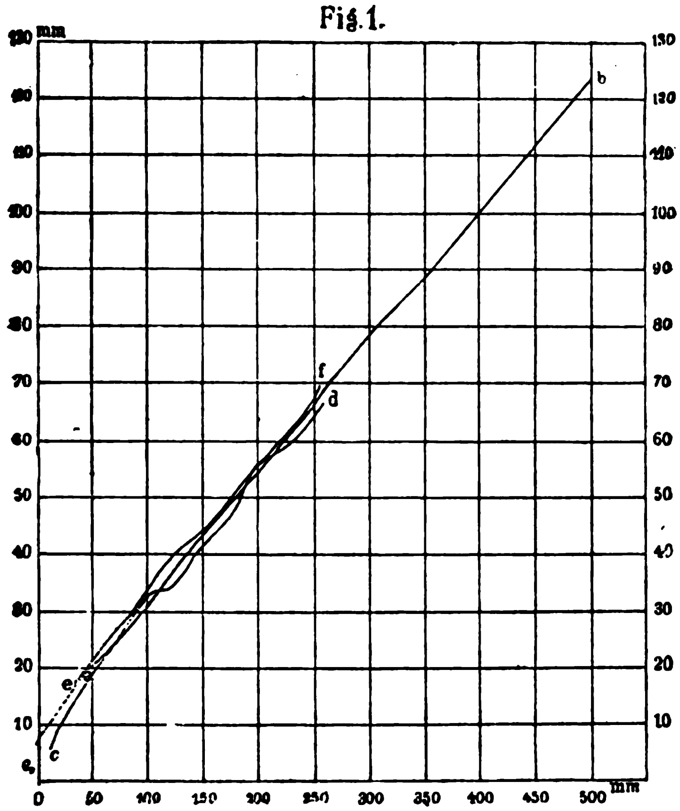
$$\varepsilon\% = e_1\% + e_{11}\%\frac{\delta}{d} \quad \ldots\ldots \quad (2b)$$

Diagrams for test-pieces of equal lengths and different diameters, in which values of δ have been laid down as abscissæ and ε or $\varepsilon\%$ as ordinates, are straight lines, which intersect the axis of Y at points from the origin at distances equal to e_1 and $e_1\%$.

Only after this problem has been solved will inspection be established on a scientific basis.

The formulæ deduced by the above-named Russian engineers and given below, present a generalization of Barba's law to some extent, and permit the determination of the relation of elongation and dimensions of test-pieces in some special cases. I pro-

Fig. 1.



pose herewith to communicate these formulæ in a concise manner, with a few apposite explanations and examples.

Actual total elongation, as is well known, consists of two parts; firstly of that uniform elongation, which is proportional to the length of specimen; and secondly, of that which is independent of the length, or so-called local elongation, which is dependent upon the contraction of area at the location of rupture. The larger the diameter of a test-piece, the greater is the local, and consequently also the total, elongation of a given length of

c. Formula (2), for equal lengths and varying cross-sections of similar shapes, still requires to be proven by a greater series of tests before it can be regarded as fully established.

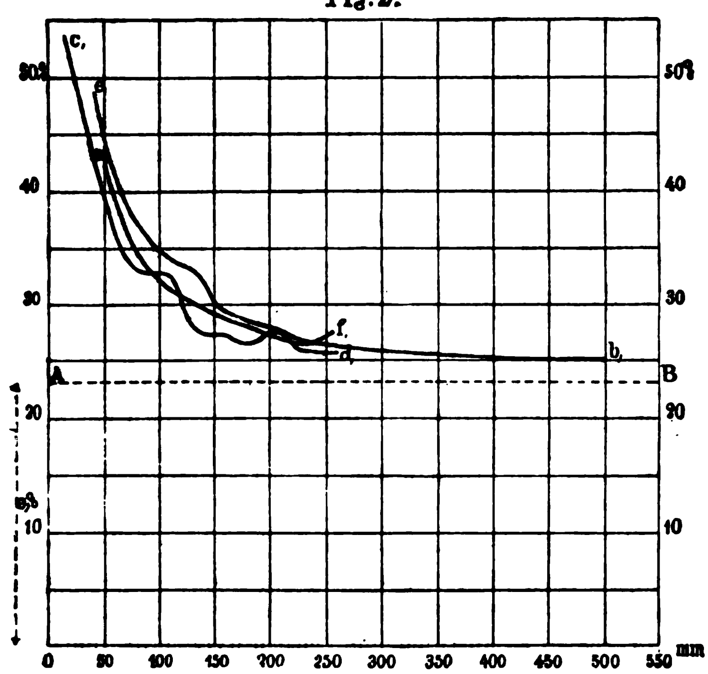
d. The logical variation of elongation of test-pieces of equal length (say the standard 8-inch length), while one or other of the cross-dimensions varies without preserving similarity of cross-section, has not as yet been investigated.

Engineers know very well that this last case is the one which causes so much trouble in inspecting materials, because it is the most convenient and appropriate to make test pieces, when they are to be strips, of a uniform length and breadth (standard) and of a thickness equal to that of the material (as in plates, etc., etc.). In investigating this problem of our conferences (stated in the title of this paper) this case particularly should be kept in view, and it would be highly desirable to test these formulæ for different particular cases when testing them. I have already stated that it is frequently customary in France to inspect material, basing the procedure on Barba's law In spite of this it is nevertheless felt to be highly desirable, as explained in the references of Engineer Cornut read by him at the "Mechanical Congress" held in Paris in 1889, to standardize methods of testing, especially in testing metals. This, in connection with the papers read at the "Building Congress," at which our guests from France now present assisted, has led to the resolution that an international standardization of methods of testing is absolutely necessary.

longation of the line $a\ b$ to the axis of Y we obtain for all the test pieces the local elongation of $e_{11}\% = 9$ mm. (.3546 inch); for either of the above 10 specimens $\varepsilon_{11}\% = 9\frac{100}{x}$, and therefore the uniform elongation for all test pieces $\varepsilon_{1}\% = \varepsilon\%$ (total elongation taken from the tables) less $\varepsilon_{11}\%$, calculated. This value $\varepsilon_{1}\%$ defines the asymptote $A\ B$ in Fig 2. (A may be considered the center of the hyperbola.)

Curves $c\ d$ and $c_1\ d_1$, in Figs. 1 and 2, represent the results of

Fig. 2.



tests made by Bennett (Description of Tensile Tests of Iron and Steel Bars, *Proc. Inst. Mech. Engrs.*, 1886) on specimens of different lengths and almost uniform oblong section [about 47.7 mm. (1.879 inch) wide and 12.4 mm. (.4985 inch) thick]. These results gave—for lengths of

12.7 (0.500 in.),	127.0 (5.00 ins.),	254.0 mm. (10.0 in.)
	(Eleven different lengths.)	
$\varepsilon = 6.9$ (2718 in.),	35.1 (1.343 in.),	66 1 mm. (2.604 ins.)
$\varepsilon\%$ 54.0	27.6	26.0%.

CCCCLXXXI.

PROCEEDINGS

OF THE

SAN FRANCISCO MEETING

(XXVth)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS,

May 16th to 19th, 1892.

LOCAL COMMITTEE OF ARRANGEMENT: Wm. R. Eckart, *Chairman;* Chas. G. Yale, *Secretary;* Geo. W. Dickie, James Spiers, John Richards, Marsden Manson, E. J. Molera, H. J. Small, Frank Van Vleck.

FIRST DAY. MAY 16TH.

THE San Francisco Convention of the American Society of Mechanical Engineers was held at the close of a transcontinental excursion of its members, beginning May 4th, and consuming twelve days. The details of this excursion, while of interest to those enjoying it, form no essential part of the convention itself, but will be recorded in a special appendix. The features of the meeting begin, therefore, with the opening reception and address of welcome given in the reception parlor of the Palace Hotel of San Francisco on the afternoon of Monday, May 16th.

The members were convened as soon as possible after arrival, and were welcomed to the Pacific Coast by His Honor Mayor George H. Sanderson, of San Francisco, in a short and fitting address. In the absence of President Charles H. Loring, of the Society, Ex-President Robert W. Hunt replied in graceful terms and the meeting was formally opened.

The Committee of Arrangements and Entertainment had provided that on this afternoon visits should be made to the power houses of the different cable railways which intersect the city, and which are a feature of notable engineering interest. These visits were under the auspices of Mr. A. S. Hallidie, recognized as the originator and pioneer of the systems in the city. The visitors were divided into groups, each with a special guide, and were thus distributed among the various plants. The cable railways had also made the Society their guests, and wearers of the Society's badge were carried free on their cars during the convention. There is considerable variety in detail at the various plants, some using cross compound engines and others triple expansion: the take-up devices differed also, but all used rope-driving, and some carried their water of condensation to open shallow tanks on the roof, so as to use it over and over again. Most of the engines were fitted with the O'Neill valve gear with automatic cut-off of admission and variable compression, using four valves of the poppet type.

The first session for business was held in the hall of the Academy of Sciences, 819 Market Street, on Monday evening. Ex-President R. W. Hunt called the meeting to order, and introduced Dr. H. W. Harkness, President of the Academy, who in a short address welcomed the Society to the Academy Building and to the other courtesies of their organization. Mr. John Richards also, as President of the Technical Society of the Pacific Coast, in a few words extended similar courtesies from the Technical Society, including the use of their library.

The Secretary then read the Report of the Tellers of Election.

The Council would 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 for the XXVth meeting of the Society in May, 1892.

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. 1 and 2, respectively pink and yellow.

There were 515 votes cast in the ballot upon the pink list, of which 4 were thrown out because of informalities.

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

There were 544 votes cast in the ballot upon the yellow list, of which 19 were thrown out because of informalities.

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

April 30th, 1892.

The lists are appended below :

MEMBERS.

Aldrich, Wm. Sleeper	Baltimore, Md.
Barth, Ernest Charles	Cleveland, O.
Bonzano, Adolphus	Phœnixville, Pa.
Bowen, Ernest S.	Auburn, N. Y.
Bryant, George Holmes	Palo Alto, Cal.
Cahoon, James Blake	Lynn, Mass.
Conklin, M. T.	Detroit, Mich.
Conover, Edwin K.	New York City.
Detrick, Jacob S.	Baltimore, Md.
Dickie, Geo. W.	San Francisco, Cal.
Flather, Fred'k A.	Newton Upper Falls, Mass.
Gibson, Arthur	Chicago, Ill.
Glasgow, Arthur Graham	New York City.
Golden, Michael Joseph	Lafayette, Ind.
Goldthwait, Abel G.	Troy, N. Y.
Grant, Geo. B.	Lexington, Mass.
Hammond, Chas. Adriance	Boston, Mass.
Hildreth, Russell W.	New York City.
Hill, Ebenezer	South Norwalk, Conn.
Hill, George	New York City.
Kendall, James Henry	Cambridgeport, Mass.
King, Frank B.	Sparrows Point, Md.
Kingsbury, Albert	Hanover, N. H.
Lawrence, J. W.	Fort Collins, Col.
Leighton, Edward I.	Cleveland, O.
Lewis, Justus P.	Hartford, Conn.
Lindsay, W. H.	Philadelphia, Pa.
Longenecker, Chas. K.	Painted Post, N. Y.
Loss, Henrik V.	Pencoyd, Pa.
Matton, Fred'k Victor	Camden, N. J.
Mesick, D. W.	Syracuse, N. Y.
Moore, R. S.	San Francisco, Cal.
Nicolson, John T.	Montreal, Canada.

Pollock, Alex............ ....New York City.
Power, Fred'k Macy..........................Syracuse, N. Y.
Rickson, Chas. Eric..........................Brooklyn, N. Y.
Sattler, Wm. R..............................Elizabeth, N. J.
Seaver, John Wright........................ Allegheny, Pa.
Seymour, Louis I............................South Africa.
Siebert, Alfred.............................New York City.
Simonds, Daniel.............................Fitchburg, Mass.
Sloat, George F.............................New York City.
Spencer, E. J...............................Lynn, Mass.
Strobel, Victor O...........................Philadelphia, Pa.
Stevens, William W..................... ....Philadelphia, Pa.
Varney, William Wesley......................Baltimore, Md.
Weeks, Charles H............................Cincinnati, O.
Wiechardt, August Julius ....Las Cruces, N. M.
Wood, Fred'k W........... ....Los Angeles, Cal.
Wyman, Horace...............................Worcester, Mass.
Yaryan, Homer T.............................Toledo, O.

PROMOTIONS TO FULL MEMBE SHIP.

Johnson, Arthur E...........................Stamford, Conn.
Lowe, Wm. Vose..............................Fitchburg, Mass.
Marx, Henry ....Cincinnati, O.
Mumford, Edwin Huidekoper...................Brooklyn, N. Y.
Rankin, Thomas L............................New York City.
Shaw, Edwin Coupland........................Buffalo, N. Y.
Smith, Albert William.......................Madison, Wis.

ASSOCIATES.

Boenig, Robert W............................Brooklyn, N. Y.
Cassier, Louis..............................New York City.
Dingey, P. S................................Chicago, Ill.
Owens, Robert Bonie........................ Lincoln, Neb.
Pitman, Stephen Minot.......................Philadelphia, Pa.
Rankin, Thomas L............................New York City.
Scott, Fred'k Spurgeon......................New York City.
Veeder, J. Irwin............................Chicago, Ill.
Wedge, Utley................................Cleveland, O.

JUNIORS.

Almirall, Juan Antonio......................Ithaca, N. Y.
Ashley, George T............................Indianapolis, Ind.
Bedell, Frederick...........................Montclair, N. J.
Blanchard, Winslow..........................Boston, Mass.
Dearborn, Wm. Langdon.......................Chicago, Ill.
Funk, William Francis.......................La Crosse, Wis.
Hale, Henry Warren Kilburn..................Philadelphia, Pa.

CCCCLXXXI.

PROCEEDINGS

OF THE

SAN FRANCISCO MEETING

(XXVth)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS,

May 16th to 19th, 1892.

LOCAL COMMITTEE OF ARRANGEMENT: Wm. R. Eckart, *Chairman;* Chas. G. Yale, *Secretary;* Geo. W. Dickie, James Spiers, John Richards, Marsden Manson, E. J. Molera, H. J. Small, Frank Van Vleck.

FIRST DAY. MAY 16TH.

THE San Francisco Convention of the American Society of Mechanical Engineers was held at the close of a transcontinental excursion of its members, beginning May 4th, and consuming twelve days. The details of this excursion, while of interest to those enjoying it, form no essential part of the convention itself, but will be recorded in a special appendix. The features of the meeting begin, therefore, with the opening reception and address of welcome given in the reception parlor of the Palace Hotel of San Francisco on the afternoon of Monday, May 16th.

The members were convened as soon as possible after arrival, and were welcomed to the Pacific Coast by His Honor Mayor George H. Sanderson, of San Francisco, in a short and fitting address. In the absence of President Charles H. Loring, of the Society, Ex-President Robert W. Hunt replied in graceful terms and the meeting was formally opened.

The Committee of Arrangements and Entertainment had provided that on this afternoon visits should be made to the power

Laforge, Fred H	Waterbury, Conn.
Lewis, Wilfred	Philadelphia, Pa.
Mansfield, A. K	Salem, O.
Martens, F	College Point, N. Y.
McBride, James	Brooklyn, N. Y.
Miller, Fred J.	New York City.
Monaghan, Wm. F	New York City.
Moore, D. G	Elizabeth, N. J.
Moore, R. S	San Francisco, Cal.
Parks, E. A	Providence, R. I.
Power, F. M	Syracuse, N. Y.
Richards, John	San Francisco, Cal.
Rinman, Gustav O	Cincinnati, O.
Rites, F. M	Pittsburgh, Pa.
Scheffler, F. A	Cleveland, O.
Schoenborn, W. E	Washington, D. C.
Sharp, Joel	Salem, O.
Small, H. T	Sacramento, Cal.
Smith, Geo. H	Providence, R. I.
Spiers, James	San Francisco, Cal.
Stahl, Albert W	San Francisco, Cal.
Stearns, Albert	Brooklyn, N. Y.
Torrey, Herbert G	New York City.
Trump, Charles N	Wilmington, Del.
Van Vleck, Frank	Los Angeles, Cal.
Wellman, S. T	Thurlow, Pa.
Wiley, Wm. H. (*Treasurer*)	New York City.
Williamson, W. C	Philadelphia, Pa.
Wood, Fred W	Los Angeles, Cal.

As guests were also registered :

Andrews, Edward	Wilmington, Del.
Cotton, Walter G	Boston, Mass.
Roux, Ph	Paris, France.
Smith, Gilbert S	Philadelphia, Pa.
Wellman, W. S	Thurlow, Pa.

And thirty-three ladies.

The professional papers were then taken up, and the following were read and discussed : By Mr. Jno. Richards, "Notes on a Problem in Water Power"; by Mr. Jno. H. Cooper, on "A Self-lubricating Fibre-graphite for the Bearings of Machinery." This was presented by Mr. Charles N. Trump in the author's absence, and was discussed by Messrs. Dickie and Spiers, of the Society, and by Messrs. Harkness, Keith, and Cummings, of San Francisco, by invitation. The paper by Mr. Harris Tabor, entitled "Machine Moulding," was discussed by

Messrs. Higgins, Moore, and Richards; that by Mr. Chas. H. Manning, on "A Novel Fly-wheel," was discussed by Messrs. Mansfield, McBride, Borden, Dickie, Laforge, and Lewis. The last paper of the evening was by Mr. W. W. Christie, entitled "An Experiment with Aluminum." The session then adjourned.

SECOND DAY. MAY 17TH.

The peculiarities of the climate at San Francisco made it seem advisable to arrange for the visits and similar entertainment by our hosts to be assigned for the mornings, instead of the afternoons, as is the more usual custom. They were also made to begin very early, for the same reasons.

On this day the hosts had arranged for a trip by water. The very large and commodious tug *Fearless* had been put at the service of the committee by her owners, the Messrs. Spreckels, of San Francisco, and bore the party up to the entrance of the Golden Gate to the harbor. Turning there before the heavy swells became too strong, the party was taken through Raccoon Straits, and past the notable islands and points up to Mare Island for a visit to the navy yard. The commandant, Admiral John Irwin, and Chief Engineer Jno. W. Moore, member of the Society, with other officers, met the party and escorted them through the shops and quarters, and also entertained visitors on board the U. S. cruiser *Boston*. The visitors viewed with interest the old flag-ship *Hartford* and other historic craft moored in the water-way. After the departure on the return, luncheon was served and the party returned home.

In the evening was held the second session for papers, at which Vice-President G. I. Alden presided, and there were presented those by Mr. A. F. Nagle, on the "Density of Water at Different Temperatures," discussed by Messrs. Molera and Jacobus; by Mr. C. H. Peabody, entitled "Economy and Efficiency of the Steam Engine"; by W. O. Webber, "On Some Tests of a Portable Boiler"; by A. W. Stahl, "On the Utilization of the Power of Ocean Waves"; by B. J. Dashiell, on "The Electric Railway as Applied to Steam Roads"; by Mr. W. F. M. Goss, on "An Experimental Locomotive," and by Mr. F. M. Rites, on "The Steam Distribution in a Form of Compound Engine." Mr. Webber's paper was discussed by Messrs. Jacobus and Scheffler; Mr. Stahl's, by Messrs. Stodder, Dickie, Jacobus, Molera, and Steen. The session then adjourned.

THIRD DAY. MAY 18TH.

The morning of Wednesday was devoted to an excursion tendered to the visiting members and their hosts by the Spring Valley Water Co., to their newest dam, coupled with a beautiful drive through the valleys which form part of their drainage area. A special train on the Southern Pacific Coast Railway conveyed the party to San Mateo, where carriages were waiting. The dam is a massive structure of concrete blocks, of great size, and is planned for a depth of 115 feet of water. After the inspection of the dam, a luncheon was served, spread under a great bay-tree, large enough to shelter the party of over 100, who were present. On the return, a stop was made at the pumping station near San Andreas.

The third session for papers was convened in the hall of the Academy of Sciences in the evening, Vice-President G. I. Alden in the chair. The papers were by Mr. W. S. Aldrich, on "Compounding Centrifugal and Load Governing by a Rotary Piston Valve"; on the "Measurement of Power," by Mr. Thomas Gray; on "Autographic Recording Apparatus for Use in the Testing of Materials," by Mr. Thomas Gray, and discussed by Mr. G. C. Henning; on the "Elastic Curve and Treatment of Steel," by Mr. G. C. Henning, with discussion by Mr. D. S. Jacobus; on "Results of Principal Experimental Measurements of Performance of Refrigerating Machines," by Messrs. J. E. Denton and D. S. Jacobus; and on "Two-cylinder *versus* Multi-cylinder Engines," by Messrs. Green and Rockwood. This latter was discussed by Messrs. Cooper, Jacobus, Stut, Alden, and Gale.

At the close of the papers, the Secretary presented and read the preliminary report of the Society's committee appointed to propose a standard method for conducting locomotive tests. The chairman, in transmitting the report, sent with it a letter making certain recommendations, which when reduced into the form of resolutions were referred by the meeting to the Council, with power.

Resolved, That it be referred to the Council to consider the advisability of conducting at the Columbian Exposition a series of locomotive tests, under the supervision of an advisory board selected from the American Society of Mechanical Engineers and the American Association of Railway Master Mechanics.

Resolved, That it be referred to the Council to consider and invite joint action by this Society and the American Association of Railway Master Mechanics, in view of the fact that this latter association has appointed a committee with the same objects in view as those before the committee of this Society.

Resolved, That it be recommended to the Council that the names of Messrs. W. F. M. Goss, D. L. Barnes, and Geo. H. Barrus be added to the committee.

These resolutions having been passed, the report of the committee was read, and discussion upon it by Arthur T. Woods. The report will be printed as a paper of the Society at a later date.

The Report of Progress from the Society's Committee on Standard Sizes for Flanges was then presented, as follows :

REPORT OF THE COMMITTEE ON FLANGE STANDARDIZATION.

Your Committee begs respectfully to report that since the last meeting of the Society they have prepared a series of blanks for general distribution among manufacturers and users of flanges, with a view to ascertaining what dimensions are now in use, and as a guidance for their conclusion as to what standard, if any, could be recommended to the Society for consideration.

Copies of the blanks are herewith appended, and as the circular-sheet which accompanies them is to a certain extent self-explanatory, we do not here enlarge upon the advantages which might obtain by the universal adoption of any well-chosen standard.

Copies of the circulars were sent to three hundred and ten manufacturers early last month, which it is thought completely covers the entire list of makers—not only of valves of all descriptions, but also steam-pumps and engines, cast-iron water pipe, and many minor special goods, such as water meters, separators, etc.

The return has been disappointingly slow and incomplete, and as correspondence with the delinquents will be advisable, we would respectfully ask for a postponement before making a full recommendation.

Your Committee would further deem this advisable in order to have a conference with a committee which has been appointed for a purpose similar to our own by the Master Steam Fitters' Association, as their judgment about actual necessities would be probably valuable.

The meeting of the association is to be held in Boston for several days, beginning the 24th inst., and it is thought that immediately after this time we shall be ready to make a complete report.

The almost universal opinion of those who have thus far made returns is that a number of bolts which is a multiple of four would be most desirable, for the reasons explained in the circular letter. If it meets the approval of the members in session in San Francisco, and there is time available for the purpose, we would suggest that a brief discussion on this question by the members, followed by a stenographer's copy of their views to your Committee, would be of assistance.

For the reasons above given we therefore respectfully ask that the Committee be continued, to give opportunity for a more complete report.

Respectfully submitted,

CARLETON W. NASON, *Chairman.*

JNO. E. SWEET,
FRANK H. BALL,
A. J. CALDWELL,
ALEX. H. JARECKI,
} *Committee.*

NEW YORK CITY, *May* 12, 1892.

APPENDIX.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.
12 West Thirty-first Street, New York City.

SUBJECT: FLANGE STANDARDIZATION.

At the autumn meeting of the Society, held in Philadelphia, 1887, a paper was read by Mr. Percy Sanguinetti on the importance of the manufacturers of this country adopting some form of flange standard, which would make them more or less interchangeable, independent of the various services to which they were to be applied, or, within reasonable limits, the pressures which flanged joints were to resist.

The subject was deemed of so great importance that a committee was immediately appointed by the President of this Society, and much good work in the way of compilation of data has already been effected; but owing to changes in business arrangements, the members of the Committee were so scattered that it was impracticable to bring their work to a termination, and they, therefore, requested to be relieved.

At the spring meeting of this Society a new committee was appointed by President Hunt, consisting of the undersigned, and they would, therefore, respectfully call your attention to the enclosed circulars, bearing upon the work in hand.

It is their intention, before submitting to the Society any standard for its consideration, to obtain from all manufacturers using flanged pipe connections for any purpose whatever, as complete data as possible of their own standards in actual use. With this in hand it is quite possible that so many may be nearly alike as to at once suggest that perhaps a mean of the figures received would be an advisable standard to recommend, as it would obviously necessitate the least amount of alterations on patterns, jigs, or templates upon all manufacturers.

After receiving full replies to the enclosed circulars, it is the intention of the Committee, before making a report to the Society, to send out a standard or standards, which in their opinion would be advisable, for your criticism, in order that a careful conclusion shall be reached before making their final report, and your advice will materially aid them.

That such a standard, if universally adopted, would be a boon is clearly obvious, as by its use a pump or engine could be purchased from one manufacturer, a flange union, fitting, or valve from another, and a gasket—which would be, of course, then kept in stock—from a third, after which the fitter could go to his work with certainty that the whole would go together, and the time which is now lost in making templates from each piece, and drilling blank flanges in a shop, would be entirely saved.

Sheet "A" is intended for manufacturers of general steam fitters' supplies, involving the use of both cast and wrought-iron pipes, flanged valves both of brass and iron, flange unions, and goods of this character.

Sheet "B" is intended for manufacturers of steam-engines and steam-pumps. It will be noticed that flanges for both steam supply and exhaust are specified, and it is hoped that this sheet will be filled by you as completely as possible in both groups.

Sheet "C" is intended for manufacturers of cast-iron pipe, both for water and gas, and for manufacturers of large gate valves.

Sheet "D" is intended for the purpose of eliciting a fuller explanation of your experience bearing on the different flanges in question, and also to get from you any serious objection which occurs to you to the adoption of a general standard, which might be grouped under two, three, or four heads, if a single one is insufficient.

From steam fitters especially, suggestions have already been received as to the importance of using in all cases such a number of bolts as will enable them to turn a right angle—this being the one most commonly used, and this can evidently be accomplished only by using such a number as is divisible by four. A preliminary examination shows that the most abrupt jump would come between the three-inch and four-inch patterns. The former size would have apparently four bolts, and the latter eight. This would make the distances between centres approximately four inches, and two and three-quarter inches. While it is true that this is a considerable difference, the additional stress which the four bolts would be called upon to sustain could be readily provided for by increasing their diameter, and further adding somewhat to the thickness of the flanges to prevent their springing.

Although the adoption of such a standard is open to serious criticism, it is possible that, if it could be adopted, the advantages gained could more than compensate for the difficulties to be met, and the Committee would, therefore, solicit a full expression of your views on this feature.

The Committee, although they admit that the work imposed upon you is considerable in filling out the enclosed sheets, hope you will agree with them that if a general standard can be found and adopted, the benefit gained and the amount saved to the country at large will be great, and that you will aid them to the fullest possible extent.

Replies may be sent to any of the Committee, but preferably to the office of its Chairman, at No. 71 Beekman Street, New York City. Respectfully yours,

Committee on Flange Standards:
JOHN E. SWEET,
FRANK H. BALL,
ANDREW J. CALDWELL,
A. H. JARECKI.

CARLETON W. NASON, *Chairman.*

P. S.—In order that a report may be made to the Society at its May meeting for action, it is respectfully urged that your reply, with figures inserted in the several blanks, be returned at as early a date as possible.

[Sheet A, No.—.]

Will you kindly fill in this blank and return to Chairman of Committee on Standard Flanges, Carleton W. Nason, 71 Beekman Street, New York City?

*Name of Firm*____________________

*Address*____________________

____

*Date*____________________ ____________________

NOTE.—When several words are used describing material, etc., please cross out those to which this sheet does not refer.

Dimensions of Standard { Cast-iron, Brass, Wrought-iron, Steel } Flanges for { Steam and Water and Gas. }

Pressures ranging from_____lbs. per sq. in., to_____lbs. per sq. in.

	Diameter of Flanges.	Thickness of Flanges.	Number of Bolts.	Size of Bolts.	Diameter of Bolt Circles.	Arrangement of Bolts: if on upper vertical centre *xy*, enter (|); if not, enter o|o.	Thickness of Pipe.	Diameter of Boss or Reenforcement.	Depth of Boss or Reenforcement.	Whether Case I. or Case II.
Size.	A	B	C	D	E	F	G	H	I	J
[illegible]										
[illegible]										
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1										
1¼										
1½										
2										
2½										
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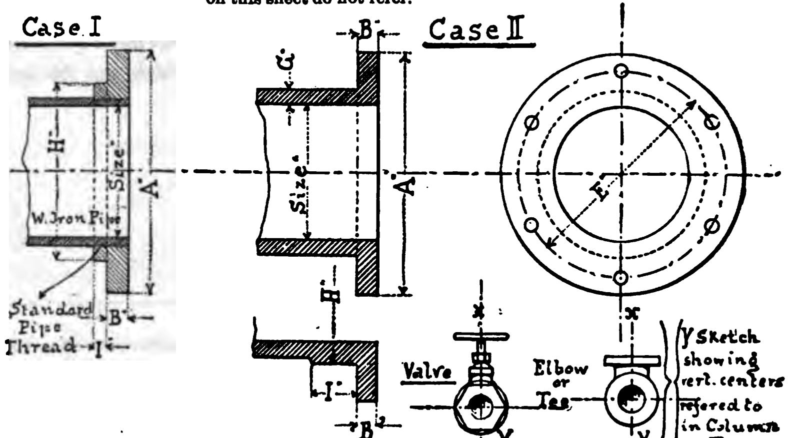
3½										
4										
4½										
5										
6										

	Diameter of Flanges.	Thickness of Flanges.	Number of Bolts.	Size of Bolts.	Diameter of Bolt Circles.	Arrangement of Bolts: if on upper vertical centre *xy*, enter (|); if not, enter o|o.	Thickness of Pipe.	Diameter of Boss or Reenforcement.	Depth of Boss or Reenforcement.	Whether Case I. or Case II.
Size.	A	B	C	D	E	F	G	H	I	J
7										
8										
9										
10										
12										
14										
16										
18										
20										
22										
24										
30										
36										
42										
48										

NOTE.—Please erase all the sketches to which your figures on this sheet do not refer.



NOTE.—Letters in cuts refer to similar letters in above columns.

The above flanges are made from wooden / metal patterns and drilled with multiple / plain drills, holes being laid out by hand. / jig. A change from ours to above standard will involve an alteration costing us an amount detailed as follows :

Drawings, Tracings and Records...
Patterns, Flasks, etc...
Jigs and Templates ....
Special Machines.................................... ....

We will (not) adopt a general standard if endorsed by the A. S. M. E., and accepted by the leading manufacturers.

NOTE.—Please either erase or leave the word (not) in the last paragraph.

NOTE.—Please say if more blanks are wanted, and return promptly.

[Sheet B, No. —.]

AMERICAN SOCIETY OF MECHANICAL ENGINEERS,

12 West Thirty-first Street, New York City.

FOR ENGINE AND PUMP BUILDERS.

Will you kindly fill in this Blank and return to Committee on Standard Flanges, care of its Chairman, Carleton W. Nason, 71 Beekman Street, New York City?

*Name of Firm*____________________

*Address*____________________

*Date*__________

STEAM SIDE. Pressures ranging from __ to __ lbs. per sq. in.								EXHAUST SIDE Back pressures ranging from __ to __ lbs. per sq. in.							
Size.	Diameter of Flange.	Thickness of Flange.	Number of Bolts.	Size of Bolts.	Diameter of Bolt Circle.	Thickness of Pipe Neck.	NOTES.	Size.	Diameter of Flange.	Thickness of Flange.	Number of Bolts.	Size of Bolts.	Diameter of Bolt Circle.	Thickness of Pipe Neck.	NOTES.
⅛								⅛							
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⅜								⅜							
½								½							
⅝								⅝							
¾								¾							
1								1							
1¼								1¼							
1½								1½							
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14								14							
16								16							
18								18							

The above flanges are made from wooden / metal patterns and drilled with multiple / plain drills, holes being laid out by hand. / jig. A change from ours to above standard will involve an alteration costing us an amount detailed as follows:

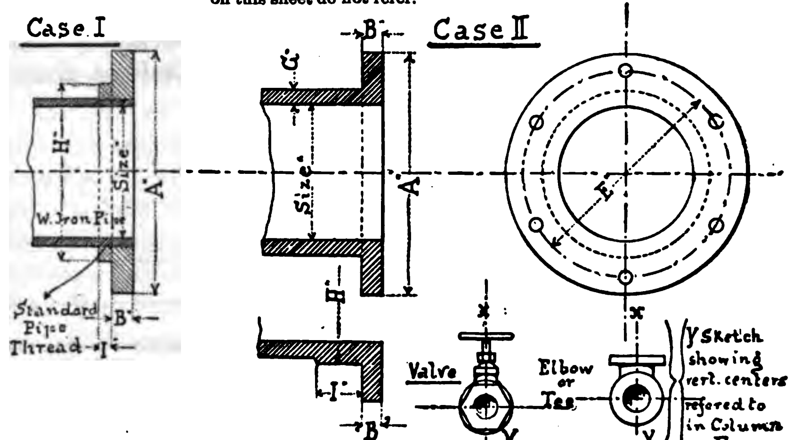
Drawings, Tracings and Records .... Jigs and Templates..........
Patterns, Flasks, etc.................... Special Machines............

We will (not) adopt a general standard if endorsed by the A. S. M. E., and accepted by the leading manufacturers.

Note.—Please either erase or leave the word (not) in the above paragraph.

Note.—Please say if more blanks are wanted, and return promptly.

NOTE.—Please erase all the sketches to which your figures on this sheet do not refer.



NOTE.—Letters in cuts refer to similar letters in above columns.

The above flanges are made from wooden metal patterns and drilled with multiple plain drills, holes being laid out by hand. jig. A change from ours to above standard will involve an alteration costing us an amount detailed as follows :

Drawings, Tracings and Records..
Patterns, Flasks, etc..
Jigs and Templates..
Special Machines...

We will (not) adopt a general standard if endorsed by the A. S. M. E., and accepted by the leading manufacturers.

NOTE.—Please either erase or leave the word (not) in the last paragraph.

NOTE.—Please say if more blanks are wanted, and return promptly.

[Sheet B, No. —.]

AMERICAN SOCIETY OF MECHANICAL ENGINEERS,

12 West Thirty-first Street, New York City.

FOR ENGINE AND PUMP BUILDERS.

Will you kindly fill in this Blank and return to Committee on Standard Flanges, care of its Chairman, Carleton W. Nason, 71 Beekman Street, New York City?

*Name of Firm*______________________

*Address*__________________________

____

Date____________ ________________________

STEAM SIDE. Pressures ranging from ___ to ___ lbs. per sq. in.								EXHAUST SIDE. Back pressures ranging from __ to __ lbs. per sq. in.							
Size.	Diameter of Flange.	Thickness of Flange.	Number of Bolts.	Size of Bolts.	Diameter of Bolt Circle.	Thickness of Pipe Neck.	NOTES.	Size.	Diameter of Flange.	Thickness of Flange.	Number of Bolts.	Size of Bolts.	Diameter of Bolt Circle.	Thickness of Pipe Neck.	NOTES.
⅛								⅛							
¼								¼							
⅜								⅜							
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18								18							

The above flanges are made from wooden / metal patterns and drilled with multiple / plain drills, holes being laid out by hand. / jig. A change from ours to above standard will involve an alteration costing us an amount detailed as follows:

Drawings, Tracings and Records .... Jigs and Templates..........
Patterns, Flasks, etc.................... Special Machines............

We will (not) adopt a general standard if endorsed by the A. S. M. E., and accepted by the leading manufacturers.

NOTE.—Please either erase or leave the word (not) in the above paragraph.

NOTE.—Please say if more blanks are wanted, and return promptly.

[Sheet C, No.—.]

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

12 West Thirty-first Street, New York City.

FOR MANUFACTURERS OF CAST-IRON PIPE AND GATE VALVES.

Will you kindly fill in this Blank and return to Committee on Standard Flanges, care of its Chairman, Carleton W. Nason, 71 Beekman Street, New York City?

Name of Firm ____________________

Address ____________________

____

Date __________ ____________________

FOR WATER.								FOR GAS, OR LOW PRESSURE.							
Size.	Diameter of Flange.	Thickness of Flange.	Number of Bolts.	Size of Bolts.	Diameter of Bolt Circle.	Arrangement of Bolts: If in vertical centre, enter ⦶; if not, enter o|o.	NOTES.	Size.	Diameter of Flange.	Thickness of Flange.	Number of Bolts.	Size of Bolts.	Diameter of Bolt Circle.	Arrangement of bolts: If in vertical centre, enter ⦶; if not, enter o|o.	NOTES.
2								2							
3								3							
4								4							
5								5							
6								6							
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22								22							
24								24							
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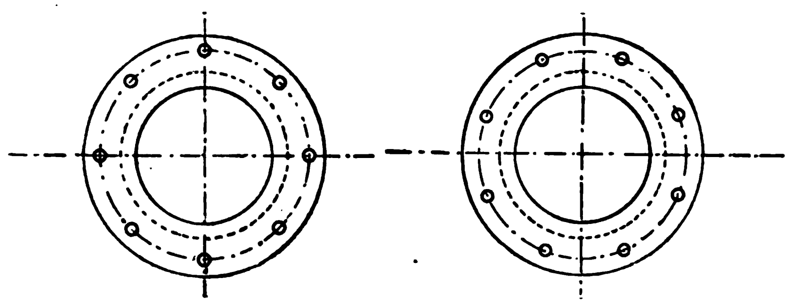
42								42							
48								48							

This sketch shows the position of holes mentioned in the last of the above columns, in case of valves.



The above flanges are made from wooden/metal patterns and drilled with multiple/plain drills, holes being laid out by hand/jig. A change from ours to above standard will involve an alteration costing us an amount detailed as follows :

Drawings, Tracings and Records.......... ....
Patterns, Flasks, etc.................................
Jigs and Templates..................................
Special Machines........ ....

We will (not) adopt a general standard if endorsed by the A. S. M. E., and accepted by the leading manufacturers.

NOTE.—Please either erase or leave the word (not) in the above paragraph.

NOTE.—Please say if more blanks are wanted, and return promptly.

[Sheet D, No.__.]

Will you kindly fill in this Blank and return to Chairman of Committee on Standard Flanges, Carleton W. Nason, 71 Beekman Street, New York City?

Name of Firm ____

Address ____

Date ____________________

STANDARD FLANGES.

	Wrought-iron pipe, with	Wrought-iron Flanges. / Steel " / Brass " / Cast-iron "	for	Steam / Water or oil / Gas /	pressure from _____ lbs. per square inch to _____ lbs. per square inch.
CASE 1 TO COVER	Steel " "				
	Copper " "				
	Brass " "				
	Cast-iron " "				

CASE 2

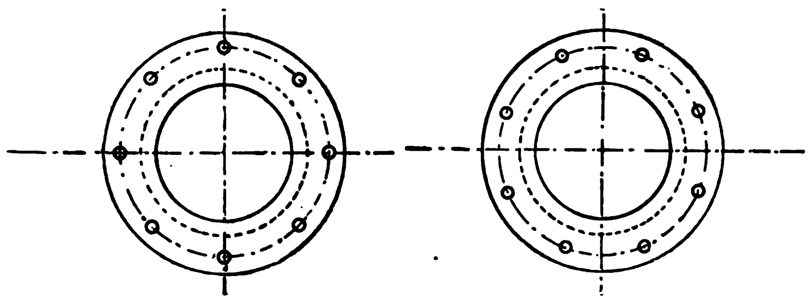
CASE 3

CASE 4

NOTE.—While the Committee desires to make the proposed standard as simple as may be, it has been suggested that it will be impossible to include the entire range of material and pressures now covered by manufactured flanges. We therefore request your views on the subject and ask you to alter the above by filling in or crossing out, so as to indicate the number of standards which you regard as necessary to cover work which should be brought under the Standard system.

NOTE.—Please say if more blanks are wanted, and return promptly.

This sketch shows the position of holes mentioned in the last of the above columns, in case of valves.



The above flanges are made from wooden / metal patterns and drilled with multiple / plain drills, holes being laid out by hand / jig. A change from ours to above standard will involve an alteration costing us an amount detailed as follows:

Drawings, Tracings and Records..........................
Patterns, Flasks, etc.................................
Jigs and Templates....................................
Special Machines......................................

We will (not) adopt a general standard if endorsed by the A. S. M. E., and accepted by the leading manufacturers.

NOTE.—Please either erase or leave the word (not) in the above paragraph.

NOTE.—Please say if more blanks are wanted, and return promptly.

The report was accepted and the Committee continued.

The following letter from Professor Thurston was read and passed for record:

ITHACA, N. Y., *April* 20, 1892.

TO THE COUNCIL, AMERICAN SOCIETY OF MECHANICAL ENGINEERS, NEW YORK CITY.

Gentlemen:—I am requested by the President of the Commission appointed to erect a monument to the memory of our late colleague and honorary member, GUSTAV ADOLPH HIRN, to present the subject to the friends and admirers of that great engineer and greater Man, in this country and in the Society. The statement is made, that, although it is intended that the memorial shall be mainly the work of his fellow-citizens at his birthplace, it is considered that it would be pleasant, also, to embody therein some testimonials of respect and affection from his colleagues in other countries, and especially in America. Contributions will therefore be received, for a limited time, from members of our Society; which contributions may be forwarded either to the officers of the society or to any member of the committee, and it is thought that a considerable number of those who are acquainted with the work of this great man, and who have admired his character as reflected in his works, may esteem it a privilege to so contribute.

Reference to Volume XI. of the *Transactions* of the Society for 1890, Paper No. CCCCXII., in which are given condensed accounts of the lives and works of deceased members, will show those of our colleagues who are unfamiliar with this branch of engineering, or with the extent and character of work done in Europe, what was the position in the profession, and what the grounds for that fame which our deceased friend attained. Those who are working in the same general direction, and are, therefore, already acquainted with the work and the worker, will need no reminder. HIRN was, in his time, undoubtedly that man who most extensively and successfully applied scientific methods and knowledge of physical science to the investigation of the phenomena of energy-transformation as illustrated by the action of the steam-engine. It was he who first and most completely revealed the action of the steam-jacket, made quantitative determinations of the wastes, by internal transfer. of heat supplied the engine in the steam. His corroboration of the work of JOULE and the thermodynamists, alone was enough to give him immortality. His study of the effects of lubricants in the reduction of wastes of power by friction was another example of his scientific and thorough methods and their fruitfulness. His numerous inventions and constructions placed him in the front rank as a practitioner; his great treatise on Thermodynamics and the Steam-engine, and his many works of a minor character on this and various other subjects in applied science, give him place among the great authors of his time.

"Take him all in all," the world has seldom seen so great a man, and our profession has good reason for congratulation that so great a mind found place in this department. Our Society has especial reason for pride in his acceptance of membership with us, and in the privilege of fellowship with so noble and so mighty an intellect. The committee in charge of the work of erection of a monument to our colleague in Alsace are entitled to our thanks for their thoughtfulness in tendering us the privilege of contributing to that expression of appreciation of the character and work of the great engineer and philosopher.

The monument which it is proposed to erect is designed by Bartholdi, the intimate of Hirn, as well as friend of the United States, the artist to whom our country is indebted for that magnificent testimonial of international friendliness which stands in the harbor of New York, as a permanent reminder of the bonds existing between France and her citizens and the United States and their citizens. It will be of bronze, and will bear simply the words and figures;

G. A. HIRN.
1815—1890.

It will be mainly constructed by his fellow-citizens of the town of Colmar; but it will be considered both appropriate and gratifying should contributions—preferably of small amount and great in number—be sent from this country by those who are most interested, and who appreciate the importance of the work done by the man, and his admirable personal qualities.

Such contributions as members may esteem it a privilege to make to this object may be sent directly to the president of the committee, M. G. Kern, at Colmar, Alsace, or to either member of the committee in the United States.

It has been suggested that the presentation, in this brief manner, of this subject, at the approaching meeting of the Society, may serve to bring it before those who would be glad of the opportunity to take part in this movement, and this letter is therefore written, not as a plea, but as a means of presenting an opportunity to those who may appreciate it.

Very respectfully yours,
ROBERT H. THURSTON.

The Chairman then read the following letter from the President of the Society.

239 Clermont Avenue,
BROOKLYN, N. Y., *May* 7, 1892.

PROF. F. R. HUTTON, *Secretary* A. S. M. E., PALACE HOTEL, SAN FRANCISCO, CAL.

Dear Sir:—Out of the regret that possesses me in being debarred from meeting with the membership of our Society in convention assembled, I find a restricted pleasure in bringing myself in touch with its transactions by the exercise of the Presidential function according to usage in the appointment, under Article 31 of the Rules of a Nominating Committee, to present the names of nominees for the offices falling vacant at the end of the current year.

For this committee I have selected the following named gentlemen:

Mr. John C. Kafer, New York.
Mr. S. T. Wellman, Thurlow, Pa.
Mr. John Fritz, Bethlehem, Pa.
Mr. J. L. Gobeille, Cleveland, O.
Mr. S. B. Whiting, Boston, Mass.

I name also as alternates, should any of the above gentlemen find it inconvenient to serve,

Mr. M. C. Bullock, Chicago.
Mr. F. G. Coggin, Lake Linden, Mich.

You will please make this known to the convention, with my earnest wishes that all the gratifications promised by the programme and itinerary are being fully realized.

Respectfully yours,
CHAS. H. LORING, *President.*

At the conclusion of this business the session adjourned.

FOURTH DAY. MAY 19TH.

The morning of this day was devoted to a visit to Sutro Heights and the residence of Mr. Adolph Sutro. The party were conveyed by cable cars and the Cliff Railway out to the grounds, and were received at the gates by their host in person. Escorted by him, they wandered about the beautiful grounds, down to the old Cliff House, where the seal rocks are in view, and thence to the new bath tanks on the shore, in which their philanthropic projector is much interested. After this tour of inspection the party returned to the house, and were entertained at lunch by their host, where, as they sat, their eyes could rest upon the ocean and its mountainous shores. A souvenir album of Sutro Heights and San Francisco lay on each plate. After luncheon, carriages drove the party back to town through the Golden Gate Park, bringing the ladies to the hotel and the gentlemen to visit the Pacific Rolling Mill and the Union Iron Works at the Potrero. They were met at both places by escort parties, who piloted them through the extensive areas of the shops. On the return, a tug at the wharf brought the party back to the principal wharf district, whence they could easily reach home.

In the evening was held the fourth and concluding session, assembled for convenience in the rooms of the California Camera Club in the Academy of Sciences building. The principal feature of the evening was a presentation, by Mr. W. R. Eckart, of a series of lantern views of machinery and other engineering attaching to the deep-mine practice in Nevada. These slides have since become the property of the Society by gift from Mr. Eckart, and will be much appreciated.

In the course of the evening the following series of resolutions was handed in and were passed with acclamation:

Resolved, That the American Society of Mechanical Engineers wishes to express to Mr. H. J. Small, and to the other officers and employees of the Southern Pacific Railway, its appreciation and hearty thanks for the entertainment of the members of the Society during its visit to Sacramento. The drive in their Garden City and the visits to the State Capitol and Crocker Art Gallery form very pleasant memories of the stay in that part of the Pacific slope.

Resolved, That the American Society of Mechanical Engineers desires to express its thanks for the most enjoyable drive and entertainment at Monterey in this State, and to couple with this vote of thanks the names of Mr. A. J. Molera and Prof. H. B. Gale, who were the channels through whom this courtesy flowed to us.

Resolved, That the American Society of Mechanical Engineers take this means of extending to Prof. H. B. Gale and to the authorities of the Leland Stanford, Jr., University, at Palo Alto, their heartiest thanks for the courtesies extended to them during their recent visit to the University and to the Stock Farm.

Resolved, That the American Society of Mechanical Engineers extend to his Honor, George H. Sanderson, Mayor of San Francisco, their cordial recognition for his eloquent and warm welcome to the city of which he is the chief executive officer.

Resolved, That the thanks of the American Society of Mechanical Engineers be extended to Mr. Irwin C. Stump, president of the Mechanics' Institute of this city, for his kindness in offering the use of the rooms and library of that Institute to the members of the Society during the Convention of that Society at San Francisco.

Resolved, That the American Society of Mechanical Engineers deeply appreciates the courtesies extended to them by Mr. John Richards, President, and Mr. Otto Von Geldern, Secretary, of the Technical Society of the Pacific Coast, and thanks the directors of that Society for the privilege extended of using the rooms and library of the Society, so kindly given.

Resolved, That the hearty thanks of the American Society of Mechanical Engineers be extended to Dr. H. W. Harkness, President of the Academy of Sciences, of San Francisco, for the hearty welcome given to the Society and for the many courtesies shown by him during their stay ; and also be it further

Resolved, That Dr. Harkness be requested to convey to the Board of Trustees their appreciation for the use of the Academy's hall for the meetings of the Society.

Resolved, That the American Society of Mechanical Engineers desires to express to the presidents and directors of the cable railway systems of San Francisco its hearty thanks for the courtesies of the roads of the city, whereby the members of the Society have been made their guests during their stay. The members appreciate how much the city owes to the enterprise and foresight of these companies, and their generous hospitality becomes by so much the more valuable in their eyes.

Resolved. That to Mr. and Mrs. A. S. Hallidie, of this city, the members of the American Society of Mechanical Engineers desire to express their most delighted appreciation for the interest which they have taken in making their stay most enjoyable. To Mr. Hallidie they owe a debt of obligation for the opportunity and guidance to visit the power houses of the cable railways of this city, and to Mrs. Hallidie they owe a most pleasant recollection of social courtesies, which they will carry with them for a long time in their distant homes.

Resolved, That the American Society of Mechanical Engineers desires to express to Mr. Henry T. Scott, President of the Union Iron Works of this city, their most sincere appreciation of the opportunity to visit their most extensive plant and yard, and to recognize how important a factor this progressive establishment is destined to be in upbuilding the industrial future of the Pacific Coast.

Resolved, That the American Society of Mechanical Engineers desires on its own behalf, and of that of its ladies, to express to Mr. Adolph Sutro, of this city, a warm sense of its appreciation of the courtesies of their host, in the invitation and entertainment at Sutro Heights. Amid scenes of so much beauty and grandeur as environed the charming home where we were entertained, the public spirit which opens such attraction to visitors seems all the more notable and to be admired. We thank him heartily.

• *Resolved*, That the American Society of Mechanical Engineers wants to convey by this resolution to the Messrs. Spreckels of San Francisco its sincere thanks for the use of their tug *Fearless*, whereby their harbor excursion has been made so enjoyable a feature of their visit to San Francisco.

Resolved, That the American Society of Mechanical Engineers desires to thank the Pacific Rolling Mill of this city for the opportunity to inspect its plant, and for the courteous and considerate attention which was there shown to the party.

Resolved, That to Mr. Charles Webb Howard, President ; to H. Schussler, Chief Engineer, and to the directors of the Spring Valley Water Company, the American Society of Mechanical Engineers desires to express their hearty thanks for their bountiful hospitality, which gave to the visiting members of the Society a day ever to be remembered with the keenest delight.

Resolved, That to the Mercantile Library of San Francisco, and to Mr. A. J. Molera, acting on its behalf, the American Society of Mechanical Engineers desires to express its sincere appreciation for the courtesies of the use of that library during the stay of the visitors.

Resolved, That the American Society of Mechanical Engineers desires to express to the faculty and other officers of the University of California, and especially to Prof. F. G. Hesse, their thanks for the invitation and opportunity to visit the University and Berkeley, and to couple with this vote an expression of their warm wishes for the success of the institution.

Resolved, That to Mrs. James Spiers the ladies of the American Society of Mechanical Engineers wish to express their thanks for her generous entertainment and drive, and for the visit to the hospitable home.

Resolved, That to the Local Executive Committee of Arrangements, Mr. W. R. Eckart, Chairman, and Charles G. Yale, Secretary, and to the devoted Ladies' Committee of Entertainment, the members of the American Society of Mechanical Engineers find it difficult to express in fitting terms even a small part of what they feel and desire to convey of recognition and appreciation for the signal success of the San Francisco meeting of the Society. They leave the Pacific Coast with regret, but can at least rejoice that they carry with them pleasant memories of new friendships formed, and anticipations that at some future time these pleasant ties now to be severed will again be reunited.

Resolved, That the American Society of Mechanical Engineers desires to express to Admiral John Irving, to Chief Engineer John W. Moore, and to the other naval officers stationed at Mare Island, their thanks for the courtesies extended to the Society during the delightful visit to the Navy Yard.

Resolved, That to Mrs. Bellingrodt, and to the other ladies whom she associated with her, the ladies of the party of the American Society of Mechanical Engineers desire to express the pleasure which they felt at the abundant and beautiful supply of flowers, by which their entry into California was greeted through their care.

Resolved, That the American Society of Mechanical Engineers desires to express to Captain Taylor, President of the Risdon Iron Works; to the Fulton Iron Works, to the National Iron Works, to the Pelton Water Wheel Co., and to Messrs. Cahill & Hall, their thanks for the courtesies of invitation to visit their establishments, and has to express regret that the limited time at their disposal precludes the possibility of a visit in a body to these points.

Resolved, That the thanks of the American Society of Mechanical Engineers are hereby tendered to the California Camera Club for the use of its room and exhibition facilities upon this our closing session.

Resolved, That the thanks of such part of the American Society of Mechanical Engineers as participated in the transcontinental excursion to San Francisco, are due, and are hereby tendered, to Mr. and Mrs. E. P. Bates, of Syracuse, N. Y., who out of their own disappointment so brightened the entire train by their beautiful gifts of flowers.

To Mr. H. F. J. Porter, of Chicago, Ill., and his associates connected with the administration at the grounds of the Columbian Exposition, for the energetic and complete manner in which the visit to the several buildings of the Exposition was arranged. It is requested that the thanks of the party may be conveyed through Mr. Porter to those whom they should reach.

To Messrs. Fraser & Chalmers, and to Mr. Chalmers in particular, for the opportunity to visit their plant, and for the entertainment at the theatre in Chicago of the delayed party. They also recognize the hands of the same generous hosts in the floral remembrance which reached the party at Salt Lake City through Mr. L. C. Trent.

To Mr. Wm. J. Silver, for his kindly devotion of his time to the pleasure of the party during their visit to Salt Lake City.

The Chairman then pronounced the Convention at an end.

While the concluding session was in progress the ladies of the party had been assembling by invitation at the residence of Mr. and Mrs. A. S. Hallidie, on Washington Street, at a reception tendered to the Society by these hosts. The members came in later, but in time to enjoy a reading by a San Francisco lady, descriptive of her mother's experiences and privations in crossing the continent in 1849. The contrast with the present luxury made the narrative the more interesting.

FIFTH DAY. MAY 20TH.

The afternoon of this day being the time set for the departure of many of the party, there was no assigned excursion provided for, but the party used the morning in different ways. Visits were paid to the Fulton, National, and Risdon Iron Works, to the Pelton Water Wheel shops, and to the elevator works of Cahill & Hall, and elsewhere. A large party, under the guidance of Mr. James Spiers, accompanied him to Oakland, and on a drive to Berkeley and to the Institution for the Deaf, Dumb, and Blind (where some wonderful illustrations of success in teaching were shown), and to the grounds of the University of California. Here the party was under the guidance of Prof. F. G. Hesse, of the university. Luncheon was served at the residence of Mr. Spiers, and the party returned to town in time to take trains for the Yosemite Valley trip and for Southern California.

CCCCLXXXII.*

ON COMPOUNDING CENTRIFUGAL AND LOAD GOVERNING BY A ROTARY-PISTON VALVE.

BY WM. S. ALDRICH, BALTIMORE, MD.

(Member of the Society.)

The matter of governing steam-engines for uniform speed, and for economy in the use of steam, becomes more difficult as the size of engine is increased. Especially exacting are these requirements in the electrical service, whether for light, power or railway work, with the extended use of marine engines of the vertical-inverted multiple-expansion type. These are directly coupled (or, at most, directly belted) to large multipolar dynamos—two or more for each engine unit. The rotative speeds are consequently much higher than in the same type of engines at sea, and the boiler pressures are also greater. The concentration of great engine power, in a limited space, becomes a serious question in the construction of large municipal electrical supply plants, in or near the centre of the area of distribution. These considerations, with the demand for economy, have led to the adoption of the marine engine in central station work.

The inherent difficulties of governing steam-engines cannot be overcome. The only effective control of the steam supply is during admission, whether by controlling the pressure or the point of cut-off, or both. The variable driving effort on the piston, due to expansive working, the variable effect of the inertia of the reciprocating parts, and the variable crank efforts, can only be more or less met by the steadying action of a well designed fly wheel.

The nature of the problem presented—in view, especially, of marine engines in central stations—may be outlined as follows:

(1) The speed regulation to be as nearly perfect as the operation of the governing forces and of the regulating mechanism will permit.

* Presented at the San Francisco Meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

(2) The steam supply to be as directly proportioned to the external load as the mechanism of the steam-engine will allow, by varying the mean effective pressure on the piston.

(3) Automatic regulation of the point of cut-off alone; or, of all the events of the steam-stroke, if desired.

(4) The valve, valve mechanism and governor mechanism to be as free as possible from any needless frictional resistance, and to require a minimum power to alter their motions in sense, direction or magnitude.

(5) The governing forces to be brought into operation upon any change of speed or of external load, or of both at the same time, by *compounding* the centrifugal and the load governing principles.

(6) The centrifugal governing forces to operate on the valve mechanism at a minimum required variation of speed.

(7) The load, or dynamometric governing forces, to be indirectly applied, and not through the intervention of mechanism by and through which the power is transmitted from the engine to the dynamo.

(8) The instantaneous variations of the external load to be as immediately and positively felt at the valve as the load governing mechanism will permit.

(9) The variations, through any cause, of the driving effort on the piston and internal friction of the engine, as distinct from the external load variations, to be controlled by effective and sensitive centrifugal governing.

(10) The compounding of the centrifugal and the load governing forces to be effected in such a way that they shall operate upon the valve independently of each other, or more or less dependently.

In discussing Mr. F. H. Ball's paper before this Society on the new principle in steam-engine governing, Mr. W. C. Kerr* states the primary requirements of an engine governor to be: (1) A reasonably small variation in speed, light or loaded; (2) a very small acceleration for an instant, following a change of load; and, where the most delicate regulation is required, this instantaneous acceleration is of much greater consequence than the difference in revolutions between light and loaded conditions.

* Transactions of the American Society of Mechanical Engineers, Vol. IX., 1887–88, pp. 315, 316.

A compound automatic cut-off valve mechanism, operated by centrifugal and load governing forces, presents a somewhat satisfactory solution of almost all of these questions. Consider the case of a single valve for a single cylinder engine. The two governing forces may be introduced by making the valve of the piston type, and giving it a variable movement of reciprocation, with a variable movement of rotation or of oscillation, the movements being independent of each other. The valve friction will be reduced to a minimum by combining reciprocation with rotation. The centrifugal governing forces control the variable reciprocation; the load governing forces, the variable rotary movement, or *vice versa*.

An electro-magnetic mechanism under control of the turning moment of the dynamo, or of its external electrical circuit, or of a circuit controlled by the variable load at a distant point, introduces the load, or dynamometric principle. This electro-magnetic control of the point of cut-off not only possesses great facility of application, but differs from all other applications of the dynamometric principle in not requiring intermediate transmissive mechanism, with gear-wheel or pulley trains, nor weights, springs, or belt-tension devices of ordinary transmission dynamometers. It also acts instantaneously upon any change of load on the dynamo or other electrical circuit; and, it is not possible to anticipate further any load changes, because the slightest change of load is instantly felt at the valve, through the electro-magnetic mechanism controlling one of its movements.

An analysis of some of the combinations more or less desirable, as to the steam supply, gives to the reciprocating movement the main control, and to the rotary movement the auxiliary or independent cut-off, or *vice versa*. Both rotary and reciprocating movements may be operated independently or in conjunction with each other. In a single rotary-piston valve all the points of the steam stroke are under control. Arranged for the Corliss type, the rotary-piston valves would be used at each end of the cylinder to control the admission and cut-off, while release and compression are fixed for the best normal working conditions. In multiple-expansion engines, each cylinder is fitted with a rotary-piston valve; though, in some cases, load governing forces may alone be used to control the steam supply for the low-pressure cylinder.

The piston valve, placed vertically in the vertical-inverted marine type of engines, reduces valve-friction to a minimum, when it is rotated and reciprocated, each to a continuously varying extent. Variable steam admission and cut-off is effected by helical or other curved ports in the piston valve and its seat. The two elements of the helical port may be designed independently or otherwise, according to the conditions under which the valve is to operate. The valve will thus have large port opening when most needed, and quick cutting-off qualities.

A compound movement of the stem of the rotary-piston valve may be accomplished, for instance, by giving the centrifugal shaft governor control of the reciprocation; and, the electro-magnetic mechanism control of the rotary movement, by and through a turning sleeve (as a guide block) fitted over a squared portion of the reciprocating valve stem. Many other simple devices, to get this compound variable helical motion of the valve, will suggest themselves as circumstances arise.

The mechanism for both variable reciprocation and rotation may be any one of the many well known forms; such as, the pendulum, or the compound eccentric, for reciprocation; and quick-return-motion mechanism of the Whitworth or other types for rotation. Or, mechanical combinations may be replaced by their kinematical equivalents in the form of electro-magnetic mechanisms. The peculiar advantage of the latter in this case is that the variable electro-magnetic forces, under control of the dynamo or other electrically transmitted load variations, may be applied directly to the valve stem at the most convenient point, and without any reducing or other intermediate mechanism of a mechanical nature. This will also reduce the friction in the governing mechanism.

The electro-magnetic mechanism employed to control the valve movement, more or less in proportion to the external load, may be of several typical forms, utilizing different principles in the operation of electro-magnetic machinery.* The solenoid mechanism, for reciprocating rectilinear or curvilinear movement, having its coils connected in electrical circuit with the dynamo, and its iron core plunger connected to the valve mechanism; the armature mechanism, for partial or complete rotatory movement, whose small shaft is the valve stem, which is thus

* See Notes on Electro-Magnetic Machinery, *Journal of the Franklin Institute*, February and March, 1892.

brought under electrical control of the dynamo current; and an armature mechanism utilizing a proportional part of the turning moment (torque) of the main dynamo which is directly coupled to the engine (or may be otherwise driven),—these are some of the combinations desirable. In fact, these are sliding or turning pairs, under direct electro-magnetic control of the dynamo or other external load, and which may be used with the complementary movement from the centrifugal governor mechanism to produce the resultant variable helical movement of the rotary-piston valve, and so control the steam supply.

The fundamental principle of a rotary-piston valve makes it possible to apply it to other cases than those here considered especially. It admits of controlling the point of cut-off by hand with reversibility, as in the Stephenson and other link motions for which it may be substituted, so adapting it to the locomotive and marine service, where no special uniformity of speed is called for. A hand-wheel on the valve stem, or connected by suitable mechanism, would give absolute control of the steam supply and point of cut-off in these engines, whose eccentrics are fixed (of constant eccentricity). The reciprocations are made by a fixed eccentric, in the "Trenton" steam-engine, built by the Phœnix Iron Company; and here the partial rotation is controlled by the centrifugal shaft governor, or tripping mechanism and dash-pot accompaniment of a fly-ball governor of the Corliss type.

On the other hand, the rotary movement may be made the chief element of the valve motion, and the reciprocation controlled by hand, by an auxiliary device. One rotary-piston valve may thus perform all the functions of an independent cut-off, as in the Buckeye and Meyer types, whether reciprocation or rotation be the chief element of its motion. In the latter, too, the rotary movement may be a partial or complete rotation, and either of these uniform or variable.

The rotary-piston valve, with its variable compound helical motion, possesses the advantages of a slide valve and a piston valve, and the inherent disadvantages of neither. There is no scoring or grooving of the valve face or its seat; hence minimum liability to leakage and a maximum duration of normal working conditions. The structural advantages of piston valves are all maintained. By giving a rotary motion to a piston valve, or a sliding motion to a rotary valve, as is here done, the governing forces are a minimum to effect any change in either motion of the valve; for, the angular movement is more easily varied than it would be with no sliding, and the travel of the valve is more easily changed than with no rotation. These features make it very easy to control one or the other movements entirely by hand (in locomotive and marine engines), or by the sensitive governing mechanism referred to, for the engines in central stations and other power plants; and in the latter case this becomes a marked advantage, as it allows of dynamometric governing by delicate adjustments of the electro-magnetic mechanism to suit the external load conditions.

The advantages of governing steam engines, by compounding the centrifugal and the load governing forces, as by a rotary-piston valve, may be thus summed up:

(1) Best combination of conditions for uniform speed under no load or extreme variations of load, and steam pressure constant or only slightly variable.

(2) Steam supply as directly and instantaneously proportioned to the external load as it is possible to make it by electromagnetically controlling the point of cut-off.

(3) Valve friction reduced to a minimum.

(4) Load governing forces capable of the most sensitive and delicate adjustment by electrical means.

(5) Independent governing against variations in the external load, compared with variations of driving effort and of the internal load of the engine.

(6) Each governing principle regulates that which the other cannot, and without interference.

(7) Rotary-piston valve, with helical ports, gives quick and sharp cut-off, with increased port area.

(8) Failure of either governing device permits of the engine running temporarily under control of the other.

(9) Dynamometric governing of central station multiple-expansion engines driving several dynamos, any one of which may be thrown on or off at any time.

brought under electrical control of the dynamo current; and an armature mechanism utilizing a proportional part of the turning moment (torque) of the main dynamo which is directly coupled to the engine (or may be otherwise driven),—these are some of the combinations desirable. In fact, these are sliding or turning pairs, under direct electro-magnetic control of the dynamo or other external load, and which may be used with the complementary movement from the centrifugal governor mechanism to produce the resultant variable helical movement of the rotary-piston valve, and so control the steam supply.

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The question to be presented and the main point in this communication is, what has produced this particular form of evolution in water wheel practice? and why has pressure instead of impulse been the principle, or mode of operation, followed in all countries?

Before attempting any answer to this inquiry, it will be well to further examine or explain, in as simple a manner as possible, the nature of the class called pressure turbine wheels.

A column of water resting upon the vanes of a turbine wheel, which are free on their reverse side, and meet no resistance there, represents complete efficiency less machine friction; and the science of turbines, so to call it, is directed to removing the impeding water and its resistance on the reverse side of the vanes, that is, on the discharge side, after the function of pressure has ceased or has been utilized. It is common to divide the effect of the water, or its functions, in this class of wheels, into gravity, impulse, and reaction, but there is no need of such assumption or introducing the complex nature of these forces thus combined, because the whole is explainable as simple pressure, and all observed phenomena point to this as the "mode of action" in pressure turbines.

I am in this assumption no doubt transgressing upon what are called established data, but the issue is not important to the present subject, and it will be sufficient to call the active force one of pressure alone, and the resistance or loss, a result of an imperfect riddance of the water on the reverse or discharge side of the vanes after it has performed its work by pressure, impulse, or otherwise.

Following this method of operating to its constructive features, it involves closed vessels, or conduits, not only to the water wheels, as in other cases, but around them. They must be enveloped in the fluid that drives them, and contained in cases strong enough to sustain not only the static head, but also the effect of water concussion, and in most cases afford support for the wheels themselves and their shafts.

The bearings of the wheels have to sustain the weight of the running parts, also in many cases a pressure of the head, equal to the area of the issues multiplied into the head. The wheels done to provide data and results for class instruction. Mention of these details is made only to show grounds for the writer's confidence in the tests.

TABLE II.

	1	2	3	4	5
1. Boiler pressure by gauge, pounds	68.7	73.0	68.5	72.9	77.4
2. Pressure of atmosphere, pounds	14.8	14.7	14.9	14.8	14.8
3. Back-pressure, absolute	14.8	14.7	14.9	14.8	14.8
4. Pressure at release, absolute	16.5	18.7	19.5	22.8	30.5
5. Priming in steam	0.02	0.02	0.02	0.02	0.02
6. Cut-off, mean	0.095	0.122	0.150	0.195	0.308
7. Efficiency of actual engine	0.059	0.067	0.068	0.073	0.078
8. " of non-conducting engine, incomplete expans'n	0.125	0.128	0.123	0.122	0.120
9. " " " " complete expansion.	0.126	0.129	0.126	0.127	0.133
10. " " Carnot's Cycle	0.133	0.136	0.133	0.136	0.140
11. Horse-power	6.53	8.46	8.86	11.13	16.35
12. B. T. U. for H. P. per min., actual	714	631	628	577	548
13. " " " " incomplete expansion	339	331	345	348	353
14. " " " " complete expansion	332	326	332	327	319
15. " " " " Carnot's cycle	319	312	319	312	303
16. Steam per H. P. per hour, actual	43.79	38.64	38.40	35.33	33.50
17. " " " " Carnot's cycle	21.49	20.95	21.49	20.95	20.42
18. Ratio of steam consumptions, actual and by Carnot's cycle	0.49	0.54	0.56	0.59	0.61
19. Ratio actual efficiency to that of Carnot's cycle	0.45	0.49	0.51	0.54	0.56
20. Ratio of actual efficiency to that for incomplete expansion	0.47	0.52	0.55	0.60	0.65

The cycle for the non-conducting engine with incomplete expansion, in this calculation, is assumed to have the same pressure at the end of the stroke, as the pressure at release, in the test with which that cycle is compared. Such a non-conducting engine would have a smaller cylinder and a longer cut-off when developing the same power as the actual engine. An assumption of equal cut-offs for the actual cycle and the cycle for the non-conducting engine with incomplete expansion, would not give logical results and would involve troublesome calculations.

In examining the table the following points may be noted:

(1) A comparison of the efficiencies and of the B. T. U. per horse-power per minute, gives the same result.

(2) The slight variations of the efficiencies of the several tests for Carnot's cycle and for the non-conducting engine with complete expansion, are due to the variations of the boiler-pressure; in any single test the boiler-pressure was maintained very nearly constant.

(3) The ratio of the actual steam consumption to the hypothetical consumption of Carnot's cycle for each test, is notably larger than the ratio of the efficiencies of the engine and of Carnot's cycle; any conclusion from such a comparison is subject to an error that may be large.

(4) The steam consumption and the B. T. U. per horse-power per minute, decrease in the familiar manner for such a series of

other branches of dynamic engineering at the present day, especially when the economic and constructive conditions are so much in favor of the impulse type of water wheels are taken into account. These we will now consider in a brief way.

There is a wide difference between a water wheel driven by impulse and one operating on the pressure system. The first cost of the former, for a given power, is one-half as much, and its maintenance is still less, in proportion.

Figuratively speaking, when a wheel is changed from the pressure to the impulse system it is taken out of its case, mounted in the open air, in plain sight. All the various inlet fittings are dispensed with and are replaced by a plain nozzle and stop valve. Its diameter is made to produce the required rotative speed, whatever that may be. The shaft and its bearings are divested of all strains except those of gravity and the stress of propulsion when the water is applied at one side only. Most important of all there are no running metallic joints to maintain against the escape of water, no friction and no leaks ; there are, indeed, no running joints or bearings whatever, except the journals of the wheel shaft.

The effect of grit and sand is eliminated, both as to vanes and bearings, and there are no working conditions which involve risk or which call for skill. If a vane is broken, another one is applied in a few minutes' time. If a large or small wheel is wanted, the change is inexpensive and does not disturb the foundations or connections.

Capacity is at complete control; the wheels can be of 10, 100, or 1000 H. P., without involving expensive special patterns. The speed of rotation is not confined to commercial dimensions because of patterns or other causes. It is merely a matter of choice with the purchaser or maker.

Now granting the efficiency of impulse wheels, which, as before remarked, can hardly be called in question for all heads exceeding 50 or even 30 feet, and conceding the constructive and operating advantages just pointed out, the question at first named arises, why has the evolution of water wheels during fifty years past been confined to the pressure class? also, why has it been proposed at Niagara Falls to employ pressure turbine wheels under a head of 100 feet or more, when the conditions point to the better adaptation of open, or impulse wheels?

It is not necessary in such an inquiry to discuss the problem

In the calculation of this table the back-pressure is assumed to be 1.5 lbs. absolute, and the steam supplied is assumed to be dry and saturated.

There is no known way of calculating the steam consumption of an actual engine from that of a non-conducting engine; and for an engine of the type used when high economy is expected, there is the added difficulty that part of the steam is condensed in the jackets and does not pass through the cylinders. A hypothetical limit of steam-engine economy has been found by taking $\frac{4}{3}$ of the steam consumption of a non-conducting engine, for the steam consumption of the actual limit. The quantities thus found are probably too small as the ratio, taken from the table on page 349, is for an engine with a less range of pressure than found in Table III.

DISCUSSION.

Prof. R. H. Thurston.—I presume that no one will dispute the main proposition of this paper: that the best scientific measure of the efficiency of the heat-engine is its consumption of heat measured in thermal units per unit of power developed. This affords a common basis of comparison of all engines transforming heat into work. It is true that the use of the weight of steam as a unit is subject to the objection that it may lead to error; though it should, I think, be said that this only can happen when it is forgotten that the same variations of steam-pressures and feed-water temperatures which affect the efficiency of the engine also affect its value. The point to be noted is that this latter unit is a variable magnitude, while the thermal unit is a constant. Exact scientific comparisons of results of experiment are only possible when all quantities compared can be reduced to a common unit, a recognized fixed standard.

I observe that the writer of this paper takes the B. T. U. as the measure of the specific heat of water at 62° Fahr., while, I think, the great majority of authorities, recognized as such throughout the world, assume the standard temperature as that of freezing or of the maximum density of water. The old British standard temperature here assumed is, if I am not very greatly mistaken, not only not retaining its place outside Great Britain, but is, with all other relics of the old system of weights and

wheels in the proportion of their diameters, and at the same time dispensed with the accurate fitting involved in the outward and downward flow turbines; and this, as before said, has been done without sacrificing efficiency.

The tangential type of open wheels has been similarly dealt with here in California. The running-water joints have been wholly dispensed with. The construction has been cheapened one half. The round jet has been applied in the most simple manner, with an increased dynamic effect, and the efficiency attained is believed to be more than is reached by the finest examples of Girard wheels in Europe.

Conceding these statements and facts brings us back again to the query forming the subject of this communication; namely, Why has the evolution of water wheels followed on the line of *pressure* instead of *impulse?*

CCCCLXXXIV.*

ECONOMY AND EFFICIENCY OF THE STEAM ENGINE.

BY C. H. PEABODY, BOSTON, MASS.
(Member of the Society.)

IT is customary to state the performance of a steam-engine in pounds of steam used per horse-power per hour, a method which is open to objection since the value of a pound of steam depends on the pressure and quality of the steam. It has frequently been urged on the attention of engineers that the British thermal unit (B. T. U.) should be used in stating the performance of engines. This unit, sometimes called the pound-degree, is the heat required to raise one pound of water one degree of temperature; exactly, it is the heat required to raise one pound of water from 62° to 63° Fahr., and it is equivalent to 778 foot-pounds. In order to obtain convenient numerical quantities it is advisable to state engine performance in B. T. U. per horse-power per minute. Incidentally, this method has the advantage that it may be used for any heat engine, such as a hot-air engine or a gas engine.

If an engine consumes S pounds of steam per horse-power per hour, supplied at a boiler pressure of p pounds absolute per square inch and exhausted against a back-pressure of p_0 pounds absolute, then the thermal units per horse-power per minute will be

$$S(xr + q - q_0) \div 60, \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

in which r is the heat of vaporization at the boiler pressure, q and q_0 are the heats of the liquid at boiler pressure and at the back-pressure, while x is the quality of the steam. One pound of moist steam contains x part dry steam and $1 - x$ part of water; so that two per cent. priming will make x equal to 0.98.

The expression (1) is deduced with the assumption that all the

* Presented at the San Francisco meeting (May, 1892), of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

steam passes through the cylinder of the engine; should part of the heat be supplied by jackets, reheaters, etc., then the calculation must be made for each separately; and such a method is the only one which can give the correct cost of running the engine.

As an example, there may be quoted a test made on the triple-expansion engine in the laboratory of the Massachusetts Institute of Technology, and reported to this society.*

In that test the engine used 13.73 lbs. of steam per horse-power per hour, of which 10.86 lbs. passed through the cylinders of the engine, and 2.87 lbs. were condensed in the several jackets of the engine. The B. T. U. supplied to the cylinders per horse-power per minute was

$$10.86\,(0.988 \times 858.3 + 334.2 - 126.4) \div 60 = 191.1;$$

the boiler pressure being 157.7 lbs. absolute, and the back-pressure 4.5 lbs. absolute, while the priming was 1.2%. The B. T. U. supplied by the jackets per horse-power per minute may be calculated by the expression

$$2.87xr \div 60 = 40.6,$$

since the condensed water from jackets may be returned directly to the boiler. The total consumption per horse-power per minute was consequently

$$191.1 + 40.6 = 231.7 \text{ B. T. U.}$$

The method of stating engine performance in B. T. U. is advocated here, not only because it is thoroughly theoretical and practical, but because it affords a simple and correct way of finding the efficiency of the engine, while an attempt to find the efficiency by other methods is liable to lead to confusion and error.

The ordinary definition of the efficiency makes it the ratio of the heat changed into work to the heat consumed. Now a horse-power is 33,000 foot-pounds per minute, equivalent to

$$33{,}000 \div 778 = 42.42 \text{ B. T. U.} \quad . \quad . \quad . \quad (2)$$

per minute. This constant, divided by the heat consumed per

* Trans. Am. Soc. Mech. Eng'rs, Vol. XII., page 740.

horse-power per minute, will give the actual efficiency of the engine. The efficiency for the test of the triple-engine just quoted is

$$42.42 \div 231.7 = 0.183,$$

which is remarkable only in that it is a very high result for an engine of the size.

In order to show how well an engine is doing it should be compared with the performance of an engine that has no waste or losses, i.e., with a perfect engine. The second law of thermodynamics makes the efficiency of such an engine

$$\frac{T - T_{e}}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

in which T and T_e are the absolute temperatures corresponding to the boiler-pressure and the back-pressure. For the test under consideration the efficiency of the temperatures is

$$\frac{822.9 - 618.4}{822.9} = 0.2485.$$

Comparing the actual efficiency of the engine with this efficiency of a perfect engine, it appears that the engine, during the test quoted, was doing

$$\frac{0.183}{0.2485} = 0.736$$

of the work that it was possible to do under the circumstances. Hirn's analysis, applied to this test, shows that the thorough steam-jacketing of the cylinder and receivers of this engine reduced the exhaust waste to an insignificant quantity, and the losses were due mainly to imperfection of the cycle and to radiation.

Carnot's cycle for a steam-engine is represented by Fig. 68. It is made up of the isothermal lines *ab* and *cd*, for a mixture of a liquid and its vapor, together with the adiabatic lines *bc* and *da*. It is supposed that at the beginning of the stroke there is a mixture of steam and water in the cylinder having the pressure and volume indicated by the point *a*; that heat is supplied to this mixture in some method, at constant temperature, while steam passes through the cylinder of the engine; should part of the heat be supplied by jackets, reheaters, etc., then the calculation must be made for each separately; and such a method is the only one which can give the correct cost of running the engine.

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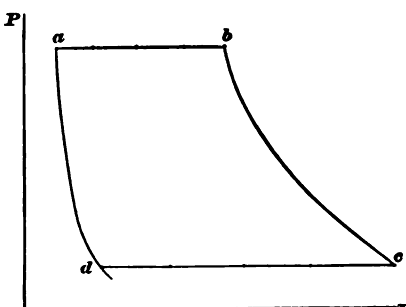

Fig. 68.

Some writers on thermo-dynamics go through the form of calculating the steam per horse-power per hour, though the quantity thus found has no physical meaning and is liable to be misleading. The B. T. U. per horse-power per minute for Carnot's cycle is evidently

$$42.42 \div \frac{T - T_0}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

This quantity multiplied by 60 and divided by r, the heat of vaporization at the temperature $t = T - 460.7$, is called the steam per horse-power per hour. The algebraic expression is

$$42.42 \times 60 \frac{T}{r\,(T - T_0)} = 2545 \frac{T}{r\,(T - T_0)} \quad . \quad . \quad . \quad (5)$$

For the test previously quoted, this calculation gives

$$2545 \frac{822.9}{858.3\,(822.9 - 618.4)} = 11.9 \text{ lbs.,}$$

CCCCLXXXV.*

THE ELECTRIC RAILWAY AS APPLIED TO STEAM ROADS.

BY B. J. DASHIELL, JR., PHILADELPHIA, PA.
(Junior Member of the Society.)

It is the intention of the writer to give some information, for record before this Society, which will throw light upon the question of high-speed train resistance, deduced from tabulated experiments at such speeds.

The first experiments of note at extremely high railroad speed were made by the Electro-Automatic Transit Co., of Baltimore City, in the year 1889, at Laurel, Md., on their circular road, with a 2½-ton car. Some of these experiments were publicly reported by Mr. O. T. Crosby, in his paper entitled "Report of High-speed Electric Railway Work." † Mr. David G. Weems, the originator, contemplated having an automatic mail and express service from one city to another, controlling the cars from central stations located on the line of the road, and which was to be a very complete "block system" in its way. In the early spring of 1888 my services were required by this company, and designs and experimental circular tracks were made and built under my direct supervision. I may here state, for the benefit of some not knowing the circumstances, that the line of road was built for a *limited sum* of money, and that in some of the experiments, owing to some "fixed ideas," the motors were designed for too high a speed. Both of these obstacles gave us quite a little trouble on the start.

The power house consisted of a 30 foot x 40 foot x 12 foot wood building, costing about $600 to build. It was divided into three rooms — boiler-room, engine and dynamo-room, and office. The plant consisted of 125 H.P. in two (2) vertical boilers, 100 H.P. Ball high-speed, automatic cut-off engine, run-

* Presented at the San Francisco meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

† *Trans. Amer. Inst. Elect. Engrs.*, Feb. 24, 1891.

same efficiency as the cycles from which it is derived. The discussion may therefore be limited to the simpler cycle of a non-conducting engine shown by Fig. 70.

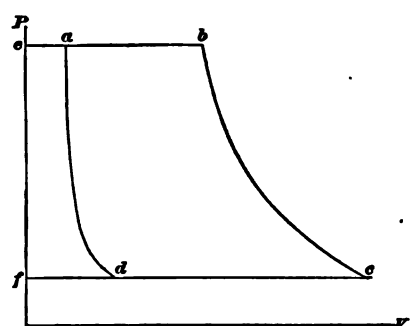

Fig. 69.

The cycle shown by Fig. 70 consists of admission of dry steam as shown by the line *ab*; of adiabatic expansion represented by *bc*; and of exhaust of the resulting mixture of water and steam, as shown by *cd*.

During the admission of steam, the external work

$$W_{ab} = Mp(u + \sigma) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

is done by the isothermal expansion of steam from the boiler into the engine; that is, the heat transformed into work is furnished at the boiler. In the expression (6), M represents the

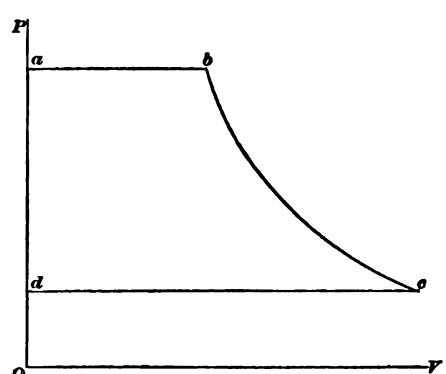

Fig. 70.

weight of steam consumed per stroke, and $u + \sigma$ represents the volume of one pound of steam consisting of σ, the specific volume of one pound of water, plus u the increase of volume due to vaporization when the water is changed into steam. p is the specific pressure in pounds on the square foot. The intrinsic energy of one pound of dry stream at the condition represented by b, *i.e.*, at the pressure p, is

$$\frac{1}{A}(\rho + q) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

in which A is the reciprocal of the mechanical equivalent of heat, q is the heat of the liquid, and ρ is the heat equivalent of the work of disgregation due to vaporization, known as the internal latent heat. After an adiabatic expansion to the pressure p_o, shown by bc, the quality of the steam will be x_o, which can be calculated by the equation

$$\frac{r}{T} + \int_{32}^{t} \frac{cdt}{T} = \frac{r_o x_o}{T_o} + \int_{32}^{t_o} \frac{cdt}{T} \quad . \quad . \quad . \quad . \quad . \quad (8)$$

in which $\frac{r}{T}$ and $\frac{r_o x_o}{T_o}$ represent the entropy due to complete or partial vaporization of one pound of water, and $\int \frac{cdt}{T}$ is the entropy of the liquid. Knowing the value of x_o, the intrinsic energy of one pound of steam and water represented by the point c, may be calculated by the expression

$$\frac{1}{A}(x_o \rho_o + q_o). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

Since the external work during expansion is done at the expense of the intrinsic energy,

$$W_{bc} = \frac{M}{A}(\rho - x_o \rho_o + q - q_o). \quad . \quad . \quad . \quad . \quad (10)$$

The external work required to force the steam from the engine during exhaust is

$$W_{cd} = Mp_o(x_o u_o + \sigma). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

The heat changed into work during the entire cycle is

$$AW = A(W_{ab} + W_{bc} - W_{cd})$$

$$\therefore AW = M\{\rho + Apu - x_o \rho_o - Ap_o x_o u_o + q - q_o + A(p - p_o)\sigma\}$$

$$= M\{r - x_o r_o + q - q_o + A(p - p_o)\sigma\}. \quad . \quad . \quad . \quad (12)$$

The heat that must be supplied by the boiler, by the same reasoning as was used to establish the expression (1), is

$$Q = M(r + q - q_o). \qquad (13)$$

The efficiency of the cycle is

$$\frac{AW}{Q} = \frac{r - x_o r_o + q - q_o + A(p - p_o)\sigma}{r + q - q_o}$$

$$\frac{AW}{Q} = 1 - \frac{x_o r_o - A(p - p_o)\sigma}{r + q - q_o}. \qquad (14)$$

In case the expansion is not carried down to the back-pressure line, the cycle is such as is shown by Fig. 71. The loss of efficiency is that caused by the loss of the area *ecf*. The steam at *e* has a quality represented by $x_e > x_o$; but during the release represented by *ef*, the steam left in the cylinder at the end of the release expels the steam thrown out during release, and does

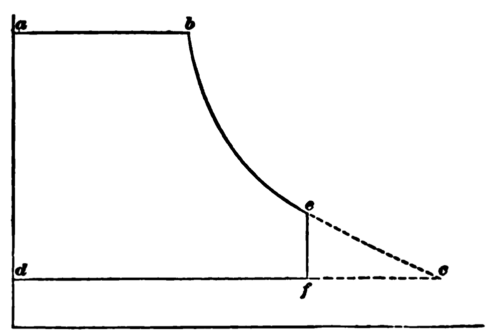

Fig. 71.

work on it by adiabatic expansion as it would do on a piston if expanded behind the piston down to the back-pressure line; consequently the quality of the steam during the exhaust from *f* to *d* is x_o, calculated by equation (8). The conclusion is that the efficiency of a non-conducting engine with incomplete expansion, and with complete compression to fill the clearance space, will be the same as for a cycle represented by Fig. 71.

At the end of expansion the quality of the steam x_e may be found by the equation

$$\frac{r}{T} + \int_{32}^{t} \frac{c\,dt}{T} = \frac{r_e x_e}{T_e} + \int^{t_e} \frac{c\,dt}{T} \qquad (15)$$

The volume at f, is the same as at e and is equal to

$$M(x_e u_e + \sigma).$$

The work during admission of steam represented by ab is

$$W_{ab} = Mp(u + \sigma) \quad \ldots \ldots \ldots \quad (16)$$

The work during expansion is

$$W_{be} = \frac{M}{A}(\rho - x_e \rho_e + q - q_e) \quad \ldots \ldots \quad (17)$$

The work during exhaust is

$$W_{fd} = Mp_o(u_e x_e + \sigma) \quad \ldots \ldots \ldots \quad (18)$$

The heat changed into work is

$$AW = A(W_{ab} + W_{be} - W_{fd})$$

$$\therefore AW = M\{\rho + Apu - x_e \rho_e - Ap_o u_e x_e + q - q_e + A(p - p_o)\sigma\}$$

$$= M\{\rho + Apu - x_e \rho_e - Ap_e u_e x_e + q - q_e + A(p_e - p_o)u_e x_e + A(p - p_o)\sigma\}$$

$$\therefore AW = M\{r - x_e r_e + q - q_e + A(p_e - p_o)u_e x_e + A(p - p_o)\sigma\} \quad (19)$$

The efficiency is

$$\frac{AW}{Q} = \frac{r - x_e r_e + q - q_e + A(p_e - p_o)u_e x_e + A(p - p_o)\sigma}{r + q - q_o} \quad (20)$$

The calculations by the equations (14) and (20) present no difficulty, and do not require much time, provided proper tables of the properties of saturated steam are at hand. In calculating the quantities $A(p - p_0)\sigma$ and $A(p_e - p_0)x_e u_e$ it must be borne in mind that the pressures are in pounds on the square foot, and are 144 times the pressures in pounds on the square inch. The first of these quantities may often be neglected, as it is small.

The B. T. U. per horse-power per minute for these cycles, with complete or incomplete expansion, may be found by dividing the constant 42.42 by the efficiency for the cycle.

The work given here for the complete and incomplete expansion in a non-conducting steam-engine, has no novelty unless it be in the manner of presentation. The same matter, or its equivalent is given by all the earlier writers on the thermodynamics of the steam-engine, and developed by them in form for designing new engines. Experiments by Isherwood, Clark, Hirn, and others showed long ago that the error of neglecting the action of the walls of the cylinder is so large that nothing but disappointment could be expected from an attempt to use the material presented in the works of Rankine, Clausius, and Zeuner, for designing engines. Practical men were wont to say that the work might be right theoretically, but that in practice things were not so, which was their way of expressing disbelief in the whole subject. The writer is of the opinion that such treatment of the problems of the steam-engine is more curious than useful; and that the use can be only in giving some indication as to the degree of perfection obtained or obtainable in engines of a given type. There appears to be a disinclination on the part of engineers to admit that their best efforts give only a moiety of the work that an engine should be able to give for the heat used up; and lately several attempts have been made to compare the actual efficiency of an engine with the efficiency of some ideal cycle intended to approximate to the conditions under which the engine runs. Many such attempts are by approximate methods, which require quite as much labor in application as those stated here, and which are liable to lead to confusion and error. Again, it will frequently be found that as the conditions of running an engine are changed in such a way as to give a better economy and better actual efficiency, the comparison of that actual efficiency with some ideal efficiency, will show that the engine finds it more and more difficult to get what is to be had, as its attempts become more ambitious.

To show the application of the equations (14) and (20) the calculations will be given for the test on the Institute of Technology engine, already quoted several times. The efficiency for an ideal cycle with complete expansion in a non-conducting cylinder for a range of pressure from 157.7 to 4.5 pounds absolute is found as follows. By equation (8)

$$\frac{858.3}{822.9} + 0.5184 = \frac{1004.0\, x_0}{618.4} + 0.2278$$

$$\therefore \quad x_0 = 0.8214.$$

By equation (14) the efficiency is

$$1 - \frac{0.8214 \times 1004.2 - \frac{144}{778}\left(157.7 - 4.5\right)\frac{1}{62.4}}{858.3 + 334.2 - 126.1} = 0.227.$$

The following extract is taken from a report of one of the leading electrical companies.

"The continued success of the electric street railway, and the demands made by street railway companies for larger and more powerful motors to handle their cars, has led others interested in transportation to investigate the advantages of electric locomotion, with the result that not a few electric tramways are in operation throughout the country, hauling freight in our mills, factories, mines, etc. In this department of work, also, there has been a constant demand for more powerful machines, so that where the electric locomotive formerly hauled a few bales of cotton, it is now called upon to handle a fair-sized train."

In compliance with the demands of this nature, one of the companies, some time ago, constructed an electric locomotive designed to obviate the necessity of employing a steam locomotive, and it may, indeed, be said to represent the *first large freight locomotive* displacing steam on a *standard gauge railroad.* The locomotive is at present running at Whitinsville, Mass., carrying merchandise from the railway freight station to the works of the Whitinsville Machine Company's plant, a distance of 1½ miles. The locomotive is illustrated in Figs. 74 and 75. The power is furnished by a large generator located at the machine works, and conveyed over a trolley wire, from which it is taken by means of a universal trolley bar attached to the locomotive. The construction of the truck, etc., is well shown in the engravings, and is built in a square form in three castings, having also a platform for carrying loads, and cow-catchers and draw-bars at each end. The motor employed is one of what is called the "G" type of the Thomson-Houston Electric Company; the power is communicated from the armature to the rear axle by means of double-reduction gearing, and from the rear axle to the forward one by means of the side parallel rods. It will be well to state here that it is planned that all future freight locomotives should have the single-reduction type of motors. The motor consists of wrought-iron field magnets which are bolted to a magnetic yoke of mitis iron. One of these yokes carries the bearings which support that end of the motor on the axle, while the other yoke is spring supported from the other axle. This keeps the gears always in line and meshing correctly with each other, and at the same time provides considerable spring support for the motor, which in designing slow-speed locomotives should be looked

TABLE I.

	Actual engine.	Non-conducting engine.		Carnot's cycle.
		Incomplete expansion.	Complete expansion.	
	1	2	3	4
Efficiency	0.183	0.225	0.227	0.245
Ratio of efficiencies	....	0.813	0.807	0.736
B. T. U. per horse-power per minute	231.7	188.5	187.1	170.7
Pounds of steam per horse-power per hour	13.7	10.6	10.5	12.0
Ratio of steam consumption	....	0.77	0.77	0.87

The ratio of the efficiencies is obtained by dividing the actual efficiency by the ideal efficiency below which the ratio is recorded. The ratio of steam consumptions is obtained by dividing the steam consumption for the ideal cycle by the steam consumption for the actual engine. A comparison of these ratios shows that steam consumption should not be used as a basis for calculating efficiency. The steam per horse-power per hour for Carnot's cycle is that hypothetical quantity mentioned on page 340. The steam per horse-power per hour for the non-conducting engine, in either case, is calculated by dividing the B. T. U. per horse-power *per hour* by

$$r + q - q_0$$

and is the true amount of steam that would be consumed by an engine of that type.

Another application of the methods developed above is made in Table II., to a series of tests made by the instructors in the engineering laboratory of the Massachusetts Institute of Technology, on a Harris-Corliss* engine. The engine has a diameter of 8 inches, a stroke of 24 inches, and ran at about 60 revolutions per minute. The cylinder was lagged with wood in the usual method, but was not steam-jacketed. The exhaust-steam was condensed, at atmospheric pressure, in a surface condenser, collected and weighed. The condensing water was also weighed in tanks, and a complete analysis according to Hirn's method was made, with very satisfactory results. The work was

* Thermo-dynamics of the Steam Engine. C. H. Peabody, J. Wiley & Sons. 1889.

carefully into, as well as in locomotives for extreme high speed. This is a matter of no small moment in designing such work. The gearing consists of aluminum bronze pinions and mitis iron spur gear wheels. This gearing runs in gear cases, in which a plentiful supply of grease is placed. This decreases the noise and friction, thus increasing the life of the gears very materially. On the intermediate shaft is heavily keyed a mitis-iron brake drum, which is covered with wood lagging. It is embraced by two half bands of steel, tightened upon it by means of the brake-drum lever, situated in the operating stand or cab.

The wheels are 42 inches in diameter, and are heavily steel tired. The frame consists of two heavy side plates in which are located the main axle bearing. Two heavy cast-iron plates, in which are cast the cow-catchers, are bolted to the side plates by means of heavy through bolts, which are a driving fit in reamed holes. These end plates carry the heavy spring draw-bars and bumpers.

The operating platform or cab is located at one end of the main platform, and is made of pipe frame-work and covered with a protecting roof. On this platform are located the lever for operating the controlling mechanism, the brake, and the double-acting sand boxes. The universal trolley bar also extends upward from the locomotive at this point, as shown.

The controlling mechanism consists of two large rheostats of the Thomson-Houston railway type. These are so arranged with their contact shoes that no reversing switch is needed. The operator stands so that he always faces the direction in which the locomotive is to go, and being in this position, he pushes the controlling lever from him to make the locomotive go forward, and pulls it toward him, past its vertical line, to make it go backward. A positive centre notch or lock is provided, so that in turning the current off there is no danger of passing the neutral point on the rheostat, and so reversing the locomotive with the current on. When the operator stands in the above-mentioned position he pushes the brake lever from him in order to apply the brake. The steel bands are so arranged on the brake drum that the friction tends to tighten them up more upon the wood lagging, and so assist the operator in braking the train.

The following data gives the detail of construction of this tests on a simple engine, as the cut-off is lengthened. The consumption for the 5th test with a cut-off at about one-third stroke is probably a minimum for the engine, and is a creditable performance for an engine of its size and type.

(5) The efficiency improves with the lengthening of the cut-off in proportion as the consumption decreases.

(6) The efficiency for a non-conducting engine with incomplete expansion is nearly as good as for such an engine with complete expansion for the first test, since the expansion is carried nearly down to the back-pressure. The efficiency for such an engine falls off as the expansion is less complete, *i.e.*, as the cut-off is lengthened. The failure to realize the excellent efficiency shown for such an engine in the first and second tests is due to the action of the walls of the cylinder.

(7) As the cut-off is lengthened and the task set before the engine becomes easier, the ratio of the actual efficiency to that of a non-conducting engine with incomplete expansion improves. For the 5th test this ratio becomes 0.65. The difference between this result and unity may be charged to radiation and to the influence of the walls of the cylinder. Anything that can mitigate the influence of the cylinder walls (such as superheating or steam-jacketing) may lessen the steam consumption by allowing the use of more expansion, *i.e.*, of a cycle that would give a better efficiency with a non-conducting cylinder.

There is a certain fascination in trying to predict the performance of future engines, or at least to set the limit for which we may strive. As a matter of curiosity the following table has been computed for a non-conducting engine with complete expansion, together with a comparison with Carnot's cycle for the same ranges:

TABLE III.

Initial pressure by the gauge above the atmosphere.	Efficiency of Carnot's cycle.	Non-conducting engine with complete expansion.			Hypothetical limit of steam-engine performance.
		Efficiency.	B. T. U. per horse-power per minute.	Steam per horse-power per hour.	
150	0.302	0.272	156	8.63	11.5
200	0.320	0.288	147	7.91	10.5
300	0.347	0.306	135	7.22	9.6

In the calculation of this table the back-pressure is assumed to be 1.5 lbs. absolute, and the steam supplied is assumed to be dry and saturated.

There is no known way of calculating the steam consumption of an actual engine from that of a non-conducting engine; and for an engine of the type used when high economy is expected, there is the added difficulty that part of the steam is condensed in the jackets and does not pass through the cylinders. A hypothetical limit of steam-engine economy has been found by taking ⅘ of the steam consumption of a non-conducting engine, for the steam consumption of the actual limit. The quantities thus found are probably too small as the ratio, taken from the table on page 349, is for an engine with a less range of pressure than found in Table III.

DISCUSSION.

Prof. R. H. Thurston.—I presume that no one will dispute the main proposition of this paper: that the best scientific measure of the efficiency of the heat-engine is its consumption of heat measured in thermal units per unit of power developed. This affords a common basis of comparison of all engines transforming heat into work. It is true that the use of the weight of steam as a unit is subject to the objection that it may lead to error; though it should, I think, be said that this only can happen when it is forgotten that the same variations of steam-pressures and feed-water temperatures which affect the efficiency of the engine also affect its value. The point to be noted is that this latter unit is a variable magnitude, while the thermal unit is a constant. Exact scientific comparisons of results of experiment are only possible when all quantities compared can be reduced to a common unit, a recognized fixed standard.

I observe that the writer of this paper takes the B. T. U. as the measure of the specific heat of water at 62° Fahr., while, I think, the great majority of authorities, recognized as such throughout the world, assume the standard temperature as that of freezing or of the maximum density of water. The old British standard temperature here assumed is, if I am not very greatly mistaken, not only not retaining its place outside Great Britain, but is, with all other relics of the old system of weights and

measures, going out even there. It would, however, be interesting to ascertain precisely what standing it has by comparing the various principal authorities. Joule assumed 62° Fahr.; but I note that Rankine followed the custom of scientific men generally, and made the temperature of maximum density the standard. French writers formerly accepted an approximation to the old British standard, taking their figure on the metric scale at 15° C.; but I am inclined to think that they now agree upon the more universal standard, as do the Germans. It is of less consequence that either the one or the other should be accepted universally than that all should agree upon one or another.

The whole tendency of recent scientific movement in this matter is, I think, toward the adoption as the universal standard temperature that of the maximum density of water, when measuring the thermal unit. The principal British authority on this subject is Rankine, who says: "The thermal unit employed in Britain is—

"*The quantity of heat which corresponds to an interval of one degree of Fahrenheit's scale in the temperature of one pound of pure liquid water, at and near its temperature of greatest density* (39°.1 *Fahr.*)."

He goes on to say that, in France, the *calorie* is measured in centigrade degrees at the temperature of maximum density. He wrote nearly thirty-five years ago; but I think he still represents the truest scientific feeling of the acknowledged leaders of science.

Professor Peabody makes a good point in his suggestion that the actual engine should be compared with its representative ideal case to exhibit its real relation of merit. To say that an engine has an efficiency of 0.183 gives less idea of its merit, as a construction, than the coördinate statement that it has 0.74 of the value of an ideal engine subject to none of those wastes which characterize, in varying degree, all constructions produced by the hand of man. The first fraction measures its effectiveness in the conversion of heat into work; the second its value as a solution of the engineer's problem in the production of a good piece of apparatus for such work. The first shows what remains to be done in further reduction of all wastes; the last indicates what is the amount of the extra-thermodynamic waste in the machine itself, apart from those inevitable, so far as known, in all heat-engines working through the same range of temperature.

The one is a measure of the imperfection of the methods in use for energy-conversion; the other shows the magnitude of the imperfection of the machine adopted for carrying into effect that system.

The discrepancy noted in the paragraph in which appear equations 4 and 5 seems to come of the fact that one result is obtained on the assumption that, in a condensing engine, the feed water is supplied at the temperature of the boiler steam; while the other takes this temperature of supply at more nearly the actual figure. For non-condensing engines, the former assumption is more nearly true in best practice; it cannot, of course, be true with condensing engines, even approximately. This does not seem to illustrate the point there made, and I do not think that writers known as authoritative are accustomed to make computations on that basis; although it is perfectly true that the best-known writers do use the computation of steam consumption in illustration of their work, and—its limitation being understood—very properly and usefully, as it seems to me.

During the past thirty years, the fact that there are other physical as well as thermodynamic phenomena to be observed in the heat-engines has become so well known that I do not think there is much likelihood that engineers will hereafter be misled by the unfortunate attempts of earlier writers to treat the engine as a purely thermodynamic machine. With our present familiarity with this later fact and the extent of the losses produced by it, the designing engineer may apply his theory, suitably held in check by experience and direct experiment upon familiar cases, with confidence that, at least, he will not be led far astray; although just how far he may expect to profit by it remains something of a question, in face of the fact that practical experience usually leads and checks our theory. But it does afford valuable information and useful suggestion when applied to the comparison of the real with the ideal, giving us a perfect measure of the extent of remaining defects and their distribution, with even, sometimes, a hint of the proper remedy.

Mr. William Kent.—Prof. Peabody states that it has been frequently urged on the attention of engineers that the British thermal unit should be used in stating the performance of engines. The fact that it has been so urged for many years, but that engineers do not adopt it, except in a very few cases, is evidence that there is no long-felt want for its use in general practice. The old custom of expressing the performance of an engine by the number of pounds of steam it uses per hour, is too well established and too simple and convenient ever to be displaced by the thermal unit, no matter how much urging may be done in favor of the latter. The objection to the use of the pound of steam as a unit—viz., that its value depends upon its pressure and quality—is not a serious one in practice, since the difference between the number of thermal units in a pound of steam at 200 lbs. gauge pressure, and one at 80 lbs., is less than 2%. If calorimeter tests are made of the percentage of the moisture in the steam, the proper correction of the number of pounds used can be made if desired. Suppose two condensing engines are tested, one with steam at 200 lbs. and the other with steam at 80 lbs.; that it is found that the steam as it leaves the boiler is dry, and that the first uses 15 lbs. of steam per hour, and the second 20 lbs. These last figures are determined simply by weighing the water fed into the boiler during the test, and no other calculation is necessary than dividing the water per hour by the indicated horse-power. The results are understood by engineers wherever the English language is spoken, and are easily compared with results of hundreds of tests that have been published.

If we wish to express the result in thermal units, we must find the additional data of the temperature due to the back-pressure in the engine, or to the pressure in the condenser, and of the quantity of steam used by the jackets, and must take from a steam table the figures corresponding to the best units and apply them in a formula. Suppose, in the cases in hand, there are no jackets, and the temperature due the pressure in the condenser is 120° Fahr. The heat which has to be supplied by the boiler to each pound of steam used is in thermal units the difference between the total heat of steam of the pressure used, and the total heat of the water fed into the boiler. In the case of the engine using steam at 200 lbs. this difference is 1,112 heat units, and in the case of the other engine it is 1,092.5 heat units, the difference between these figures being less than 2%. Multiplying these figures respectively by 15 and by 20, the number of pounds used per hour, and dividing by 60 to reduce the result to minutes, we have for the engine using 15 lbs. of 200-lb. steam, 278 B. T. U. per minute, and for the engine using 20 lbs. of 80-lb.

TABLE III.

Speed.	Areas Exposed per Ton (2,000 lbs.) in Square Feet.											
	5.	4.	3.	2.5	2.	1.5	1.	0.75	0.5	0.25	0.2	0.1
	Total Resistances in Horse-power per Ton.											
1	0.01698	0.01657	0.01617	0.01596	0.01577	0.01556	0.01537	0.01526	0 01519	0.01507	0.01505	0.01497
2	0.03867	0.03708	0.03537	0.03468	0.03388	0.033097	0.03229	0.031894	0.03149	0.0311	0.031	0 030856
3	0.0652	0.0616	0.0576	0.0572	0.0544	0.052	0.0508	0.0496	0.0494	0 048	0.04792	0.04756
4	0.0906	0.0846	0.0786	0.0756	0.0726	0.0696	0.0666	0.0651	0.0636	0.0621	0.0618	0.0612
5	0.1325	0.12236	0.1117	0.10746	0.10257	0.09725	0.092568	0.08991	0.0876	0.085	0.08448	0.083657
10	0.3815	0.34125	0.30526	0.28153	0.26153	0.241273	0.22154	0.21131	0 20155	0.192	0.18928	0 1855
15	0.7444	0.6564	0.5644	0.52	0.4764	0.4308	0.3864	0.364	0.3412	0.3188	0.3104	0.3054
20	1.2259	1.066	0.906	0.826	0.7462	0.66625	0.563	0.5463	0.50635	0.466	0.45838	0.4424
25	1.8215	1.57176	1.3186	1.1888	1.07226	0.9457	0.8225	0.76	0.7	0.635	0.6227	0.5977
30	2.536	2.176	1.856	1.636	1.456	1.299	1.096	1.017	0.916	0.83	0.808	0.772
40	4.04	3.44	2.84	2.64	2.24	1.94	1.64	1.49	1.34	1.19	1.16	1.1
50	6.2	5.5328	4.3258	3.87	3.5378	2.94	2.54	2.11	1.875	1.7755	1.742	1.6425
60	9.248	7.808	6.368	5.648	4.928	4.208	3.488	3.128	2.768	2.409	2.336	2.192
70	12.4	10.444	8.579	7.5066	6.5275	5.595	4.579	4 10678	3 59	3.10895	3.	2.80682
80	16.0176	13.4616	10 9056	9.6276	8.35	7.0716	5.79	5.1546	4.5156	3.8766	3.7488	3.4932
90	20.036	16.416	13.632	11.80	10.416	8 784	7.176	6.36	5.568	4.746	4.584	4.26
100	24.6316	20.64	16.6	14.66	12.6616	10.64	8.6716	7.6608	6.7166	5.6250	5.48	4.6806
110	29.7	24.8464	19.924	17.58	14.9774	12.7162	10.2429	9.1123	7.9256	6.71556	6.4753	5.992
120	35.3	29.44	23.04	20.80	17.92	15.04	12.16	9.72	9.28	7.84	7.552	6.976

pound of water from 62° to 63° Fahr. Both Rankine and Clark define it as the amount required to raise the temperature one degree from 39.1°.

*Prof. C. H. Peabody.**—While it may be admitted that it makes but little difference, in the case quoted by Mr. Kent, whether the comparison is made in terms of steam per horse-power per hour, or in thermal units, it must be noted that he has chosen a case that is very favorable to his view of the question. Let us consider a comparison of tests made on the triple engine in the laboratory of the Massachusetts Institute of Technology, with and without steam in the jackets on the cylinders. The steam pressure, revolutions per minute, and cut-off of the high-pressure cylinder were very nearly the same, and the tests are strictly comparable. With steam at boiler pressure in the jackets, the engine used 13.74 lbs. of steam per horse-power per hour, and 237 B. T. U. per minute; without steam in the jackets the engine used 15.81 lbs. of steam per horse-power per hour, and 285 B. T. U. per horse-power per minute. The ratios of these quotations are:

$$13.74 : 15.81 = 1 : 1.15$$
$$237 : 285 = 1 : 1.20,$$

so that the gain from the use of the jackets appears to be 15% when the steam consumptions are compared, while a comparison of the thermal units shows the gain actually to be 20%. Suppose, for sake of the argument, that the engine used the same quantity of steam per horse-power per hour; then the heat consumption of the engine without jackets would have been 247 B. T. U. per horse-power per minute, which is about 4% larger than the consumption found with steam in the jackets. I think it must now be evident that the only way of presenting clearly the action of steam-jackets is by the use of the definite, simple, and practical thermal unit. Examples could be given to show that the same statement is true if moist or if superheated steam is used instead of dry steam. The calculations involved are no more abstruse or difficult than the determination of the equivalent evaporation from and at 212° Fahr., for a boiler test, a matter which is well understood by all engineers. It will be found also that the thermal unit is very convenient in that it allows the statement

* Author's closure.

of engine performance with all desirable accuracy without the use of fractions.

The proper pressure to be used as the lower limit may possibly be an open question; for example, whether it should be the back-pressure in the cylinder, the pressure in the exhaust pipe, or the pressure in the condenser. In a well-designed engine these will all be nearly the same. My own preference (which, however, I do not wish to urge) is for the pressure in the exhaust pipe, because the feed water for either a condensing or a non-condensing engine may be brought to the temperature corresponding to that pressure by aid of a coil heater. Further, the comparison of engines ought not to depend on the properties or the arrangement of the boiler nor of the piping connecting the engine and the boiler. It has been my experience that the drain from steam-jackets has the temperature due to the pressure of the steam in the jackets, at the place where the jacket drain naturally mingles with the feed water from the hot well or coil heater; and, further, that the loss of temperature from the engine to the boiler need not exceed 10°, or 12° Fahr., even though there is considerable space from the engine to the boiler. In conclusion, it may be claimed that the statement of engine performance in thermal units is thoroughly definite, scientific, and practical.

In answer to the questions concerning the definition of the thermal unit, I will say that I have no authority to quote except the experiments of Rowland on the specific heat of water and on the mechanical equivalent of heat, and the latest experiments of Joule on the mechanical equivalent of heat. The commonly accepted mechanical equivalent, or the foot-pounds necessary to raise the temperature of water one degree, is 772; the later experiments by Joule, however, give 778 at about 62° Fahr. Rowland's experiments give the same quantity—namely, 778 at 62° Fahr., and show that the equivalent varies in a marked manner with the temperature; for example—at 39° Fahr., the equivalent is probably about 781, but is not well determined, for Rowland in his report begins to give the work units required to increase the temperature one degree at 5° C. or 41° Fahr. This appears to preclude the use of freezing point as the standard temperature, and also of the somewhat indefinite temperature of maximum density—*i.e.*, 39.1° Fahr. If, however, we choose 62° Fahr. for the standard temperature, we may use directly the work of both Joule and

Rowland for the mechanical equivalent of heat, and we fall into line with the most recent work in physics, in which 15° C. is chosen for the standard temperature, and with the notable work of the scientists who fixed the standards of weights and measures for the British Empire, and at the same time for the English world of which we are a part.

CCCCLXXXV.*

THE ELECTRIC RAILWAY AS APPLIED TO STEAM ROADS.

BY B. J. DASHIELL, JR., PHILADELPHIA, PA.

(Junior Member of the Society.)

IT is the intention of the writer to give some information, for record before this Society, which will throw light upon the question of high-speed train resistance, deduced from tabulated experiments at such speeds.

The first experiments of note at extremely high railroad speed were made by the Electro-Automatic Transit Co., of Baltimore City, in the year 1889, at Laurel, Md., on their circular road, with a $2\frac{1}{2}$-ton car. Some of these experiments were publicly reported by Mr. O. T. Crosby, in his paper entitled "Report of High-speed Electric Railway Work." † Mr. David G. Weems, the originator, contemplated having an automatic mail and express service from one city to another, controlling the cars from central stations located on the line of the road, and which was to be a very complete "block system" in its way. In the early spring of 1888 my services were required by this company, and designs and experimental circular tracks were made and built under my direct supervision. I may here state, for the benefit of some not knowing the circumstances, that the line of road was built for a *limited sum* of money, and that in some of the experiments, owing to some "fixed ideas," the motors were designed for too high a speed. Both of these obstacles gave us quite a little trouble on the start.

The power house consisted of a 30 foot x 40 foot x 12 foot wood building, costing about $600 to build. It was divided into three rooms — boiler-room, engine and dynamo-room, and office. The plant consisted of 125 H.P. in two (2) vertical boilers, 100 H.P. Ball high-speed, automatic cut-off engine, run-

* Presented at the San Francisco meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions.*

† *Trans. Amer. Inst. Elect. Engrs.*, Feb. 24, 1891.

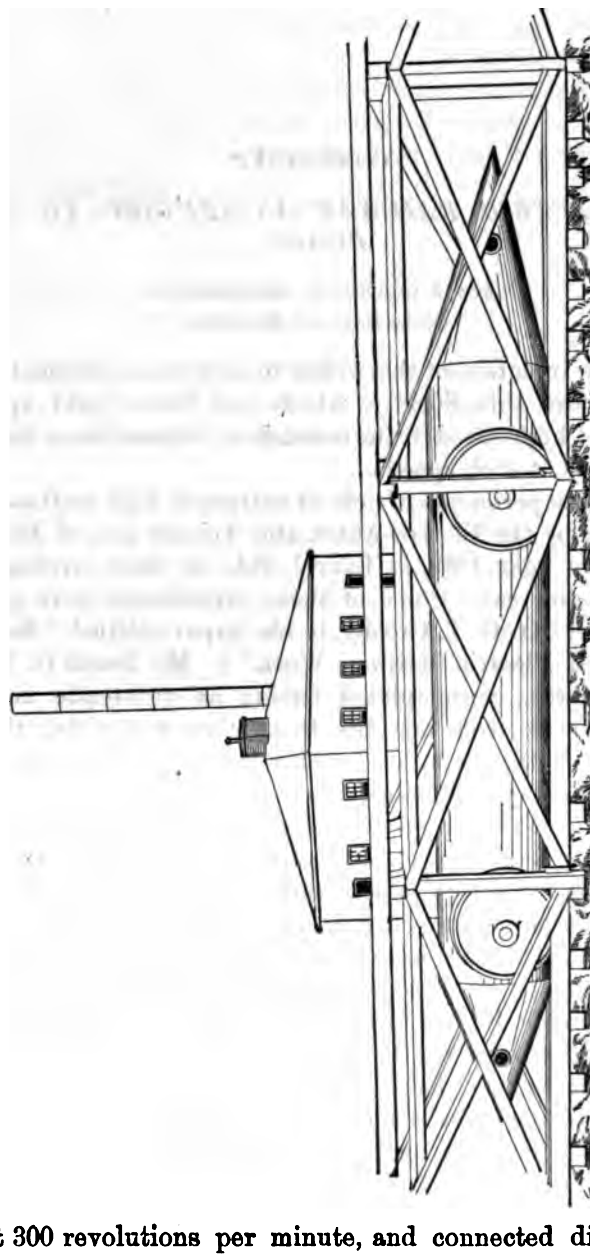

Fig. 72.

ning at 300 revolutions per minute, and connected direct by a double belt with a No. 20 Edison generator of 50,000 watts capacity, which supplied the travelling car with energy.

The motor car (Figs. 72, 73) had a wheel base of 9 feet; 4 driv-

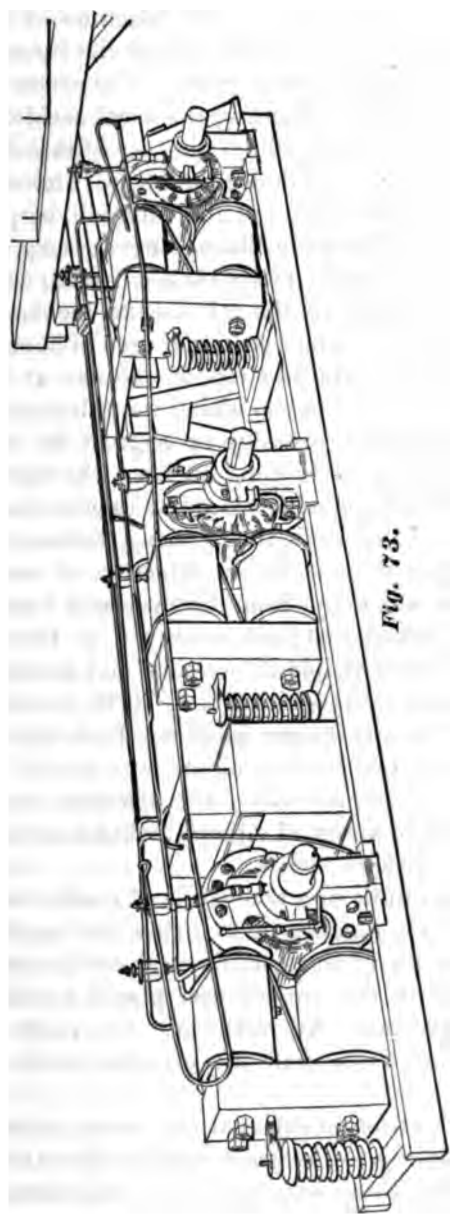

Fig. 73.

ing wheels, 28 inches diameter, of cast-steel plate web, 1¼-inch flange, and 2¼-inch tread; gauge of track, 28 inches; two (2) axles of 3-inch machinery steel, each carrying a gearless motor of tests on a simple engine, as the cut-off is lengthened. The consumption for the 5th test with a cut-off at about one-third stroke is probably a minimum for the engine, and is a creditable performance for an engine of its size and type.

(5) The efficiency improves with the lengthening of the cut-off in proportion as the consumption decreases.

(6) The efficiency for a non-conducting engine with incomplete expansion is nearly as good as for such an engine with complete expansion for the first test, since the expansion is carried nearly down to the back-pressure. The efficiency for such an engine falls off as the expansion is less complete, *i.e.*, as the cut-off is lengthened. The failure to realize the excellent efficiency shown for such an engine in the first and second tests is due to the action of the walls of the cylinder.

(7) As the cut-off is lengthened and the task set before the engine becomes easier, the ratio of the actual efficiency to that of a non-conducting engine with incomplete expansion improves. For the 5th test this ratio becomes 0.65. The difference between this result and unity may be charged to radiation and to the influence of the walls of the cylinder. Anything that can mitigate the influence of the cylinder walls (such as superheating or steam-jacketing) may lessen the steam consumption by allowing the use of more expansion, *i.e.*, of a cycle that would give a better efficiency with a non-conducting cylinder.

There is a certain fascination in trying to predict the performance of future engines, or at least to set the limit for which we may strive. As a matter of curiosity the following table has been computed for a non-conducting engine with complete expansion, together with a comparison with Carnot's cycle for the same ranges:

TABLE III.

Initial pressure by the gauge above the atmosphere.	Efficiency of Carnot's cycle.	Non-conducting engine with complete expansion.			Hypothetical limit of steam-engine performance.
		Efficiency.	B. T. U. per horse-power per minute.	Steam per horse-power per hour.	
150	0.302	0.272	156	8.63	11.5
200	0.320	0.288	147	7.91	10.5
300	0.347	0.306	135	7.22	9.6

In the calculation of this table the back-pressure is assumed to be 1.5 lbs. absolute, and the steam supplied is assumed to be dry and saturated.

There is no known way of calculating the steam consumption of an actual engine from that of a non-conducting engine; and for an engine of the type used when high economy is expected, there is the added difficulty that part of the steam is condensed in the jackets and does not pass through the cylinders. A hypothetical limit of steam-engine economy has been found by taking $\frac{4}{3}$ of the steam consumption of a non-conducting engine, for the steam consumption of the actual limit. The quantities thus found are probably too small as the ratio, taken from the table on page 349, is for an engine with a less range of pressure than found in Table III.

DISCUSSION.

Prof. R. H. Thurston.—I presume that no one will dispute the main proposition of this paper: that the best scientific measure of the efficiency of the heat-engine is its consumption of heat measured in thermal units per unit of power developed. This affords a common basis of comparison of all engines transforming heat into work. It is true that the use of the weight of steam as a unit is subject to the objection that it may lead to error; though it should, I think, be said that this only can happen when it is forgotten that the same variations of steam-pressures and feed-water temperatures which affect the efficiency of the engine also affect its value. The point to be noted is that this latter unit is a variable magnitude, while the thermal unit is a constant. Exact scientific comparisons of results of experiment are only possible when all quantities compared can be reduced to a common unit, a recognized fixed standard.

I observe that the writer of this paper takes the B. T. U. as the measure of the specific heat of water at 62° Fahr., while, I think, the great majority of authorities, recognized as such throughout the world, assume the standard temperature as that of freezing or of the maximum density of water. The old British standard temperature here assumed is, if I am not very greatly mistaken, not only not retaining its place outside Great Britain, but is, with all other relics of the old system of weights and

measures, going out even there. It would, however, be interesting to ascertain precisely what standing it has by comparing the various principal authorities. Joule assumed 62° Fahr.; but I note that Rankine followed the custom of scientific men generally, and made the temperature of maximum density the standard. French writers formerly accepted an approximation to the old British standard, taking their figure on the metric scale at 15° C.; but I am inclined to think that they now agree upon the more universal standard, as do the Germans. It is of less consequence that either the one or the other should be accepted universally than that all should agree upon one or another.

The whole tendency of recent scientific movement in this matter is, I think, toward the adoption as the universal standard temperature that of the maximum density of water, when measuring the thermal unit. The principal British authority on this subject is Rankine, who says: "The thermal unit employed in Britain is—

"*The quantity of heat which corresponds to an interval of one degree of Fahrenheit's scale in the temperature of one pound of pure liquid water, at and near its temperature of greatest density* (39°.1 *Fahr.*)."

He goes on to say that, in France, the *calorie* is measured in centigrade degrees at the temperature of maximum density. He wrote nearly thirty-five years ago; but I think he still represents the truest scientific feeling of the acknowledged leaders of science.

Professor Peabody makes a good point in his suggestion that the actual engine should be compared with its representative ideal case to exhibit its real relation of merit. To say that an engine has an efficiency of 0.183 gives less idea of its merit, as a construction, than the coördinate statement that it has 0.74 of the value of an ideal engine subject to none of those wastes which characterize, in varying degree, all constructions produced by the hand of man. The first fraction measures its effectiveness in the conversion of heat into work; the second its value as a solution of the engineer's problem in the production of a good piece of apparatus for such work. The first shows what remains to be done in further reduction of all wastes; the last indicates what is the amount of the extra-thermodynamic waste in the machine itself, apart from those inevitable, so far as known, in all heat-engines working through the same range of temperature.

The one is a measure of the imperfection of the methods in use for energy-conversion; the other shows the magnitude of the imperfection of the machine adopted for carrying into effect that system.

The discrepancy noted in the paragraph in which appear equations 4 and 5 seems to come of the fact that one result is obtained on the assumption that, in a condensing engine, the feed water is supplied at the temperature of the boiler steam; while the other takes this temperature of supply at more nearly the actual figure. For non-condensing engines, the former assumption is more nearly true in best practice; it cannot, of course, be true with condensing engines, even approximately. This does not seem to illustrate the point there made, and I do not think that writers known as authoritative are accustomed to make computations on that basis; although it is perfectly true that the best-known writers do use the computation of steam consumption in illustration of their work, and—its limitation being understood—very properly and usefully, as it seems to me.

During the past thirty years, the fact that there are other physical as well as thermodynamic phenomena to be observed in the heat-engines has become so well known that I do not think there is much likelihood that engineers will hereafter be misled by the unfortunate attempts of earlier writers to treat the engine as a purely thermodynamic machine. With our present familiarity with this later fact and the extent of the losses produced by it, the designing engineer may apply his theory, suitably held in check by experience and direct experiment upon familiar cases, with confidence that, at least, he will not be led far astray; although just how far he may expect to profit by it remains something of a question, in face of the fact that practical experience usually leads and checks our theory. But it does afford valuable information and useful suggestion when applied to the comparison of the real with the ideal, giving us a perfect measure of the extent of remaining defects and their distribution, with even, sometimes, a hint of the proper remedy.

Mr. William Kent.—Prof. Peabody states that it has been frequently urged on the attention of engineers that the British thermal unit should be used in stating the performance of engines. The fact that it has been so urged for many years, but that engineers do not adopt it, except in a very few cases, is evidence that there is no long-felt want for its use in general practice. The old custom of expressing the performance of an engine by the number of pounds of steam it uses per hour, is too well established and too simple and convenient ever to be displaced by the thermal unit, no matter how much urging may be done in favor of the latter. The objection to the use of the pound of steam as a unit—viz., that its value depends upon its pressure and quality—is not a serious one in practice, since the difference between the number of thermal units in a pound of steam at 200 lbs. gauge pressure, and one at 80 lbs., is less than 2%. If calorimeter tests are made of the percentage of the moisture in the steam, the proper correction of the number of pounds used can be made if desired. Suppose two condensing engines are tested, one with steam at 200 lbs. and the other with steam at 80 lbs.; that it is found that the steam as it leaves the boiler is dry, and that the first uses 15 lbs. of steam per hour, and the second 20 lbs. These last figures are determined simply by weighing the water fed into the boiler during the test, and no other calculation is necessary than dividing the water per hour by the indicated horse-power. The results are understood by engineers wherever the English language is spoken, and are easily compared with results of hundreds of tests that have been published.

If we wish to express the result in thermal units, we must find the additional data of the temperature due to the back-pressure in the engine, or to the pressure in the condenser, and of the quantity of steam used by the jackets, and must take from a steam table the figures corresponding to the best units and apply them in a formula. Suppose, in the cases in hand, there are no jackets, and the temperature due the pressure in the condenser is 120° Fahr. The heat which has to be supplied by the boiler to each pound of steam used is in thermal units the difference between the total heat of steam of the pressure used, and the total heat of the water fed into the boiler. In the case of the engine using steam at 200 lbs. this difference is 1,112 heat units, and in the case of the other engine it is 1,092.5 heat units, the difference between these figures being less than 2%. Multiplying these figures respectively by 15 and by 20, the number of pounds used per hour, and dividing by 60 to reduce the result to minutes, we have for the engine using 15 lbs. of 200-lb. steam, 278 B. T. U. per minute, and for the engine using 20 lbs. of 80-lb.

steam, 364.2 B. T. U. per minute. The trouble, slight as it may appear to a college professor, of looking up steam tables and working a formula, the unfamiliar appearance of the numerical result and of the terms used, will no doubt lead the practical engineer to continue to state his results in such simple terms as 15 and 20 lbs., rather than in such as 278 and 364 B. T. U.

One more serious objection, however, to the use by the practical engineer of the thermal unit as a standard of performance is the difference of opinion which may arise as to the correct method of calculation. For instance, Prof. Peabody in his formula uses for q the heat of water at the temperature due the back-pressure. Why should he not take instead the heat units due the temperature in the hot well, or in case of the non-condensing engine, the heat units in water at 212°, or the temperature of the feed water? We supply to the engine steam of a certain pressure. It consumes, usefully or wastefully, all the difference between the heat in that steam and such portion of the heat as we are able to recover from it and put back in the boiler. In case of a throttled exhaust, causing excessive back-pressure in the engine, there may be a great error in assuming that q should be taken as the heat units in water of the temperature due to that back-pressure.

Again, in the case given by Professor Peabody, of the engine in the Massachusetts Institute of Technology, it appears that he may have made a mistake in figuring its consumption in thermal units per minute. He charges against the engine for the jacket steam merely the latent heat of that steam, 858.3 heat units, which would indicate that that was all the heat given up by a pound of steam in its circuit from the boiler through the steam-jacket, and back to the boiler in the shape of hot water; that is, that the water drained from the jackets was pumped back into the boiler at the temperature of the steam in the boiler, which is scarcely possible in practice.

I have no objection whatever to the use of the thermal unit in scientific studies of the steam-engine; in fact, it is a necessity in such studies; but I do object to its use in ordinary practice, as more liable to lead to confusion and error than the old and simple custom of using pounds per hour.

I would like to know where Professor Peabody gets his definition of the heat unit as exactly the heat required to raise one

The writer has taken a hand, during the past few months, with the inventor, in the development of this anti-frictional material, and is fully prepared to formulate and present the following well-considered conclusions.

With fibre-graphite bearings properly prepared and fitted to the supports and journals of machinery, the cost of oil together with all the appliances necessary to store, retain, convey and conduct the same to the bearings is entirely saved.

No wearing of the shaft journals has as yet been observed, and very little abrasion of the bearings has been reported by those who have used them.

During the first revolution of the shaft its concavities are filled with graphitic particles which are worn off the bearing by its rubbing action upon it. When the surface of the shaft is completely covered and evened up upon its whole exterior, the sliding will be conducted thereafter wholly upon the newly-formed graphitic surfaces, to which even the disengaged particles of the bearing will assist in lubrication; friction will in consequence be reduced to a minimum, and the shaft journal will be protected from subsequent wearing.

If we start with roundness of shaft and if we grant a certain roughness of surface, the journal after running a short time acquires smoothness by borrowing material from the bearing upon which it runs, and simultaneously fits itself the closer thereto, for its better support and running, after which the motion is all conducted upon the original and upon the acquired graphitic surfaces, with a measure of friction due to the natural lubricity of the graphite used, and without added lubricant of any kind or any attention whatever.

The labor of cleaning and oiling and the cost of waste and wiping are saved, the soil-greasing of fabrics, machinery and buildings cannot happen, and the serious danger of fire from lubricating oils, and the spontaneous combustion of them with discarded waste, are wholly removed.

The making of these bearings, much of it having been done by rude and hastily designed tools and appliances, has amply proven that the cost of them need not be *above* and may be *below* the cost of metal ones which they replace, leaving a very fair margin of profit to those who, with superior machinery and appointments, engage to manufacture them, while the user is spared the cost and nuisance of oil and oiling, and in addition thereto

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A peculiarity of graphite bearings, which may have great value in the arts over the usual materials now employed, where sliding is involved, is that its friction at starting of motion is about the same as its moving friction. This property will enable the engineer to reduce the gross allowances for power to a minimum, when arranging for and providing the machinery of transmission, in cases where graphite replaces the usual metals with lubricants for bearings.

The great importance of this invention to the manufacturing world must be apparent without further reference. Every observing mind will see, although the boundary lines of its extent in any direction are not within sight, that its practical applications, numerically stated, are as multitudinous as the details of existing machines, and any list of their frictional parts would make a catalogue too voluminous for reading.

DISCUSSION.

*Mr. H. W. Harkness.**—Is the graphite pulverized which is used in these bearings?

Mr. C. N. Trump.—It is very finely pulverized and mixed with wood pulp.

Mr. Harkness.—You have to select the very best graphite, I take it?

Mr. Trump.—The graphite should be purified from all gritty matter.

Mr. Harkness.—In doing so, do you not have more or less clay?

Mr. Trump.—There does not seem to be any difficulty of that kind.

*Mr. N. S. Keith.**—This matter is of exceeding interest to me. I am a manufacturer on this coast of dynamos and electric motors. In making dynamos and motors, which are all speedy-running, the manufacturers have hitherto had a great deal of difficulty in producing a proper box and a proper lubricant. I have tried a great many experiments in this matter, using brass boxes, babbitt boxes, self-oiling boxes, sight-feed oil cups, etc. We get the best satisfaction from self-oiling arrangements. I

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am led to believe by the paper read that these new bearings are of immense value.

Mr. F. A. Scheffler.—I would like to ask if this is sold by the pound?

Mr. Trump.—I think not; I think it is sold by the piece, depending on the class of construction, in all probability. If it is in order, I will simply say that the graphite and wood pulp in various proportions are mixed in water alone; they are then put into a mould, which is usually made of brass with grooves on the outside, provided with small holes, possibly $\frac{1}{64}$ of an inch in diameter, spaced, I should judge from recollection, from $\frac{1}{4}$ to $\frac{5}{8}$ of an inch apart. The mould is enclosed in a heavy case which will support hydraulic pressure, and the mixed graphite, water, and wood pulp are pumped into the mould with an ordinary hand-pump; then the whole thing is put under an hydraulic press, a follower put in, and the mass is compressed to about $\frac{1}{3}$ of its original bulk, the water flowing out through those little holes and along the grooves in the outside of the brass mould and running off. The piece is then removed from the mould and simply dried in an oven; after drying it is saturated with drying linseed oil and baked so as to be thoroughly hard. You can cut it with ordinary tools, and some of it, having more wood pulp and pressed harder than others, is a great deal stronger than that which has a proportionately smaller quantity of wood pulp in. I have seen some 18 bearings on ordinary 2-inch and other sizes of shafting, running. They had been in use for over a year, running 200 revolutions per minute, 6 to 8 hours a day. You will find occasionally a bright spot where the box was not quite straight perhaps, but there is no destructive wear on the material, under ordinary machine-shop work; and I have seen several samples of pretty difficult work, where there did not appear to be any chance for the box at all. I saw, just before I came away from Nicetown, a shaft about 8 feet long, loaded with 5,000 lbs. of wheel castings; the bearings were $9\frac{1}{2}$ inches long by $2\frac{7}{16}$ inches in diameter, simply a pair of hangers reversed, and the cast iron box supported by two opposite screws which would necessarily, with the weight on them, make the box pinch, and it did pinch pretty hard when it was first started, so that the shaft was sufficiently hot to melt an excellent quality of babbitt metal, say ^{1} x $\frac{1}{4}$ inch in section, at the rate of about $\frac{1}{4}$ inch in 10 seconds; it would melt it right off, just make a continual stream of babbitt metal running from the piece which was held against it. After running this shaft 10 days at about 200 revolutions per minute, there was no appreciable wear on the graphite box other than that it relieved itself from this pinching. Generally the journals of shafting become covered with a thin coating of graphite, and my friend Mr. Cooper assured me that he had never seen the slightest mark of abrasion on any shaft, and he has witnessed its running during about eighteen months.

The President.—Was the shaft running cold?

Mr. Trump.—The shaft was running cold, so that you could bear the hand on it. I saw those boxes taken out after it had been running two days, and there was no appreciable wear.

Mr. Geo. W. Dickie.—I should like to ask if any engineer has had the courage to use this bearing on engine work, either for journals which were rotating or for the reciprocating parts, and if any report has been obtained from its use on engines. It is a very interesting subject, this, and I know it is a subject which touches the engineer pretty closely. If we could construct our large engines and send them out without the necessity of using oil, it would be a great relief, if nothing more. I would like to know if any engineer has attempted to use these bearings on engine work of considerable size, and if there is any report of its use.

Mr. Trump.—At No. 123 North Fifth Street in Philadelphia, they had an engine of about 20-horse power fitted all through with bearings of this material. They also had a dynamo running five arc lights fitted throughout with it, and the shafting, as I said before, has been fitted with it, and that had been running since last June in Philadelphia from 8 to 10 hours a day. The dynamo and some of this shafting had been previously used in Gardiner, Maine, where the inventor lived. In the body of the article there are some certificates of use. The superintendent of the Market Street Traction Road, in Philadelphia, put in some 24 of those bearings for grooved pulleys carrying the cable, two at the crown where the cable came from the driving plant up into the street—a very difficult place for anything to stand. They had had a great deal of difficulty with ordinary bearings. The fibre graphite bearings had been run for some six months, when the superintendent gave a certificate that they required no attention. Some of them were in places where they were covered with water running in from the street into the conduit carrying the cable, and he states in his certificate that they had never had any extra attention, and were apparently in as good order as when they were put in. Mr. Cooper's experience with that is all that I have to make any reference to, and he gives that in his paper.

*Mr. Geo. Cummings.** —I would ask the speaker if he knows of a use of graphite in thin pencils inserted in brass bushings. It was presented to engineers some fifteen years ago. I applied the method to a pulley which gave us a great deal of trouble by reason of its high speed and the strain of the belt, together with its location in flying dust. With an ordinary lubricator the shaft wore so, that it had to be renewed every six months. With the inserted sticks of graphite it ran for three years, but it gave out all at once and the shaft began to heat again. The brass bushing was bored $\frac{1}{32}$ of an inch larger than the shaft, and then drilled with holes for the graphite pencils so that they stood about $\frac{1}{8}$ of an inch apart in the clear. The graphite projected inside the bush about $\frac{1}{64}$ of an inch into the bore. I put a large sign over the pulley with the date of the beginning of the trial, and a warning against interfering with it. After one year's running I took it apart, and could detect no wear on the shaft or bush. There was some loose graphite powder on the bushing, but it never heated or troubled us. The remaining two years it ran without inspection, and I do not know why it should suddenly have heated up and cut the bushing to pieces. Our experience with it was so satisfactory, that at the end of the year we discussed the question of putting graphite into all bearings of one of the iron works in this city, but we found the royalty was commercially prohibitory.

I know also of a similar experience at a mechanical mill in this city, where a pulley similarly treated has been in operation for two years, and was still running in good order when I last saw it. Mr. Stevens, then Superintending Engineer of the Pacific Railroad at Sacramento, told me that he had tried this arrangement in connecting-rods having a reciprocating motion, but it was not found to answer, because the reciprocating motion pounded the graphite out. Do you know how that new compound will act in the connecting-rod of an engine under similar strain?

* Of San Francisco; by invitation.

Mr. Trump.—Only, as I say, from seeing that engine fitted up in that place in Philadelphia.

Mr. Keith.—The author states, I think, that the dynamo brushes were made of this material, and that they were running successfully.

Mr. Trump.—Yes, sir.

Mr. Keith.—That is next in importance to the electrical engineer to that of the bearings themselves.

Mr. Trump.—I might say, in answer to that question in reference to the use of them, that I saw on this dynamo that I speak of, which was running five arc lights, a pencil $\frac{1}{4}$ inch square, carrying the whole current for those five lights, and there was not a particle of roughness on the commutator, and simply a very little wear on the pencil, scarcely appreciable, so that the brush required but little adjustment. I don't know that I could give any better evidence of its conductivity. I am not an electrician, and consequently could not meet all the points that may be raised, but am informed by the manufacturers that the conductivity is 75 amperes per square inch of cross-section, without heating above 50° C.

Mr. Keith.—That is certainly better than a copper brush. Carrying, I presume, 10 amperes of current from the commutator, a copper brush would have to have a greater bearing than $\frac{1}{16}$ of a square inch, which is the amount of bearing on the commutator of a brush $\frac{1}{4}$ inch square.

Mr. G. W. Spiers.—I should think a series of experiments based upon the pressure per square inch and velocity would be exceedingly useful. If Mr. Cooper has not made such experiments, it seems to me it would be a very desirable thing for him to do, for the reason that it will give mechanics a good idea as to what the bearing will do under different conditions. What we want to know is the pressure per square inch and the velocities under these varying pressures, up to the heating or destructive point.

Mr. Trump.—I had from Mr. Cooper, since I saw him, a letter, in which he gives the size of the bearings of that loaded (5,000 lb.) shaft of which I spoke; and he also says a tubular bearing, in which a shaft ran 3,300 revolutions for five months, ten hours per day, showed wear upon a few spots only, which were as bright and smooth as glass. The bearing had not been machined at all; it was simply as it came from the mould, and the circular part of it was not true, as it would have been if we had bored it. The shaft simply touched it in spots, and those spots would be bright.

*Mr. John H. Cooper.**—The use of this material for the heavier class of machinery bearings and for working steam-engines, has not been attempted on any scale from which definite results can be given. The inventor is now conducting experiments with boxes for railroad car axles, loaded to from 300 to 400 lbs. per square inch of bearing surface, covering the usual practice, which promise commercial success fairly well, either with or without added lubricants.

The use of lubricants, which may be applied in these cases for certain reasons, will reduce friction and heat; on the other hand, should oiling be neglected, the box would not be injured, where destruction of the same might follow from the use of metallic ones. So that at the same price for boxes and for oil, there will be again in favor of fibre graphite, which may further be emphasized by the reduced wear of the axles, which all experiments prove.

A substance which would at once endure the usual excessive pressures incident to the employment of heavy machinery, and at the same time permit its use without lubrication and without materially increasing the friction, as we now find it, would, of course, be a welcome addition to our stock of materials and possess great value, but such is not now known among the metals. Compressed fibre in such proportions as to give strength, if mixed with purified graphite to secure the most perfect lubricant material, must in some combination under excessive pressure approach nearest to fulfilling the conditions stated above. Such a compound is clearly within the cover of Mr. Holmes' claims, but has not received thorough treatment and test up to this writing.

Experiments in all the departments of application cannot be made at once by an inventor, while those which can be readily tried, adapted to his locality, to the most accessible machinery, would claim precedence. Also such bearings as would be needed in largest number of similar pieces would naturally be fitted first, as the preparation of fixtures and moulds is attended with some expense.

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* Author's Closure.

The fitting of bearings to mill spindles, as well as to the machinery of preparation for the manufacture of textile fabrics, offers perhaps the largest field for the use of these bearings. To this class of machinery, situated as he was, the inventor first turned his attention. Experience with this type of machinery abundantly proves that the incidental expenses and the consequences of the use of oil are immeasurably greater than the original cost of the same. A single "hot box," or the spontaneous combustion of neglected oily "waste," may start a fire which cannot be arrested till the mill and all its contents are consumed. As combustion cannot be created by the use of this material for bearings, and there being no oil present to soil fabrics and carry fire, a wide field is at once opened for the supply of bearings which promise such superior advantages. I take pleasure in referring you to the report of the Boston Committee, given in the body of this paper, which clinches these statements.

A strong argument in favor of the extended use of this material, even in easy places under least pressure, is derived from the fact that no metal will continue to slide or rotate upon another metal without some intervening lubricant to prevent destruction of their rubbing surfaces, and that that damage will result as certainly under the lightest pressures. To repeat an experiment in proof of this assertion, I have placed the Rabbeth form of mill spindle, fitted with their usual metallic bearings for oil (after wiping the bearings clean), into a frame with 24 similar spindles, the same in every respect, except they are fitted with fibre graphite bearings, and I drive them all by the same running cord, under the same tension, and at the same speed; the result comes quickly: the metal bearing will not permit the spindle to run 10 minutes before heating and squealing begin and which increasingly continue; a few drops of oil—the usual remedy—of course, stops all this and permits quiet running for hours thereafter. On the other hand, the graphite spindles, without added lubrication or least attention, have been running for a year, most of the time 8 hours daily, at 9,000 revolutions per minute, with promise on inspection of continued satisfactory running.

To one important requirement I wish to refer, that of giving this material proper backing and end support, when used for shaft and other bearings, the same as already done for "Babbitt" and other of the soft and yielding anti-friction metals.

The nature of this material, however, forbids any expanding process, by hammering, for filling its prepared cavities tightly. The needed support must be gained by a cement filling where the fit is to a casting, and in case of cylindrical bearings fitted to bored receptacles, the end support may be secured by flanged nuts tightened against the ends of the bearings.

With the addition of elastic intermedia to accommodate slight changes of alignment, or of dimensions due to changes of temperature, which will also permit of easy interchangeability, these bearings may be said to be fully equipped for use and to possess ready adaptability to almost every machine.

CCCCLXXXVII.*

AN EXPERIMENT WITH ALUMINUM.

BY W. WALLACE CHRISTIE, NEW YORK CITY.
(Member of the Society.)

ABOUT two years ago the writer was detailed by Mr. F. W. Snow, superintendent of the Ramapo Iron Works, to make a series of mixtures for cast metal and to test the castings, ready recorded.

Mixture No. 1:

Wrought-iron turnings	10 lbs.
Cast-iron turnings	10 lbs.
Steel-rail chips	10 lbs.
Ferro-silicate of iron and aluminum	2 lbs.

Mixture No. 2:

Wrought-iron turnings	10 lbs.
Cast-iron turnings	5 lbs.
Steel-rail chips	15 lbs.
Ferro-silicate of iron and aluminum	2 lbs.

The melting was done by a well-known brass founding firm in their brass furnace. In order to melt the mixtures very high temperature was required on account of the wrought iron, which requires 3,000°; so the crucible was covered with a carbon lid and coal heaped upon it. Even then about three hours' time was required to melt it, and after being melted the ferro-silicate of iron and aluminum, which had been left out, was added, and thoroughly stirred through. The castings made were $1\frac{1}{2}$ inches diameter by 14 inches long, and in green sand without any charcoal facing, and after the skin of sand had been removed from the castings they were very smooth and clean.

Mixture No. 1 was very fluid when hot and white, but had to be poured quickly, as it soon cooled.

* Presented at the San Francisco Meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the Transactions.

Mixture No. 2 was not as fluid nor as white as No. 1.

Mixture No. 1 made a very homogeneous casting; No. 2 not nearly so much so, and its fracture duller than No. 1, which latter was very bright.

It may also be said that pieces of both mixtures which have been on my desk since April, 1890, when they were cast, have retained their original brightness, which speaks well for the small percentage of aluminum in them.

Mixture No. 1 could not be touched by a specially tempered cold-chisel, but its edge was entirely destroyed. The piece of mixture No. 2 shows where a tool maker had used an hour's time cutting off but little and during that time the tool required many sharpenings; I believe, five or six. When heated to a high red heat they both crumble when struck with a hammer, but when heated to a dull red heat No. 1 was placed under a steam hammer, and though quite resisting, allowed itself to be flattened to about 1¼ inches thick before crumbling, but gave better results when annealed over one night.

No. 2, when heated in the forge to a dull red heat, could be flattened to about ¾ inch thick.

Having in his possession a piece of No. 1 when doing some laboratory work at Cornell University, the writer had it remelted and cast into the usual shape for tension tests. This piece, though but 8½ inches long, was put in a Fairbanks testing machine, but as it was uncertain as to just how it would act, no extensometer was used for fear of the test piece breaking suddenly. Breaking occurred at a scale reading of 13,860 lbs. The piece broke, however, in the jaws of the machine, and in the larger section of the piece, as there was a flaw in it (cinder flaw). For fear of breaking the jaws of the machine the test ended here. After breaking the smaller section in the impact machine, the area was obtained by a planimeter as .31 square inch, which makes the tensile strength per square inch at the time of breaking 44,710 lbs. This would have been higher, and probably considerably, but for the flaw and untrue grip of the jaws, which caused a combined transverse and torsional strain. The area of smaller section was less than that of the sound portion of larger section, hence its use. When placed on a Heisler impact machine, between supports 6 inches apart, a weight of 25 lbs. falling 1¾ inches was required to break a circular section of .31 square inch. It is not desirable to use it for work requiring finishing, as it is too hard for that, except when done on a grindstone.

The ferro-silicate of iron and aluminum used was an ordinary commercial article, purchased in the open market, and whose composition the writer was unable to learn.

No. 1 is much harder to grind than No. 2, and both present very smooth surfaces, as can be seen by the inspection of the specimens.

DISCUSSION.

Mr. Henry D. Hibbard.—It is unfortunate that the author has not given the composition of the alloys which he used, but in my judgment the mixtures given would have given a low-silicon, white cast iron, if there had been no chemical change in the melting, and this iron would have had somewhat the character which the resulting metals seem to have had. The long time taken to melt the mixture, during which the wrought iron was dissolved (not melted), insured the further elimination of silicon, so that there was no chance of a gray casting being the result, unless more silicon or more aluminum was added and not the silicate of aluminum. The addition of the ferro-silicate of iron and aluminum amounted to nothing in my judgment, and would simply form a slag if heated hot enough.

Mr. W. Wallace Christie.—I have tried, since writing this paper, to obtain the chemical combination of the ferro-silicate of iron and aluminum, but without success, which I regret.

The paper is incomplete as regards an analysis of the resulting metals, but, as far as it goes, is a complete report of all tests which we have made with them.

CCCCLXXXVIII.*

THE DENSITY OF WATER AT DIFFERENT TEMPERATURES.

BY A. F. NAGLE, CHICAGO, ILLS.

(Member of the Society.)

THE writer has frequent occasion to use a table of "weight of water at different temperatures," and he is somewhat uncertain as to the most reliable table published. There does not appear to be any unanimity among authors as to the density of water at *any* temperature. One would suppose that at its maximum density, and at the boiling point, there would be perfect agreement, but there is not.

J. W. Nystrom, "Steam Engineering," gives a table computed for every degree to four places of decimals from the original experiments of Kopp, beginning with its maximum density at 40° as 62.3880 lbs. c.f., and ending at 212° with 59.8376 lbs. c.f.

"Steam," published by the Babcock & Wilcox Co., gives a table for nearly every degree, beginning with its maximum density at 39.1° as 62.425 lbs. c.f., and ending at 212° with 59.760 lbs. c.f.

This table is evidently computed from Rankine's approximate formula as given by Clark.

Mr. Clark, "Steam Engine," gives a table for every five degrees computed from Rankine's formula, giving the same results as above quoted from "Steam," but not worked out with as much care.

Thurston, "Steam Engine and Boiler Trials," does not mention his authority, but gives a table for every 10° with its maximum density at 39.2° as 62.425 lbs. c.f., and at 212° as 59.707 lbs. c.f.

The British imperial standard gallon is based upon 10 lbs. of pure water at 62° Fah., 277.274 cubic inches for its volume. A simple calculation will give the weight of a cubic foot at this temperature as 62.3210 lbs., and Mr. Clark gives it as 62.3550 lbs.

* Presented at the San Francisco Meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the Transactions.

Grouping these figures in a table enables us to compare them the better.

WEIGHT OF A CUBIC FOOT OF WATER AS GIVEN BY DIFFERENT AUTHORITIES.

Tem. Fah.	Clark.	Nystrom.	Thurston.	British Gallon.	American Gallon.
39.1	62.425	62.3879	62.4250	‡ 62.3880	‡ 62.3791
40.	62.425	62.3880	62.4230		
62.	62.355	62.3211		62.3210	
212	59.760	59.8376	59.7070		
"	By		* 59.8445		
"	measurement 59.640		† 59.8330		

* Corrected (?). † Kopp corrected by Porter. ‡ Calculated.

Mr. Nystrom says: "The most reliable experiments made on this subject are probably those of Kopp, by which the greatest density of water is indicated to be between 39° and 40°, or nearer 39°. But however accurate these experiments might have been made, it is impossible without the aid of mathematics to determine correctly the temperature of the greatest density, because the curve tangents the abscissa at that point.

"The writer treated Kopp's experiments with very careful mathematical and graphical analysis, the result of which located the greatest density of water at 40°." (Now, however, generally accepted as being at 39.1°.)

"The formula for volume of water deduced from Kopp's experiments is

$$V=\frac{(t-40)^2}{1400\,t+398{,}500}, \qquad (1)$$

"The volume deduced from the same experiments, but with the assertion that the greatest density of water is at 39°, will be

$$V=\frac{(t-39)^2}{1400\,t+405{,}400}, \qquad (2)$$

"The formula (1) is the most correct." (?)

Thurston does not mention his authority. It is evident it is not Kopp, nor Kopp corrected by Porter, both tables being

given on the same page with his own, for it is not in agreement with either.

The maximum density Thurston gives at 39.1° as 62.425 lbs., but at the boiling point, while he gives the same comparative volume as Kopp, namely, 1.04312, yet the weight deduced from this ratio of volumes is given as 59.707 lbs. There is either an error in the comparative volumes, or in the weight, for they do not check up correctly, because,

$$\frac{62.425}{1.04312} = 59.8445 \ lbs.$$

instead of 59.707 lbs.

If we use Porter's corrected figure, we have

$$\frac{62.425}{1.04332} = 59.8330 \ lbs.$$

Mr. Clark does not say upon whose experimental data Professor Rankine's formula is based, I suspect it is Kopp.

Rankine's formula is as follows:

$$D_1 \ nearly = \frac{2 \ D_0}{\frac{t + 461}{500} + \frac{500}{t + 461}}$$

at which $D_0 = 62.425$ lbs. per cubic foot, the maximum density of water.

And $D_1 =$ its density at a given temperature, t, Fahr.

Mr. Clark says: "The results given by this rule are very nearly exact for the lower temperatures, but for the higher temperatures they are too great. For 212° Fahr., the density of water by the rule is 59.76 lbs., but it is actually only 59.64 lbs."

I presume it is safe to accept Mr. Clark's figure for this corrected weight by direct measurement at 212°, although no other author quoted gives it. One would suppose that the weight of a cubic foot for 62°, from which the Imperial gallon is established, would be absolutely correct, but I believe this standard was adopted when it was *supposed that* to be the relation of volume to weight, but as it is not, the volume of 277.274 cubic inches remains to be the correct standard, and the weight of water may as well be left out of the discussion.

So great, however, is my confidence in Mr. Clark's accuracy of

statement, that I will adopt his weights at the maximum density of water at 39.1° as 62.425 lbs. c.f., and at 212° as 59.640 lbs. c.f.

Having accepted these terminal points, how shall we obtain correct intermediate ones?

Turning to Mr. Clark's table, the third column of which is reprinted in Table I of this paper, the first differences for the comparative volumes were taken and plotted as shown in dotted lines in Figure 76. There are evidently errors in the computations of the table. I did not recompute the table, but constructed a smooth and graceful curve through the dotted lines, so that the total of the ordinates would be exactly the same as before. The heavy full line shows this line, and the ordinates measured therefrom are given in Table II. By taking second differences, the

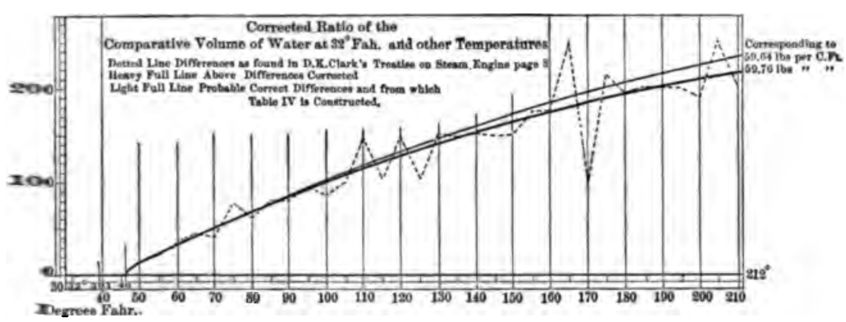

FIG. 76.

table was made somewhat more accurate than could be obtained from measurements.

This curve, and consequent table, could be received as perhaps a perfect exemplification of Rankine's formula, and yet it is possible, if the formula were worked out for every degree, that it would give as perfect a curve as I have laid out—but it is not *this* curve we want. Acting upon Mr. Clark's remark that the formula gives results almost in exact agreement with the facts at the lower temperatures, but too great at the higher, I drew a line of similar curvature as the former, tangent to it at the lower end, and elevated it at the upper end until the sum total of the ordinates, representing differences, gave the required amount necessary to make the weight of a cubic foot equal to 59.640 lbs., instead of 59.760 lbs.

The curve is shown in a light, full line and the ordinates obtained are given in Table III. for every 5°.

The weights were computed from the comparative volumes given in Table III., and a final Table IV. was worked up with great care for every degree from 32° to 212° for both weights and comparative volumes. First a table was constructed from Table

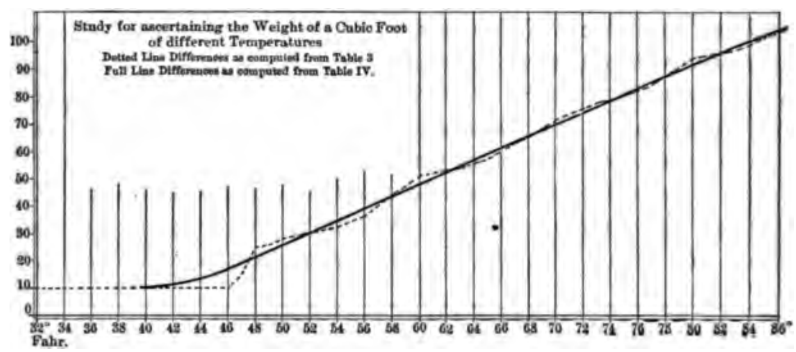

FIG. 77.

III. for every degree with the purpose of keeping as closely as possible to the important figures given by Mr. Clark, and at one time I was disposed to leave the matter there, but plotting those results as shown in Figure 77, it was so unsatisfactory

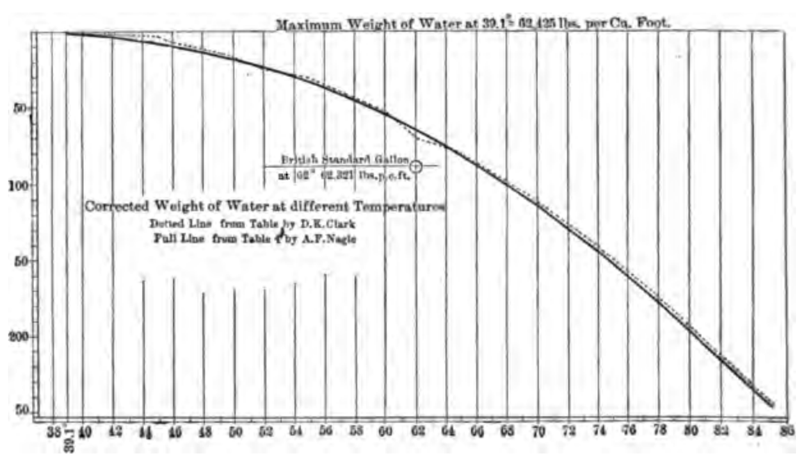

FIG. 78.

that I decided to draw a new line of differences of weights. The dotted line shows the change made and I believe it is justifiable. The decimals only are plotted. From these differences a new table was constructed and the comparative volumes computed with water at its maximum density, 39.1°, instead of at 32°.

In Figure 78 the decreasing weight of a cubic foot is laid out, up to 86°, and what discrepancies exist between what Mr. Clark has given and what the writer gives I believe are in favor of the latter.

AGREEMENTS AND DISCREPANCIES.

I was very reluctant to change any figures at the lower temperatures which might be considered as standards:—

At 39.1° I adopted 62.425 lbs., although Chas. T. Porter gives it as 62.4245 lbs. Mr. Clark gives for 46° and 32°, 62.4180 lbs. I could not get nearer to it than 62.41623 lbs.

A study of the diagrams will, I believe, justify this change.

I have given the weights below 39.1° in the same decreasing order and ratio as above 39.1° thus reaching at 32°, 62.4162 lbs., instead of 62.4180 lbs., as given by Mr. Clark. It was because I felt some uncertainty as to the absolute correctness of this figure that I thought it safer and better to base the comparative volumes upon its maximum density rather than at 32°.

At 52.3° given by Mr. Clark as 62.400 lbs., the agreement is almost perfect, it being 62.40062 lbs.

At 62° occurs the greatest deviation, but a study of the diagram will justify the figure I have adopted, namely, 62.3596 lbs., instead of 62.3550 lbs.

At 65° there is almost perfect agreement.

After this my line gradually draws away from that obtained by Rankine's formulæ, so that the terminal at 212° may agree with what Mr. Clark says is the weight by actual measurement.

The American Gallon has a volume of 231 cubic inches, and that is enough to define it. Mr. Trautwine, however, says it contains 8.33888 lbs. of pure water at its maximum density of 39.1°.

Upon that basis a cubic foot would weigh

$$\frac{8.33888 \times 1728}{231} = 62.3791 \text{ lbs.}$$

TABLE I.

By D. K. Clark.

Temperature. Fahr.	Comparative Volume with water at 32 degrees.	1st Differences.	2d Differences.	Temperature. Fahr.	Comparative Volume with water at 32 degrees.	1st Differences.	2d Differences.
32	1.00000	......	......	120	1.01139	150	50
35	0.99993	7	......	125	1.01239	100	−50
39.1	0.99989	4	−3	130	1.01390	151	51
40.	0.99989	0	−4	135	1.01539	149	−2
45	0.99993	4	4	140	1.01690	151	2
46	1.00000	7	3	145	1.01839	149	−2
50	1.00015	15	8	150	1.01989	150	1
52.3	1.00029	......	......	155	1.02164	175	25
55	1.00038	23	8	160	1.02340	176	1
60	1.00074	36	13	165	1.02589	249	73
62	1.00101	......	—	170	1.02690	101	−148
65	1.00119	45	9	175	1.02906	216	115
.70	1.00160	41	−4	180	1.03100	194	−22
75	1.00239	79	38	185	1.03300	200	6
80	1.00299	60	−19	190	1.03500	200	0
85	1.00379	80	20	195	1.03700	200	0
90	1.00459	80	0	200	1.03889	189	−11
95	1.00554	95	15	205	1.04140	251	62
100	1.00639	85	−10	210	1.04340	200	−51
105	1.00739	100	15	212	1.04440	100	
110	1.00889	150	50				
115	1.00989	100	−50			4440	

TABLE II.

COMPARATIVE VOLUME OF WATER AT DIFFERENT TEMPERATURES.

By D. K. Clark, corrected by A. F. Nagle.

Temperature. Fahr.	Comparative Volume with water at 32 degrees.	1st Differences.	2d Differences.	Temperature. Fahr.	Comparative Volume with water at 32 degrees.	1st Differences.	2d Differences.
32	1.00000	......	......	125	1.01253	134	6
35	0.99993	......	......	130	1.01393	140	6
39.1	0.99989	......	......	135	1.01539	146	6
40.	0.99989	......	......	140	1.01691	152	6
45	0.99993	......	......	145	1.01849	158	6
46	1.00000	......	......	150	1.02013	164	6
50	1.00015	15	......	155	1.02182	169	5
52.3	1.00029	......	......	160	1.02356	174	5
55	1.00040	25	10	165	1.02535	179	5
60	1.00074	34	9	170	1.02719	184	5
62	1.00101	......	......	175	1.02908	189	5
65	1.00117	43	9	180	1.03102	194	5
70	1.00169	52	9	185	1.03300	198	4
75	1.00229	60	8	190	1.03502	202	4
80	1.00297	68	8	195	1.03708	206	4
85	1.00378	76	8	200	1.03918	210	4
90	1.00457	84	8	205	1.04132	214	4
95	1.00549	92	8	210	1.04350	218	4
100	1.00649	100	8				
105	1.00756	107	7	212	1.04440	4350	
110	1.00870	114	7			90	
115	1.00991	121	7				
120	1.01119	128	7			4440	

TABLE III.

NEW COMPARATIVE VOLUME OF WATER AT DIFFERENT TEMPERATURES.

By A. F. Nagle.

Temperatures. Fahr.	Comparative volume, with water at 32°.	1st Differences.	2d Differences.	Temperatures. Fahr.	Comparative volume, with water at 32°.	1st Differences.	2d Differences.
32	1.00000	....	....	125	1.01294	139	7
35	0.99993	−7	....	130	1.01440	146	7
39.1	0.99989	−4	....	135	1.01593	153	7
40	0.99989	0	....	140	1.01752	159	6
45	0.99993	4	....	145	1.01917	165	6
46	1.00000	7	....	150	1.02088	171	6
50	1.00015	15	....	155	1.02265	177	6
52.3	1.00029	....	....	160	1.02448	183	6
55	1.00040	25	10	165	1.02637	189	6
60	1.00075	35	10	170	1.02832	195	6
62	....	....	....	175	1.03032	200	5
65	1.00119	44	9	180	1.03237	205	5
70	1.00172	53	9	185	1.03447	210	5
75	1.00234	62	9	190	1.03662	215	5
80	1.00305	71	9	195	1.03881	219	4
85	1.00384	79	8	200	1.04105	224	5
90	1.00471	87	8	205	1.04333	228	4
95	1.00566	95	8	210	1.04565	232	4
100	1.00669	103	8	212	1.04659		
105	1.00780	111	8			4,565	
110	1.00898	118	7			94	
115	1.01023	125	7				
120	1.01155	132	7			4,659	

TABLE IV.

NEW TABLE OF COMPARATIVE VOLUME AND WEIGHTS OF WATER AT DIFFERENT TEMPERATURES.

By A. F. Nagle.

Temp. Fahr.	Comparative volume. Water at 39.1° = 1.	Differences.	Weight of one cubic ft. Pounds.	Differences.	Temp. Fahr.	Comparative volume. Water at 39.1° = 1.	Differences.	Weight of one cubic ft. Pounds.	Differences.
32	1.00014	.....	62.41623	....	93	1.00538	19	62.0908	117
33	1.00011	3	62.41790	16.7	94	1.00557	19	62.0789	119
34	1.00009	2	62.41935	14.5	95	1.00577	20	62.0665	121
35	1.00007	2	62.42063	12.8	96	1.00597	20	62.0545	123
36	1.00005	2	62.42181	11.6	97	1.00617	20	62.0420	125
37	1.00003	2	62.42292	11.7	98	1.00638	21	62.0293	127
38	1.00001	2	62.42395	10.6	99	1.00659	21	62.0164	129
39.1	1.00000	1	62.42500	10.2	100	1.00680	21	62.0033	131
40	1.00001	1	62.42398	10.2	101	1.00702	22	61.9900	133
41	1.00003	2	62.42292	10.6	102	1.00724	22	61.9765	135
42	1.00005	2	62.42181	11.1	103	1.00746	22	61.9629	136
43	1.00007	2	62.42063	11.8	104	1.00768	22	61.9491	138
44	1.00009	2	62.41935	12.8	105	1.00791	23	61.9351	140
45	1.00011	2	62.41790	14.5	106	1.00814	23	61.9210	141
46	1.00014	3	62.41623	16.7	107	1.00837	23	61.9067	143
47	1.00017	3	62.41434	18.9	108	1.00861	24	61.8922	145
48	1.00020	2	62.41223	21.1	109	1.00885	24	61.8775	147
49	1.00024	4	62.40990	23.3	110	1.00909	24	61.8626	149
50	1.00028	4	62.40735	25.5	111	1.00934	25	61.8475	151
51	1.00032	4	62.40458	27.7	112	1.00959	25	61.8323	152
52	1.00037	5	62.40159	29.9	113	1.00984	25	61.8170	153
52.3	1.00039	.....	62.40068	....	114	1.01009	25	61.8015	155
53	1.00042	5	62.39838	32.1	115	1.01034	25	61.7859	156
54	1.00048	6	62.39495	34.3	116	1.01060	26	61.7701	158
55	1.00054	6	62.39130	36.5	117	1.01086	26	61.7541	160
56	1.00060	6	62.38743	38.7	118	1.01112	26	61.7380	161
57	1.00067	7	62.38334	40.9	119	1.01139	27	61.7217	163
58	1.00074	7	62.37903	43.1	120	1.01166	27	61.7053	164
59	1.00081	7	62.37450	45.3	121	1.01193	27	61.6887	166
60	1.00089	8	62.36975	47.5	122	1.01221	28	61.6719	168
61	1.00097	8	62.36478	49.7	123	1.01249	28	61.6549	170
62	1.00105	8	62.35959	51.9	124	1.01277	28	61.6378	171
63	1.00113	8	62.35418	54.1	125	1.01306	29	61.6206	172
64	1.00122	9	62.34855	56.3	126	1.01334	29	61.6032	174
65	1.00132	10	62.34270	58.5	127	1.01363	29	61.5856	176
66	1.00142	10	62.33663	60.7	128	1.01392	29	61.5679	177
67	1.00152	10	62.33034	62.9	129	1.01421	29	61.5500	179
68	1.00162	10	62.32383	65.1	130	1.01451	30	61.5320	180
69	1.00173	11	62.31710	67.3	131	1.01481	30	61.5138	182
70	1.00184	11	62.31015	69.5	132	1.01511	30	61.4954	184
71	1.00196	12	62.30298	71.7	133	1.01542	31	61.4769	186
72	1.00208	12	62.29559	73.9	134	1.01573	31	61.4581	187
73	1.00220	12	62.28798	76.1	135	1.01604	31	61.4392	189
74	1.00232	12	62.28015	78.3	136	1.01635	31	61.4202	190
75	1.00245	13	62.27210	80.5	137	1.01667	32	61.4011	191
76	1.00258	13	62.26380	83	138	1.01699	32	61.3819	192
77	1.00272	14	62.25525	85.5	139	1.01731	32	61.3626	193
78	1.00286	14	62.2465	87.5	140	1.01763	32	61.3432	194
79	1.00301	15	62.2375	90	141	1.01795	32	61.3237	195
80	1.00316	15	62.2283	92	142	1.01828	33	61.3040	197
81	1.00331	15	62.2189	94	143	1.01861	33	61.2841	199
82	1.00346	15	62.2093	96	144	1.01894	33	61.2641	200
83	1.00362	16	62.1995	98	145	1.01928	34	61.2440	201
84	1.00378	16	62.1894	100	146	1.01962	34	61.2238	202
85	1.00395	17	62.1792	102	147	1.01996	34	61.2034	204
86	1.00412	17	62.1688	104	148	1.02030	34	61.1829	205
87	1.00429	17	62.1583	105	149	1.02064	34	61.1622	207
88	1.00446	17	62.1475	106	150	1.02099	35	61.1413	209
89	1.00464	18	62.1365	110	151	1.02134	35	61.1203	210
90	1.00482	18	62.1253	112	152	1.02169	35	61.0992	211
91	1.00500	18	62.1140	113	153	1.02204	35	61.0780	212
92	1.00519	19	62.1025	115	154	1.02240	36	61.0568	212

TABLE IV.—*Continued.*

NEW TABLE OF COMPARATIVE VOLUME AND WEIGHTS OF WATER AT DIFFERENT TEMPERATURES.

By A. F. Nagle.

Temp. Fahr.	Comparative volume. Water at 39.1° = 1.	Differences.	Weight of one cubic ft. Pounds.	Differences.	Temp. Fahr.	Comparative volume. Water at 39.1° = 1.	Differences.	Weight of one cubic ft. Pounds.	Differences.
155	1.02276	36	61.0355	213	184	1.03416	42	60.3626	246
156	1.02312	36	61.1040	215	185	1.03459	43	60.3379	247
157	1.02348	36	60.9923	217	186	1.03502	43	60.3141	248
158	1.02384	36	60.9705	218	187	1.03545	43	60.2881	250
159	1.02421	37	60.9486	219	188	1.03588	43	60.2630	251
160	1.02458	37	60.9266	220	189	1.03631	43	60.2379	251
161	1.02496	38	60.9045	221	190	1.03674	43	60.2128	251
162	1.02534	38	60.8822	224	191	1.03717	43	60.1876	252
163	1.02572	38	60.8597	225	192	1.03760	43	60.1624	252
164	1.02610	38	60.8370	227	193	1.03804	44	60.1370	254
165	1.02648	38	60.8140	229	194	1.03848	44	60.1115	255
166	1.02687	39	60.7912	229	195	1.03892	44	60.0860	255
167	1.02726	39	60.7682	230	196	1.03937	45	60.0604	256
168	1.02765	39	60.7451	231	197	1.03982	45	60.0346	258
169	1.02804	39	60.7220	231	198	1.04027	45	60.0087	259
170	1.02843	39	60.6988	232	199	1.04072	45	59.9828	259
171	1.02883	40	60.6755	233	200	1.04117	45	59.9569	259
172	1.02923	40	60.6520	235	201	1.04162	45	59.9309	260
173	1.02963	40	60.6284	236	202	1.04207	45	59.9049	260
174	1.03003	40	60.6048	236	203	1.04252	45	59.8788	261
175	1.03043	40	60.5812	236	204	1.04298	46	59.8525	263
176	1.03084	41	60.5574	238	205	1.04344	46	59.8261	264
177	1.03125	41	60.5334	240	206	1.04390	46	59.7997	264
178	1.03166	41	60.5093	241	207	1.04436	46	59.7733	264
179	1.03207	41	60.4851	242	208	1.04482	46	59.7467	266
180	1.03248	41	60.4608	243	209	1.04529	47	59.7201	266
181	1.03290	42	60.4363	245	210	1.04576	47	59.6935	266
182	1.03332	42	60.4117	245	211	1.04623	47	59.6668	267
183	1.03374	42	60.3872	246	212	1.04670	47	59.6400	268

DISCUSSION.

*Mr. E. J. Molera.**—Mr. Nagle has taken the data as given in the manuals which contain compilations from different authors for the use of engineers. It is a pity, because this subject has been very exhaustively treated and very beautifully treated, and the problem which Mr. Nagle proposes to solve has been already done. The curve has been constructed by M. Despretz, but instead of starting from the point of maximum density, he started at 14° Fahr. Despretz found the volumes of water corresponding to the temperatures from 14° Fahr. to the boiling point. This curve is perfect; it is practically a parabola for the lower temperatures.

Nystrom, one of the authors which Mr. Nagle takes for his

* Of San Francisco; by invitation.

authority, establishes the temperature of the maximum density of water from his own calculations from the data given by Kopp. Taking such data, I make the temperature corresponding to the maximum density of water 4.02 instead of 4.01, which agrees also with the values that Pierre and Despretz, Hällström and Rosetti give. I have here the curve and the method followed by M. Despretz, but I am afraid that if I attempted to give the details to the Society I would consume more time than the members care to devote to it. However, it is here. I have a list of authors and tables which the writer is welcome to have. Rosetti has done fully what the writer proposes to do; he has taken the values given by all the scientific investigators from the degrees of temperature of minus 10° C., or plus 14° Fahr., to the boiling point. The number of authors which he considers, to form his table, are: First, the values of Kopp, which were obtained in 1847. I may say that before that the great physicist Dalton gave already, in 1801, the density of water for the most important points of temperature; but he did not follow as accurate a method as the later authors, and, therefore, his results are not considered in the table. Then follows Isidore Pierre, who followed a method of his own. I may here mention that every one of the authors mentioned not only computed from the work of others, but they themselves experimented, by their own different methods, upon the expansion of liquids. Pierre conducted his investigations in 1845 to 1850; Despretz in 1839; Hagen in 1855; Matthiessen in 1866. Rosetti has two tables—one of values obtained in 1866, and another obtained in 1868; Weidner, 1866; Kremers, 1861. Rosetti obtains from all these values the mean one; the result, in my opinion, is as accurate as it will ever be found. However, through the courtesy of an old chief of mine, the late General Ibañez, the founder and late president of the International Bureau of Weights and Measures, I have learned that Dr. O. J. Broch, the secretary of said Society, has again gone over the whole field, in order to determine exactly the coefficient of expansion, not only of solids, but of liquids as well. He has published a table for the volume of water corresponding to temperatures from 0° C. to 30° C. or 86° Fahr., to be found in the publications of the International Commission of Weights and Measures.

Prof. D. S. Jacobus.—Mr. Molera has gone into this matter with much more care and exactness than I have done, but as I

given on the same page with his own, for it is not in agreement with either.

The maximum density Thurston gives at 39.1° as 62.425 lbs., but at the boiling point, while he gives the same comparative volume as Kopp, namely, 1.04312, yet the weight deduced from this ratio of volumes is given as 59.707 lbs. There is either an error in the comparative volumes, or in the weight, for they do not check up correctly, because,

$$\frac{62.425}{1.04312} = 59.8445 \textit{ lbs.}$$

instead of 59.707 lbs.

If we use Porter's corrected figure, we have

$$\frac{62.425}{1.04332} = 59.8330 \textit{ lbs.}$$

Mr. Clark does not say upon whose experimental data Professor Rankine's formula is based, I suspect it is Kopp.

Rankine's formula is as follows:

$$D_1 \textit{ nearly} = \frac{2\,D_0}{\frac{t + 461}{500} + \frac{500}{t + 461}}$$

at which $D_0 = 62.425$ lbs. per cubic foot, the maximum density of water.

And D_1 = its density at a given temperature, t, Fahr.

Mr. Clark says: "The results given by this rule are very nearly exact for the lower temperatures, but for the higher temperatures they are too great. For 212° Fahr., the density of water by the rule is 59.76 lbs., but it is actually only 59.64 lbs."

I presume it is safe to accept Mr. Clark's figure for this corrected weight by direct measurement at 212°, although no other author quoted gives it. One would suppose that the weight of a cubic foot for 62°, from which the Imperial gallon is established, would be absolutely correct, but I believe this standard was adopted when it was *supposed that* to be the relation of volume to weight, but as it is not, the volume of 277.274 cubic inches remains to be the correct standard, and the weight of water may as well be left out of the discussion.

So great, however, is my confidence in Mr. Clark's accuracy of

statement, that I will adopt his weights at the maximum density of water at 39.1° as 62.425 lbs. c.f., and at 212° as 59.640 lbs. c.f.

Having accepted these terminal points, how shall we obtain correct intermediate ones?

Turning to Mr. Clark's table, the third column of which is reprinted in Table I of this paper, the first differences for the comparative volumes were taken and plotted as shown in dotted lines in Figure 76. There are evidently errors in the computations of the table. I did not recompute the table, but constructed a smooth and graceful curve through the dotted lines, so that the total of the ordinates would be exactly the same as before. The heavy full line shows this line, and the ordinates measured therefrom are given in Table II. By taking second differences, the

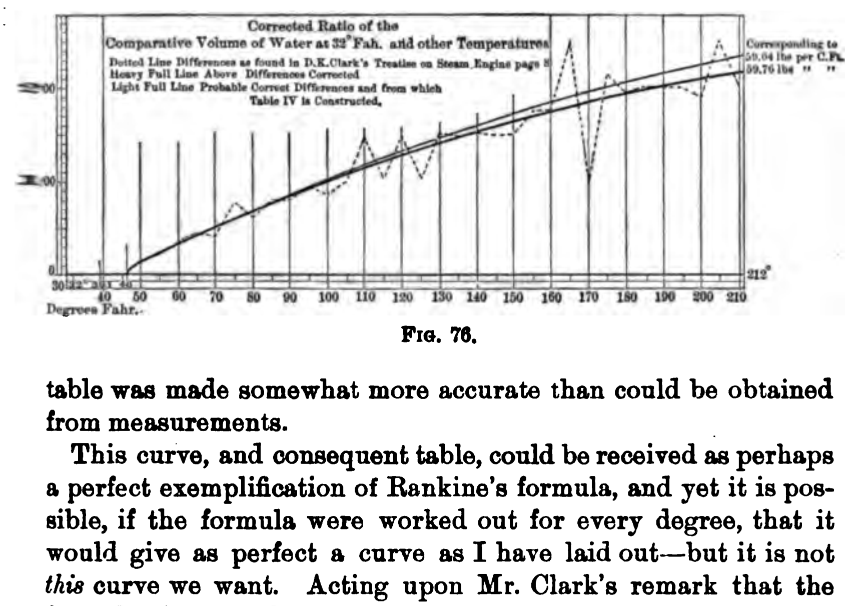

FIG. 76.

table was made somewhat more accurate than could be obtained from measurements.

This curve, and consequent table, could be received as perhaps a perfect exemplification of Rankine's formula, and yet it is possible, if the formula were worked out for every degree, that it would give as perfect a curve as I have laid out—but it is not *this* curve we want. Acting upon Mr. Clark's remark that the formula gives results almost in exact agreement with the facts at the lower temperatures, but too great at the higher, I drew a line of similar curvature as the former, tangent to it at the lower end, and elevated it at the upper end until the sum total of the ordinates, representing differences, gave the required amount necessary to make the weight of a cubic foot equal to 59.640 lbs., instead of 59.760 lbs.

The curve is shown in a light, full line and the ordinates obtained are given in Table III. for every 5°.

given on the same page with his own, for it is not in agreement with either.

The maximum density Thurston gives at 39.1° as 62.425 lbs., but at the boiling point, while he gives the same comparative volume as Kopp, namely, 1.04312, yet the weight deduced from this ratio of volumes is given as 59.707 lbs. There is either an error in the comparative volumes, or in the weight, for they do not check-up correctly, because,

$$\frac{62.425}{1.04312} = 59.8445 \text{ } lbs.$$

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at which $D_0 = 62.425$ lbs. per cubic foot, the maximum density of water.

And $D_1 =$ its density at a given temperature, t, Fahr.

Mr. Clark says: "The results given by this rule are very nearly exact for the lower temperatures, but for the higher temperatures they are too great. For 212° Fahr., the density of water by the rule is 59.76 lbs., but it is actually only 59.64 lbs."

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statement, that I will adopt his weights at the maximum density of water at 39.1° as 62.425 lbs. c.f., and at 212° as 59.640 lbs. c.f.

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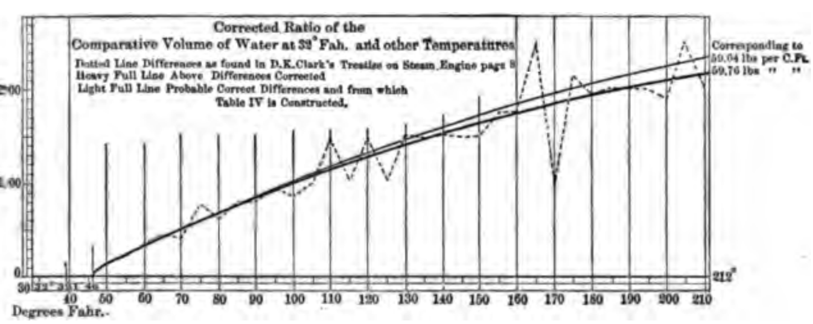

FIG. 76.

table was made somewhat more accurate than could be obtained from measurements.

This curve, and consequent table, could be received as perhaps a perfect exemplification of Rankine's formula, and yet it is possible, if the formula were worked out for every degree, that it would give as perfect a curve as I have laid out—but it is not *this* curve we want. Acting upon Mr. Clark's remark that the formula gives results almost in exact agreement with the facts at the lower temperatures, but too great at the higher, I drew a line of similar curvature as the former, tangent to it at the lower end, and elevated it at the upper end until the sum total of the ordinates, representing differences, gave the required amount necessary to make the weight of a cubic foot equal to 59.640 lbs., instead of 59.760 lbs.

The curve is shown in a light, full line and the ordinates obtained are given in Table III. for every 5°.

The weights were computed from the comparative volumes given in Table III., and a final Table IV. was worked up with great care for every degree from 32° to 212° for both weights and comparative volumes. First a table was constructed from Table

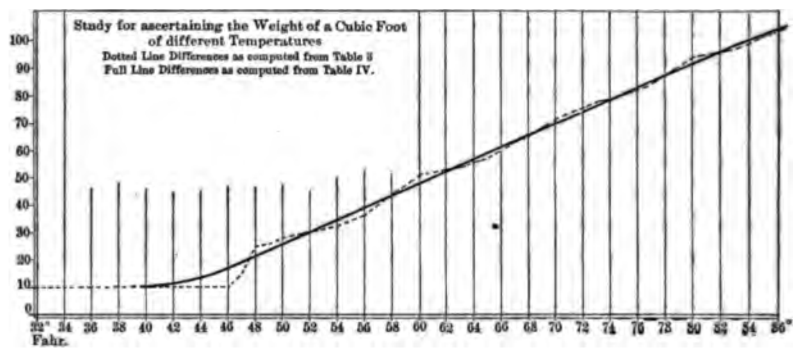

FIG. 77.

III. for every degree with the purpose of keeping as closely as possible to the important figures given by Mr. Clark, and at one time I was disposed to leave the matter there, but plotting those results as shown in Figure 77, it was so unsatisfactory

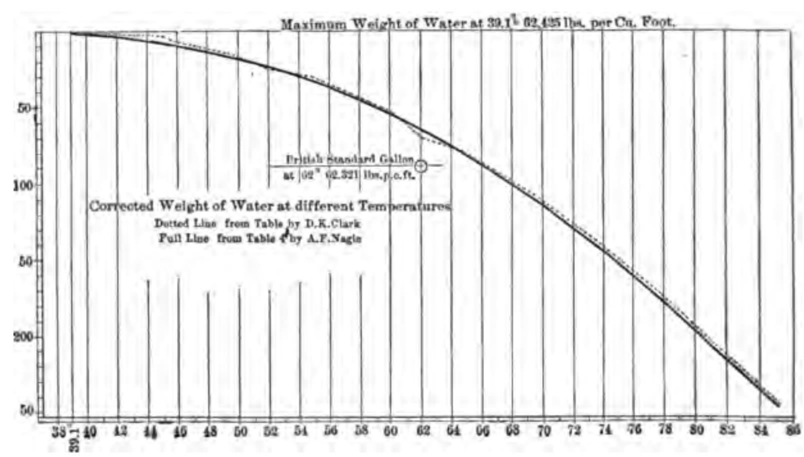

FIG. 78.

that I decided to draw a new line of differences of weights. The dotted line shows the change made and I believe it is justifiable. The decimals only are plotted. From these differences a new table was constructed and the comparative volumes computed with water at its maximum density, 39.1°, instead of at 32°.

In Figure 78 the decreasing weight of a cubic foot is laid out, up to 86°, and what discrepancies exist between what Mr. Clark has given and what the writer gives I believe are in favor of the latter.

AGREEMENTS AND DISCREPANCIES.

I was very reluctant to change any figures at the lower temperatures which might be considered as standards:—

At 39.1° I adopted 62.425 lbs., although Chas. T. Porter gives it as 62.4245 lbs. Mr. Clark gives for 46° and 32°, 62.4180 lbs. I could not get nearer to it than 62.41623 lbs.

A study of the diagrams will, I believe, justify this change.

I have given the weights below 39.1° in the same decreasing order and ratio as above 39.1° thus reaching at 32°, 62.4162 lbs., instead of 62.4180 lbs., as given by Mr. Clark. It was because I felt some uncertainty as to the absolute correctness of this figure that I thought it safer and better to base the comparative volumes upon its maximum density rather than at 32°.

At 52.3° given by Mr. Clark as 62.400 lbs., the agreement is almost perfect, it being 62.40062 lbs.

At 62° occurs the greatest deviation, but a study of the diagram will justify the figure I have adopted, namely, 62.3596 lbs., instead of 62.3550 lbs.

At 65° there is almost perfect agreement.

After this my line gradually draws away from that obtained by Rankine's formulæ, so that the terminal at 212° may agree with what Mr. Clark says is the weight by actual measurement.

The American Gallon has a volume of 231 cubic inches, and that is enough to define it. Mr. Trautwine, however, says it contains 8.33888 lbs. of pure water at its maximum density of 39.1°.

Upon that basis a cubic foot would weigh

$$\frac{8.33888 \times 1728}{231} \; 62.3791 \text{ lbs.}$$

TABLE I.

By D. K. Clark.

Temperature. Fahr.	Comparative Volume with water at 32 degrees.	1st Differences.	2d Differences.	Temperature. Fahr.	Comparative Volume with water at 32 degrees.	1st Differences.	2d Differences.
32	1.00000	....	....	120	1.01139	150	50
35	0.99993	7	....	125	1.01239	100	−50
39.1	0.99989	4	−8	130	1.01390	151	51
40.	0.99989	0	−4	135	1.01539	149	−2
45	0.99993	4	4	140	1.01690	151	2
46	1.00000	7	3	145	1.01839	149	−2
50	1.00015	15	8	150	1.01989	150	1
52.3	1.00029	....	....	155	1.02164	175	25
55	1.00038	23	8	160	1.02340	176	1
60	1.00074	36	13	165	1.02589	249	73
62	1.00101	....	—	170	1.02690	101	−148
65	1.00119	45	9	175	1.02906	216	115
70	1.00160	41	−4	180	1.03100	194	−22
75	1.00239	79	38	185	1.03300	200	6
80	1.00299	60	−19	190	1.03500	200	0
85	1.00379	80	20	195	1.03700	200	0
90	1.00459	80	0	200	1.03889	189	−11
95	1.00554	95	15	205	1.04140	251	62
100	1.00639	85	−10	210	1.04340	200	−51
105	1.00739	100	15	212	1.04440	100	
110	1.00889	150	50			4440	
115	1.00989	100	−50				

TABLE II.

COMPARATIVE VOLUME OF WATER AT DIFFERENT TEMPERATURES.

By D. K. Clark, corrected by A. F. Nagle.

Temperature. Fahr.	Comparative Volume with water at 32 degrees.	1st Differences.	2d Differences.	Temperature. Fahr.	Comparative Volume with water at 32 degrees.	1st Differences.	2d Differences.
32	1.00000	....	... ...	125	1.01258	134	6
35	0.99993	....	....	130	1.01398	140	6
39.1	0.99989	....	....	135	1.01539	146	6
40.	0.99989	....	....	140	1.01691	152	6
45	0.99993	....	....	145	1.01849	158	6
46	1.00000	.... ..	....	150	1.02013	164	6
50	1.00015	15	....	155	1.02182	169	5
52.3	1.00029	....	....	160	1.02356	174	5
55	1.00040	25	10	165	1.02535	179	5
60	1.00074	34	9	170	1.02719	184	5
62	1.00101	....	....	175	1.02908	189	5
65	1.00117	43	9	180	1.03102	194	5
70	1.00169	52	9	185	1.03300	198	4
75	1.00229	60	8	190	1.03502	202	4
80	1.00297	68	8	195	1.03708	206	4
85	1.00373	76	8	200	1.03918	210	4
90	1.00457	84	8	205	1.04132	214	4
95	1.00549	92	8	210	1.04350	218	4
100	1.00649	100	8				
105	1.00756	107	7	212	1.04440	4350	
110	1.00870	114	7			90	
115	1.00991	121	7				
120	1.01119	128	7			4440	

worn the shaft, and in sixteen months' trial have not shown any wear themselves."

The overseer of the spinning department of a mill in Lewiston, Me., says "that he had run a spindle fifteen weeks, and during that time he had drawn a new driving cord upon it every morning as hard as a strong man could draw it, and that it had not worn any, but is precisely the same as when he started it. He determines that it has not worn by means of fitting a follower into it that exactly fitted the bushing before it was used. He was amazed that treatment so severe did not destroy it."

The judges of the seventeenth exhibition of the Massachusetts Charitable Association of 1890, held in Boston, "recognize in the Holmes exhibit of lubricant bearings a discovery and invention of almost inestimable importance and value, and one capable of great development.

"It being a self-lubricant, its usefulness for all kinds of woollen and cotton machinery cannot be overestimated, not only from the great saving in the use of oil, but as a protection from fire caused by overheated working bearings, as it is impossible to create combustion by the use of this material.

"Your committee have devoted much time in looking up the great possibilities for usefulness, and the practibility of this discovery and invention, making careful examination of the material where it has been in practical use for several months without finding any perceptible alterations in any of its component parts. It is justly entitled to the fullest recognition of the Association. The award is a gold medal."

Another committee of this same exhibition awarded Mr. Holmes a silver medal for his graphite dynamo brushes, which are made of same materials, but with a somewhat modified treatment.

"They have been used on arc and incandescent light dynamos and also on motors for many months. They have also been tested upon a dynamo by one of the board, with satisfactory results."

In the dynamo brush made of this material we have the combination of perfect lubricity and ample electrical conductivity in one solid piece, which may be adapted in form to every requirement of dynamos and electric motors.

The Philadelphia Traction Company have used these bearings

TABLE IV.

NEW TABLE OF COMPARATIVE VOLUME AND WEIGHTS OF WATER AT DIFFERENT TEMPERATURES.

BY A. F. NAGLE.

Temp. Fahr.	Comparative volume. Water at 39.1° = 1.	Differences.	Weight of one cubic ft. Pounds.	Differences.	Temp. Fahr.	Comparative volume. Water at 39.1° = 1.	Differences.	Weight of one cubic ft. Pounds.	Differences.
32	1.00014	.....	62.41623	.....	93	1.00538	19	62.0908	117
33	1.00011	3	62.41790	16.7	94	1.00557	19	62.0789	119
34	1.00009	2	62.41935	14.5	95	1.00577	20	62.0668	121
35	1.00007	2	62.42063	12.8	96	1.00597	20	62.0545	123
36	1.00005	2	62.42181	11.8	97	1.00617	20	62.0420	125
37	1.00003	2	62.42292	11.1	98	1.00638	21	62.0293	127
38	1.00001	2	62.42398	10.6	99	1.00659	21	62.0164	129
39.1	1.00000	1	62.42500	10.2	100	1.00680	21	62.0033	131
40	1.00001	1	62.42398	10.2	101	1.00702	22	61.9900	133
41	1.00003	2	62.42292	10.6	102	1.00724	22	61.9765	135
42	1.00005	2	62.42181	11.1	103	1.00746	22	61.9629	136
43	1.00007	2	62.42063	11.8	104	1.00768	22	61.9491	138
44	1.00009	2	62.41935	12.8	105	1.00791	23	61.9351	140
45	1.00011	2	62.41790	14.5	106	1.00814	23	61.9210	141
46	1.00014	3	62.41623	16.7	107	1.00837	23	61.9067	143
47	1.00017	3	62.41434	18.9	108	1.00861	24	61.8922	145
48	1.00020	3	62.41223	21.1	109	1.00885	24	61.8775	147
49	1.00024	4	62.40990	23.3	110	1.00909	24	61.8626	149
50	1.00028	4	62.40735	25.5	111	1.00934	25	61.8475	151
51	1.00032	4	62.40458	27.7	112	1.00959	25	61.8323	152
52	1.00037	5	62.40159	29.9	113	1.00984	25	61.8170	153
52.3	1.00039	.....	62.40063	.....	114	1.01009	25	61.8015	155
53	1.00042	5	62.39838	32.1	115	1.01034	25	61.7859	156
54	1.00048	6	62.39495	34.3	116	1.01060	26	61.7701	158
55	1.00054	6	62.39130	36.5	117	1.01086	26	61.7541	160
56	1.00060	6	62.38743	38.7	118	1.01112	26	61.7380	161
57	1.00067	7	62.38334	40.9	119	1.01139	27	61.7217	163
58	1.00074	7	62.37903	43.1	120	1.01166	27	61.7053	164
59	1.00081	7	62.37450	45.3	121	1.01193	27	61.6887	166
60	1.00089	8	62.36975	47.5	122	1.01221	28	61.6719	168
61	1.00097	8	62.36478	49.7	123	1.01249	28	61.6549	170
62	1.00105	8	62.35959	51.9	124	1.01277	28	61.6378	171
63	1.00113	8	62.35418	54.1	125	1.01305	28	61.6206	172
64	1.00122	9	62.34855	56.3	126	1.01334	29	61.6032	174
65	1.00132	10	62.34270	58.5	127	1.01363	29	61.5856	176
66	1.00142	10	62.33663	60.7	128	1.01392	29	61.5679	177
67	1.00152	10	62.33034	62.9	129	1.01421	29	61.5500	179
68	1.00162	10	62.32383	65.1	130	1.01451	30	61.5320	180
69	1.00173	11	62.31710	67.3	131	1.01481	30	61.5138	182
70	1.00184	11	62.31015	69.5	132	1.01511	30	61.4954	184
71	1.00196	12	62.30298	71.7	133	1.01542	31	61.4768	186
72	1.00208	12	62.29559	73.9	134	1.01573	31	61.4581	187
73	1.00220	12	62.28798	76.1	135	1.01604	31	61.4392	189
74	1.00232	12	62.28015	78.3	136	1.01635	31	61.4202	190
75	1.00245	13	62.27210	80.5	137	1.01667	32	61.4011	191
76	1.00258	13	62.26380	83	138	1.01699	22	61.3819	192
77	1.00272	14	62.25525	85.5	139	1.01731	32	61.3626	193
78	1.00286	14	62.2465	87.5	140	1.01763	32	61.3432	194
79	1.00301	15	62.2375	90	141	1.01795	32	61.3237	195
80	1.00316	15	62.2283	92	142	1.01828	33	61.3040	197
81	1.00331	15	62.2189	94	143	1.01861	33	61.2841	199
82	1.00346	15	62.2093	96	144	1.01894	33	61.2641	200
83	1.00362	16	62.1995	98	145	1.01928	34	61.2440	201
84	1.00378	16	62.1894	100	146	1.01962	34	61.2238	202
85	1.00395	17	62.1792	102	147	1.01996	34	61.2034	204
86	1.00412	17	62.1688	104	148	1.02030	34	61.1829	205
87	1.00429	17	62.1583	105	149	1.02064	34	61.1622	207
88	1.00446	17	62.1475	108	150	1.02099	35	61.1413	209
89	1.00464	18	62.1365	110	151	1.02134	35	61.1203	210
90	1.00482	18	62.1253	112	152	1.02169	35	61.0992	211
91	1.00500	18	62.1140	113	153	1.02204	35	61.0780	212
92	1.00519	19	62.1025	115	154	1.02240	36	61.0568	213

The writer has taken a hand, during the past few months, with the inventor, in the development of this anti-frictional material, and is fully prepared to formulate and present the following well-considered conclusions.

With fibre-graphite bearings properly prepared and fitted to the supports and journals of machinery, the cost of oil together with all the appliances necessary to store, retain, convey and conduct the same to the bearings is entirely saved.

No wearing of the shaft journals has as yet been observed, and very little abrasion of the bearings has been reported by those who have used them.

During the first revolution of the shaft its concavities are filled with graphitic particles which are worn off the bearing by its rubbing action upon it. When the surface of the shaft is completely covered and evened up upon its whole exterior, the sliding will be conducted thereafter wholly upon the newly-formed graphitic surfaces, to which even the disengaged particles of the bearing will assist in lubrication; friction will in consequence be reduced to a minimum, and the shaft journal will be protected from subsequent wearing.

If we start with roundness of shaft and if we grant a certain roughness of surface, the journal after running a short time acquires smoothness by borrowing material from the bearing upon which it runs, and simultaneously fits itself the closer thereto, for its better support and running, after which the motion is all conducted upon the original and upon the acquired graphitic surfaces, with a measure of friction due to the natural lubricity of the graphite used, and without added lubricant of any kind or any attention whatever.

The labor of cleaning and oiling and the cost of waste and wiping are saved, the soil-greasing of fabrics, machinery and buildings cannot happen, and the serious danger of fire from lubricating oils, and the spontaneous combustion of them with discarded waste, are wholly removed.

The making of these bearings, much of it having been done by rude and hastily designed tools and appliances, has amply proven that the cost of them need not be *above* and may be *below* the cost of metal ones which they replace, leaving a very fair margin of profit to those who, with superior machinery and appointments, engage to manufacture them, while the user is spared the cost and nuisance of oil and oiling, and in addition thereto

is saved much of the cost of renewals as well as the anxiety which ever attends the use of metallic bearings.

A peculiarity of graphite bearings, which may have great value in the arts over the usual materials now employed, where sliding is involved, is that its friction at starting of motion is about the same as its moving friction. This property will enable the engineer to reduce the gross allowances for power to a minimum, when arranging for and providing the machinery of transmission, in cases where graphite replaces the usual metals with lubricants for bearings.

The great importance of this invention to the manufacturing world must be apparent without further reference. Every observing mind will see, although the boundary lines of its extent in any direction are not within sight, that its practical applications, numerically stated, are as multitudinous as the details of existing machines, and any list of their frictional parts would make a catalogue too voluminous for reading.

DISCUSSION.

*Mr. H. W. Harkness.**—Is the graphite pulverized which is used in these bearings?

Mr. C. N. Trump.—It is very finely pulverized and mixed with wood pulp.

Mr. Harkness.—You have to select the very best graphite, I take it?

Mr. Trump.—The graphite should be purified from all gritty matter.

Mr. Harkness.—In doing so, do you not have more or less clay?

Mr. Trump.—There does not seem to be any difficulty of that kind.

*Mr. N. S. Keith.**—This matter is of exceeding interest to me. I am a manufacturer on this coast of dynamos and electric motors. In making dynamos and motors, which are all speedy-running, the manufacturers have hitherto had a great deal of difficulty in producing a proper box and a proper lubricant. I have tried a great many experiments in this matter, using brass boxes, babbitt boxes, self-oiling boxes, sight-feed oil cups, etc. We get the best satisfaction from self-oiling arrangements. I

* Of San Francisco; by invitation.

have already prepared a chart showing how near the values given by D. K. Clark and Nagle's tables approach those by direct experiment, I will present this and state what conclusions may be drawn from it.

Fig. 197 shows two curves: one of D. K. Clark's values, and the other of Nagle's. The unbroken curve is by Nagle, and the dotted curve by Clark. It is seen that, with two exceptions, Clark's values make quite a smooth curve, notwithstanding the irregularity of the differences as given by Nagle. Clark's curve is practically correct—that is, as near as it is needed for

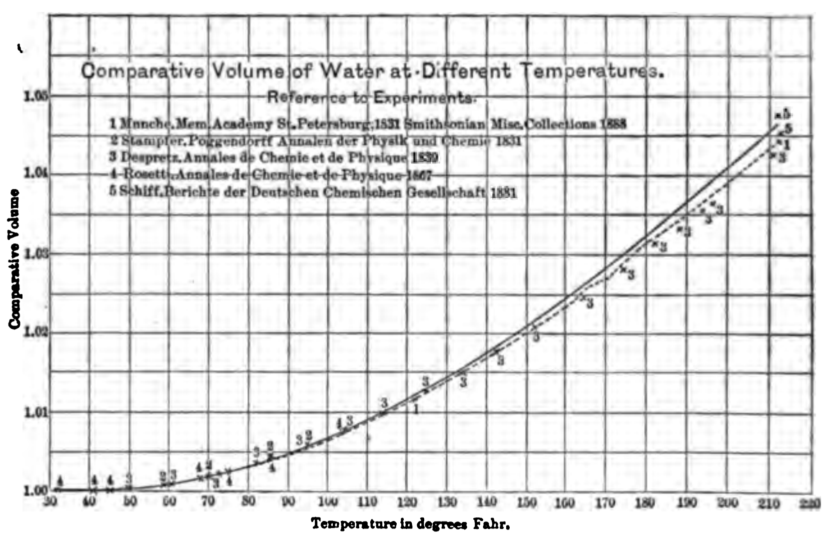

Fig. 197.

any engineering purpose in the way of standardizing meters for boiler tests, etc. In the neighborhood of from 32° to 150°, all the experimental values practically agree with Clark's curve. The experimental values are plotted, and they come so close that one mark overlaps the other. I have used the experiments of Rosetti that were mentioned by Mr. Molera. These are marked "4" on the diagram. Other experiments overlapped those by Rosetti, so that they appear confused on the diagram. At 212° only four experiments appear, which vary among themselves to such an extent that it cannot be stated which curve is the most reliable at high temperatures. The numerous experiments between 30° and 150°, however, prove that D. K. Clark's table is as accurate as the results given by Nagle for the ordinary temperatures at which water is measured.

I hope that Mr. Molera will carry on his valuable investigation, and present us with a table representing the experimental values which he considers the most reliable.

Mr. William Kent.—Mr. Nagle has adopted the same method of constructing a table of weights of water at different temperatures from the table given by Clark for every five degrees that I adopted in 1884 in my "Table for Facilitating Calculations of Boiler Tests" (*Transactions*, vol. vi. p. 90)—that is, plotting Clark's figures, and thereby discovering errors in his table, and correcting these errors by running a regular curve through the irregular plotted one. I did not, however, use Clark's figure of 59.64 (said to be obtained by direct measurement) for the weight of water at 212°, as the balance of authority seemed to be in favor of some higher figure; and after comparing the figures given by several authorities, I accepted the figure 59.76 as being as likely to be correct as any other, until other experiments are made to determine it with accuracy.

In the introduction to my tables in Vol. VI., I referred to the figures for weight of water as follows: "Also a column showing weight of water per cubic foot, according to Rankine's formula, as given in D. K. Clark's *Rules, Tables, and Data*, but corrected for apparent errors and interpolated for the degrees of temperature not given by Clark. As there is considerable difference in figures in the second decimal place of weights of water given by different authors, it is considered unnecessary to put figures beyond the second decimal place in the table, although the third decimal place was used in making the interpolations."

Taking the figures given by different authorities, as quoted by Mr. Nagle, for the weight of water at 212°, we have the following:

Nystrom, computed from Kopp	59.8376
Porter, " " "	59.8380
Thurston, authority not given	59.707
Rankine's formula	59.76
Clark, authority not given	59.64
Average	59.7555

I fail to see a sufficient reason for the entire confidence which Mr. Nagle shows in the accuracy of Clark's figure when he finds such evident errors in Clark's computed table. Clark himself makes no attempt to correct his own table by means of this figure, except for the single temperature of 212°; all the other figures in it, from 32° to 390°, being based on the formula.

A still lower figure than Clark's, however, may be obtained if we accept Rankine's statement that the formula at 212° gives too great a result by about $\frac{1}{300}$. This would make the figure only 59.56. Clark says the formula is $\frac{1}{500}$ in excess at 212°.

On the other hand, a higher figure than any of those above given is obtained from the figure for density of water at 100° C. given in Table No. 25 of Professor Whiting's work on *Physical Measurements*. This figure is .95863, and the relative densities given in the table are said to lie between the estimates of Rosetti and Volkmann, founded upon observations by Despretz, Hagen, Jolly, Kopp, Matthiessen, Pierre, and Rosetti. If we take 62.425 as the correct figure for the weight of one cubic foot of water at its maximum density, the above-quoted figure for density at 212° gives a result of 59.8425 lbs.—a higher figure than any of those quoted by Mr. Nagle.

With this weight of authority in favor of a higher figure than that given by Clark, I see no reason to doubt that the figure I accepted in 1884—viz.: 59.76—is at least as likely to be correct as Clark's figure, 59.64, and that my table is still worthy of as much confidence as Mr. Nagle's.

In view of the discrepancies between the figures of different authorities, which make all the figures in the third decimal place uncertain for temperatures below 100° Fahr., and all figures in the second decimal place uncertain for temperatures above 100° Fahr., it seems unnecessary to have a table with more than two decimal places. For all practical engineering problems the figures given in any of the tables—Mr. Clark's, Mr. Nagle's, or my own—are close enough, the possible error of any figure below 100° Fahr. being not greater than 1 part in 6,000. Even in boiler tests, where the water fed, measured in tanks or barrels, may be at as high a temperature as 160°, the possible error in using either table is not greater than 1 part in 1,000.

Prof. R. H. Thurston.—Mr. Nagle has evidently performed a large amount of patient and careful work in the preparation of this table. Assuming that his value of density for the higher temperature, taken as one of his fundamental data, is right, the table is the best that we have. I think it will be found, on investigation, if anything can be settled at all, that Kopp's work

Mr. Trump.—Only, as I say, from seeing that engine fitted up in that place in Philadelphia.

Mr. Keith.—The author states, I think, that the dynamo brushes were made of this material, and that they were running successfully.

Mr. Trump.—Yes, sir.

Mr. Keith.—That is next in importance to the electrical engineer to that of the bearings themselves.

Mr. Trump.—I might say, in answer to that question in reference to the use of them, that I saw on this dynamo that I speak of, which was running five arc lights, a pencil $\frac{1}{4}$ inch square, carrying the whole current for those five lights, and there was not a particle of roughness on the commutator, and simply a very little wear on the pencil, scarcely appreciable, so that the brush required but little adjustment. I don't know that I could give any better evidence of its conductivity. I am not an electrician, and consequently could not meet all the points that may be raised, but am informed by the manufacturers that the conductivity is 75 amperes per square inch of cross-section, without heating above 50° C.

Mr. Keith.—That is certainly better than a copper brush. Carrying, I presume, 10 amperes of current from the commutator, a copper brush would have to have a greater bearing than $\frac{1}{16}$ of a square inch, which is the amount of bearing on the commutator of a brush $\frac{1}{4}$ inch square.

Mr. G. W. Spiers.—I should think a series of experiments based upon the pressure per square inch and velocity would be exceedingly useful. If Mr. Cooper has not made such experiments, it seems to me it would be a very desirable thing for him to do, for the reason that it will give mechanics a good idea as to what the bearing will do under different conditions. What we want to know is the pressure per square inch and the velocities under these varying pressures, up to the heating or destructive point.

Mr. Trump.—I had from Mr. Cooper, since I saw him, a letter, in which he gives the size of the bearings of that loaded (5,000 lb.) shaft of which I spoke; and he also says a tubular bearing, in which a shaft ran 3,300 revolutions for five months, ten hours per day, showed wear upon a few spots only, which were as bright and smooth as glass. The bearing had not been machined at all; it was simply as it came from the mould, and derived the data for the following table." The British standard made the weight of a cubic foot at 62° = 62.321 lbs., as taken by Mr. Francis, and I see no reason for substituting Mr. Clark's estimate of 62.425, as Mr. Nagle does.

MR. FRANCIS' TABLE.

WEIGHT OF A CUBIC FOOT OF WATER AT DIFFERENT TEMPERATURES.

TEMPERATURE, FAHR.	Wt., lbs. av.	TEMPERATURE, FAHR.	Wt., lbs. av.
32°	62.375	59°	62.336
33°	62.377	60°	62.331
34°	62.378	61°	62.326
35°	62.379	62°	62.321
36°	62.380	63°	62.316
37°	62.381	64°	62.310
38°	62.381	65°	62.304
39°	62.382	66°	62.298
39.88° Max	62.382	67°	62.292
40°	62.382	68°	62.285
41°	62.381	69°	62.278
42°	62.381	70°	62.272
43°	62.380	71°	62.264
44°	62.379	72°	62.257
45°	62.378	73°	62.249
46°	62.376	74°	62.242
47°	62.375	75°	62.234
48°	62.373	76°	62.225
49°	62.371	77°	62.217
50°	62.368	78°	62.208
51°	62.365	79°	62.199
52°	62.363	80°	62.190
53°	62.359	81°	62.181
54°	62.356	82°	62.172
55°	62.352	83°	62.162
56°	62.349	84°	62.152
57°	62.345	85°	62.142
58°	62.340	86°	62.132

Mr. Francis' table, given here, covers all ordinary temperatures required in hydraulic engineering, and as it gives 62⅓ lbs. per foot at an average temperature of 60°, it is very convenient for practical operations and for wheel testing. I do not attempt to make any further refinements.

The changes of temperature in a cotton or woollen mill during a day will affect the power required ten-fold the variation to be ascertained by the weight of the water applied to the turbine, or pumped into the boiler.

CCCLXXXIX.*

SOME TESTS OF A PORTABLE BOILER.

BY WM. O. WEBBER, ERIE, PA.
(Member of the Society.)

THE accompanying series of tests made upon a new form of Portable Boiler, in which improvements have been recently made, was undertaken to demonstrate: First, the desirability of the changes since made, which have reference to the loss by radiation of heat from the back end, or combustion chamber of the boiler; secondly, the effect of loss by radiation from the uncovered heating surface of the boiler; and, lastly, the proper size of injector and safety valve required per square foot of heating surface and per square foot of grate surface, for boilers of the different sizes, in order to obtain the greatest possible evaporation during a given period. A brief description of the boiler is therefore not out of place at first.

The boiler might be described as of the simplest form of return tubular type, occupying but little space, and combining with the safety of the stationary return tubular boiler the convenience and portability of a Portable (Fig. 79). The front end of the boiler is cylindrical in form, and extends over the furnace, forming the crown sheet (Fig. 80). The rear end is oval, the lower portion extending below the cylindrical portion far enough to hold the short tubes leading from the fire-box to the back connection (Fig. 81). The short tubes in this lower portion are larger in diameter than the upper longer series of tubes, as will be seen from the accompanying table, and are arranged in the proportions of about 1 square foot to 2 square feet of heating surface, respectively. The furnace is lined with fire brick, which can be detached when desirable, these fire

* Presented at the San Francisco meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

brick being held in place by vertical iron rods which are protected from the fire and can be removed and replaced when neces-

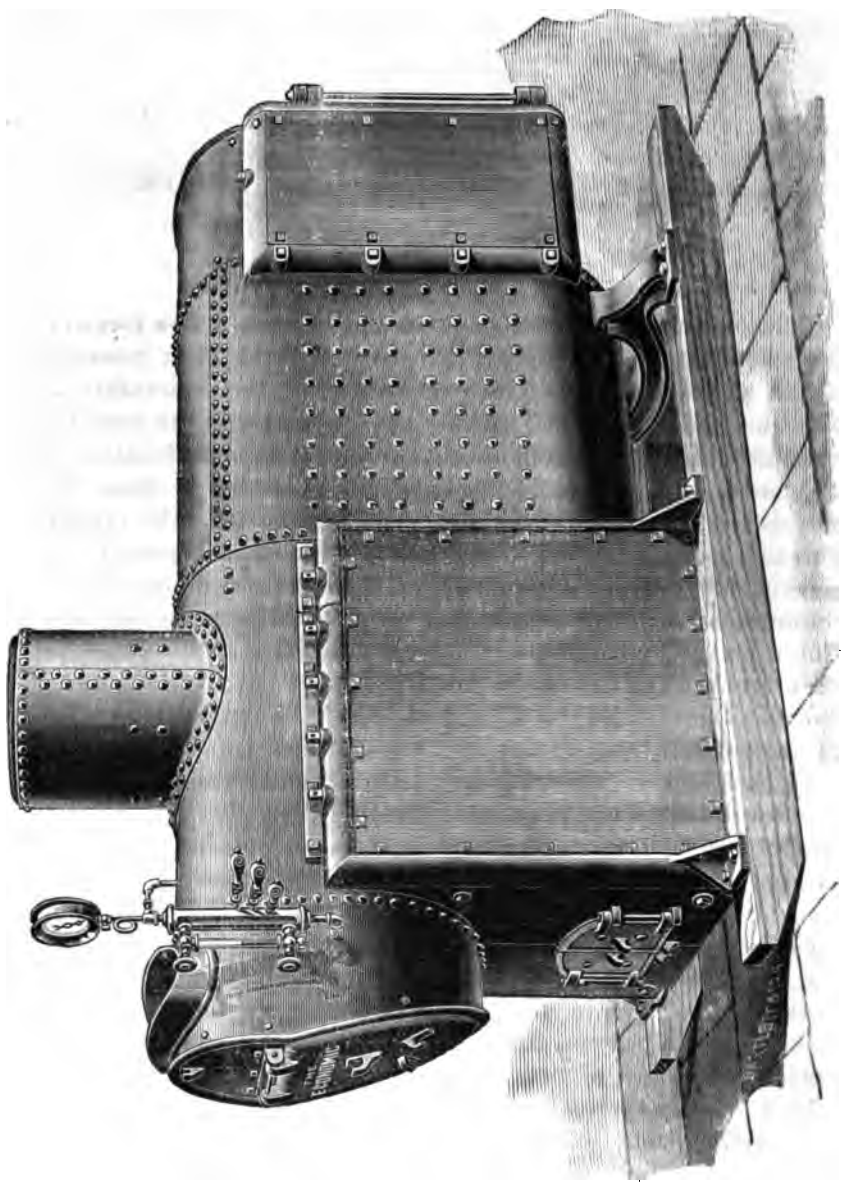

Fig. 20

sary. These iron rods are square in cross section, and are set at an angle to fit the V-notch in the ends of the fire brick, and the

CCCCLXXXVII.*

AN EXPERIMENT WITH ALUMINUM.

BY W. WALLACE CHRISTIE, NEW YORK CITY.
(Member of the Society.)

About two years ago the writer was detailed by Mr. F. W. Snow, superintendent of the Ramapo Iron Works, to make a series of mixtures for cast metal and to test the castings, ready recorded.

Mixture No. 1:

Wrought-iron turnings	10 lbs.
Cast-iron turnings	10 lbs.
Steel-rail chips	10 lbs.
Ferro-silicate of iron and aluminum	2 lbs.

Mixture No. 2:

Wrought-iron turnings	10 lbs.
Cast-iron turnings	5 lbs.
Steel-rail chips	15 lbs.
Ferro-silicate of iron and aluminum	2 lbs.

The melting was done by a well-known brass founding firm in their brass furnace. In order to melt the mixtures very high temperature was required on account of the wrought iron, which requires 3,000°; so the crucible was covered with a carbon lid and coal heaped upon it. Even then about three hours' time was required to melt it, and after being melted the ferro-silicate of iron and aluminum, which had been left out, was added, and thoroughly stirred through. The castings made were $1\frac{1}{2}$ inches diameter by 14 inches long, and in green sand without any charcoal facing, and after the skin of sand had been removed from the castings they were very smooth and clean.

Mixture No. 1 was very fluid when hot and white, but had to be poured quickly, as it soon cooled.

* Presented at the San Francisco Meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the Transactions.

ing cuts show the general construction of the boiler in a very plain manner, Fig. 79 being a side elevation of the boiler, and Figs. 80, 81, and 82 being respectively transverse and vertical cross sections and a back elevation of the boiler, mounted on skids as a portable.

The first series of trials were made on a Number 6, or 25 H.P. boiler, without the brick lining in the combustion chamber, the first trial having a stack only 25 feet long, the boiler being

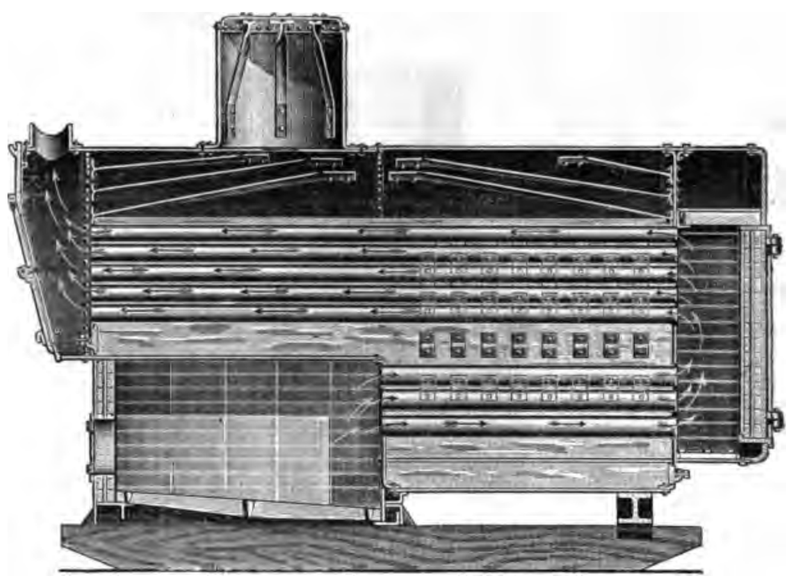

FIG. 81.

entirely uncovered, exposed to a brisk, sharp northeast wind, blowing at about the rate of fifteen miles per hour; and the usual injector and safety valve commercially supplied with the boiler. The steam generated was blown off by the safety valve into the atmosphere. This trial was conducted to maintain steam-pressure of 70 pounds of the gauge, so as to keep the data as nearly as possible in accordance with what is known as the Centennial Standard Boiler test.

The second trial, under similar conditions, was made with the boiler pressure kept as high as possible, the wind in the meantime having increased in velocity, and a cold rain falling.

The third trial was made some time later, with a stack the full length (40 feet), as required by the specifications for the boiler (see table next page), the weather having changed and become

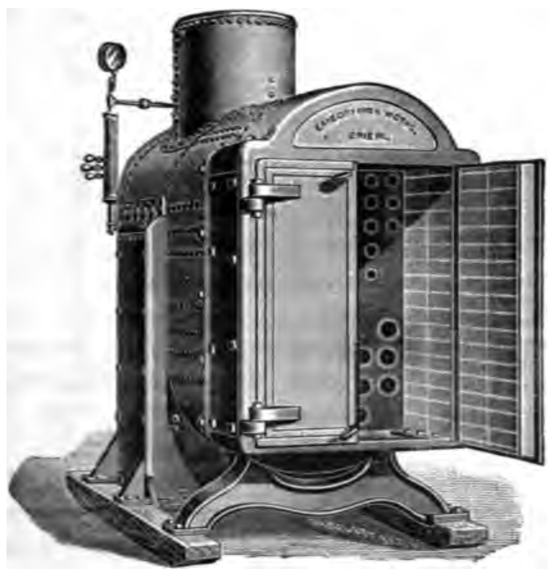

FIG. 82.

milder, and not nearly the amount of cold air blowing upon the boiler; and the steam generated was used by a 30 H.P. automatic engine exhausting into the air.

SPECIFICATIONS.

Number of Size	6	8
Horse Power, as usually rated	25	40
Diameter of boiler ... in inches	36	40
Length of furnace, inside ... "	44	58
Width " " ... "	36	40
Number of 3-inch tubes	80	88
Length " " used ... in feet	8	10
Number of 4-inch tubes	18	20
Length " " used ... in inches	49	64
Thickness of shell ... "	$\frac{1}{4}$	$\frac{9}{32}$
Thickness of tube sheets ... "	$\frac{3}{8}$	$\frac{3}{8}$
Diameter of dome ... "	24	26
Height of dome ... "	22	24
Diameter of stack ... "	18	20
Length of stack ... in feet	40	50
Length of boiler, over all, in feet ... about	11½	14
Width " " " ... "	4½	5
Height " " " ... "	8	8½
Weight of boiler ... about	7800	10800
Weight of fixtures ... "	1100	1700
Weight of boiler and fixtures ... about	8900	12000

The fourth trial was made under similar conditions to the third, with the exception that the exhaust of the engine was led into the base of the stack of the boiler, creating a forced draught.

The next series of trials were made on a Number 8, or 40 H.P. boiler, with the brick-lined combustion chamber. The first and second trials were conducted principally to determine the proper size of injector and safety valve necessary to supply all the water that the boiler would evaporate and exhaust into the air all the steam which the boiler would generate, without increasing the steam-pressure; and the results proved that neither the safety valve nor the injector was large enough, which were then changed to larger sizes. The data of these two trials are therefore omitted in column.

The third and fourth trials were made for a similar purpose, the injector and safety valves being again changed to still larger sizes.

The fifth and sixth trials were made with the radiating surface of the boiler covered with asbestos, to prevent radiation, with a still larger injector and safety valve than those used in the third and fourth trials.

The results of these several tests are tabulated below, and we think will fully explain themselves.

We think that these tests (with the excessively high evaporative efficiency shown) will demonstrate, to any one who will take the trouble to compare them with the total weights as given in the table of specifications above, that this form and type of boiler will give a very large and efficient total horse-power for a given weight and space occupied. This and the fact that the fire-box and back connection castings, together with their respective brick linings, can be readily detached from the boilers and shipped separately, makes this boiler one which can be transported easily to an otherwise nearly inaccessible point.

The figures shown by these tests, which give the actual horse-power of these boilers as averaging nearly 75% more than the builders' rating, strengthens the statement made above, and bears out more fully these assertions.

The writer was ably assisted in these tests by one of his apprentices, Mr. W. T. Foster, who made the observations and kept the tally and log sheets, and the results are represented graphically for each series in Figs. 82 to 89 herewith.

		Number 6, or 25 H.P. Boiler.				Number 8, or 40 H.P. Boiler.			
		1	2	3	4	3	4	5	6
2	Duration of test	5 hrs.	4 hrs.	5 hrs.	4 hrs.	8 hrs.	6½ hrs.	6 hrs.	6 hrs.
3	Grate surface	11 sq. ft.	11 sq. ft.	11	11	14.7	14.7	14.7	14.7
4	Water heating surface	296.1 "	296.1	296.1	296.1	452	458	452	458
5	Superheating "	0	0	0	0	0	0	0	0
5½	Radiating surface	75 sq. ft.	75 sq. ft.	75	75	116	116	116	116
6	Ratio of grate to heating surface	1 to 26.6	1 to 26.6	1 to 26.6	1 to 26	1 to 30	1 to 30	1 to 30	1 to 30
6½	" " radiating "	1 to 4	1 to 4	1 to 4	1 to 4	1 to 4	1 to 4	1 to 4	1 to 4
	AVERAGE PRESSURES.								
7	Steam-pressure by gauge	70.3	78.4	86.4	99.7	70.	69.	69.4	70.2
8	Absolute steam-pressure	84.6	92.8	100.7	114.	84.3	83.3	83.7	84.4
9	Atmospheric pressure per barometer	29.94	29.4	29.38	29.35	29.2	29.2	29.32	29.08
10	Forces of draught in cu. ft. per minute	....	....	....	....	....	....	....	....
	AVERAGE TEMPERATURES.								
11	Of external air	72°	51°	70°	63°	57°	42°	58.3°	57.3°
12	Of fire room	72°	51°	70°	63°	57°	42°	58.3°	57.3°
13	Of steam	....	....	....	....	....	....	....	....
14	Of escaping gases	....	....	....	....	....	....	....	....
15	Of feed water	63.7°	60.8°	66°	63°	62°	60°	56°	54°
	FUEL.								
16	Total amount of coal consumed	1094. lbs.	731 lbs.	1079.	1055.	2345.	2000.	2070.	1825.
17	Moisture in coal consumed	10%	10%	10%	10%	7%	7%	7%	10%
18	Dry coal consumed	934.6 lbs.	658. lbs.	971.1	949.5	2180.	1860.	1925.	1642.5
19	Total refuse, dry	111. lbs.	72. lbs.	171.	128.	400.	336.	340.	254. or 15%
20	Total combustible	873.6 lbs.	586 lbs.	800.1	821.5	1780.	1524.	1585.	1388.5
21	Dry coal consumed per hour	196.9 lbs.	164.4 lbs.	194.2	237.3	272.5	276.	320.8	273.7
22	Combustible consumed per hour	174.7 lbs.	146.4 lbs.	160.	205.3	222.5	236.	264.1	231.6
	RESULTS OF CALORIMETRIC TESTS.								
23	Quality of steam	.993	.984	.996	.984	.993	.992	.992	.992
24	Percentage of moisture	.7%	1.6%	.4%	1.6%	.7%	.8%	.8%	.8%
25	No. of degrees superheated	0	0	0	0	0	0	0	0
	WATER.								
26	Total weight of water pumped into boiler and apparently evaporated	5987. lbs.	4095. lbs.	5911. lbs	5958. lbs.	14800	13050	13200	12230
27	Water actually evaporated	5895. lbs.	4029. lbs.	5887.3 lbs.	5862.6 lbs.	14696	12945	13094	12231
28	Equivalent water evaporated into dry steam from and at 212°	6983.	4799.5	7003.8	7011.7	17475	15436	15651	14645

29	Equivalent total heat derived from fuel in B. T. U.	6,733,947	4,623,788	6,745,668	6,752,628	16,816,632	14,883,675	15,062,028	14,093,781
30	Equivalent water evaporated into dry steam from and at 212° per hour	1386	1199.8	1401	1752.9	2184	2285	2608.5	2440.8
	ECONOMIC EVAPORATIONS.								
31	Water actually evaporated per lb. of dry coal	6.	6.12	6.06	6.17	6.7	6.9	6.8	7.44
31½	" " " " " combustible	6.7	6.87	7.35	7.14	8.3	8.5	8.25	8.8
32	Equivalent water evaporated per lb. of dry coal from and at 212°	7.	7.44	7.21	7.38	8.	8.3	8.1	8.9
33	" " " " " combustible from and at 212°	7.9	8.2	8.75	8.53	9.8	10.1	9.87	10.5
33½	Equivalent water evaporated per lb. of combustible from and at 212°, corrected for radiation	8.3	8.6	9.18	8.95	10.4	10.7	Boiler	covered.
	COMMERCIAL EVAPORATION.								
34	Equivalent water evaporated per lb. of dry coal, with ⅙ refuse, at 70 lbs. and from temperature of 100°	5.74	5.93	6.34	6.18	7.1	7.3	7.1	7.6
	RATE OF COMBUSTION.								
35	Dry coal actually consumed per sq. ft. of grate surface per hour	17.9	14.9	17.64	21.5	18.5	18.7	21.8	18.8
36	Consumption of dry coal per hour. Coal assumed with ⅙ refuse per sq. ft. of grate surface	19.	15.9	17.4	22.4	18.1	18.4	21.5	18.6
37	Consumption of dry coal per hour. Coal assumed with ⅙ refuse per sq. ft. of heating surface	.71	.6	.65	.84	.59	.6	.7	.61
38	Consumption of dry coal per hour. Coal assumed with ⅙ refuse per sq. ft. of least area for draught	149.1	126.	137.1	175.9	127.	129.	150.7	132.3
	RATE OF EVAPORATION.								
39	Water evaporated from and at 212° per sq. ft. of heating surface per hour	4.73	4.09	4.78	5.98	4.8	5.	5.7	5.4
40	Water evaporated per hour from 100° into steam of 70 lbs. per sq. ft. of grate surface	109.6	94.8	101.68	138.6	129.	135.	154.3	144.4
41	Water evaporated per hour from 100° into steam of 70 lbs. per sq. ft. of heating surface	4.1	3.55	3.8	5.2	4.2	4.3	5.	4.6
42	Water evaporated per hour from 100° into steam of 70 lbs. per sq. ft. of least area for draught	861.5	745.4	798.9	1089.	904.	946.	1079.	1010.9
	COMMERCIAL HORSE-POWER.								
43	On basis of 30 lbs. of water per hour evaporated from temperature of 100° into steam at 70 lbs.	40.1	34.7	40.6	50.8	63.	66.	75.3	70.4
43½	Horse-power corrected, allowing for radiation in boiler	42.	36.4	42.6	53.	67.	70.	....	....
44	Horse-power, builders' rating	25.	25.	25.	25.	40.	40.	40.	40.
45	Per cent. developed above rating	60%	39%	60%	100%	67½%	75%	88%	75%
	Extra:								
33	Equivalent water evaporated per lb. of combustible from and at 212°, corrected for injector	....	....	....	....	10.4	10.7	10.4	11.1
33½	Equivalent water evaporated per lb. of combustible, corrected for radiation and injector	....	....	....	....	11.	11.8	....	....
43½	Horse-power, corrected, allowing for radiation and injector	....	....	....	....	67.5	70.5	80	75

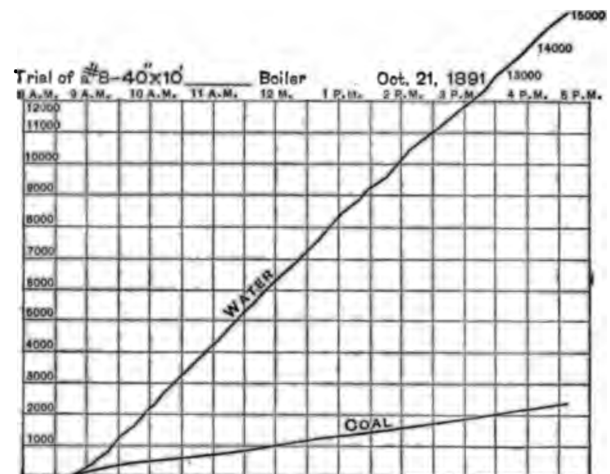

Fig. 82.

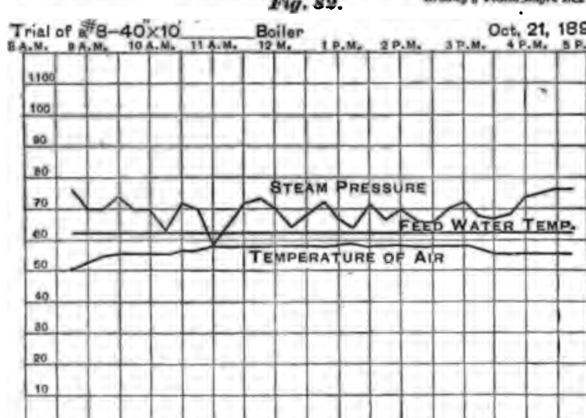

Fig. 83.

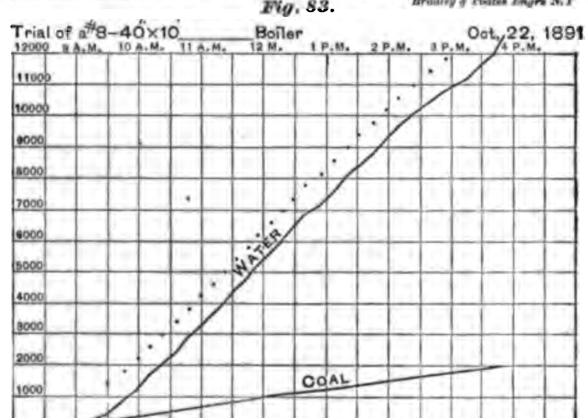

Fig. 84.

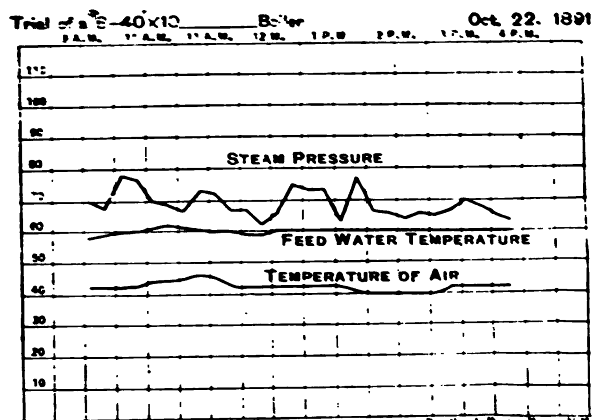

Fig. 85.

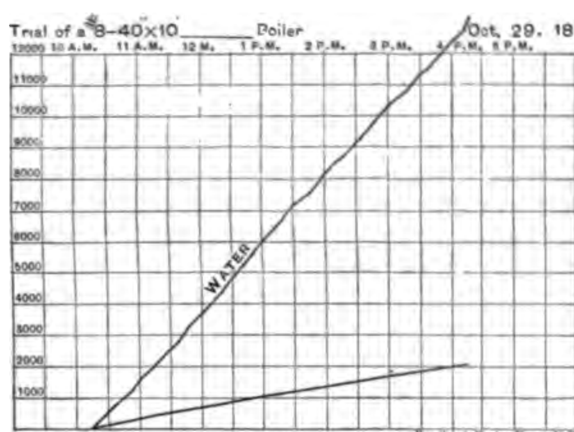

Fig. 86.

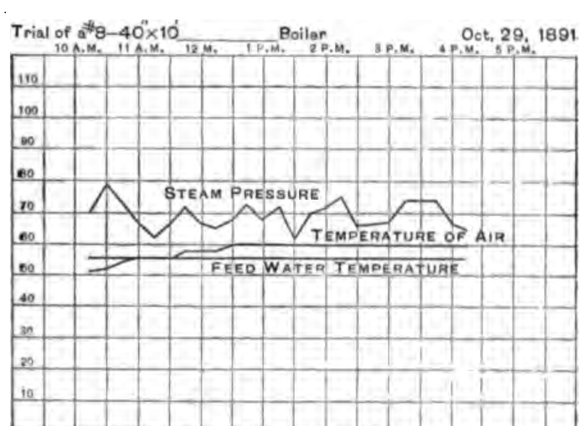

Fig. 87.

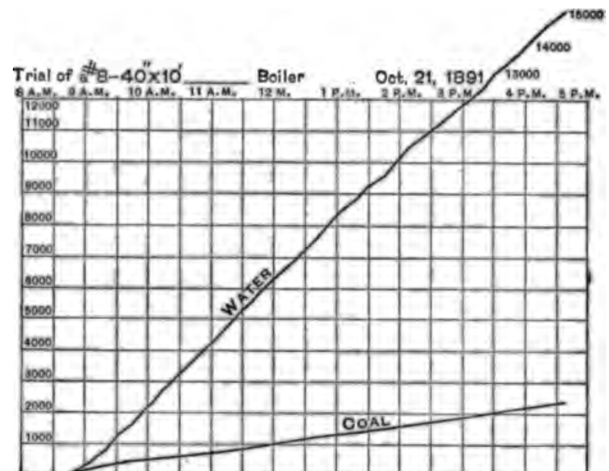

Fig. 82.

Bradley & Poates Engrs N.Y

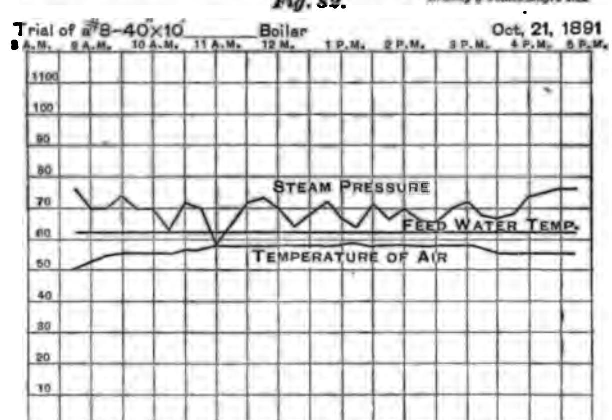

Fig. 83.

Bradley & Poates Engrs N.Y

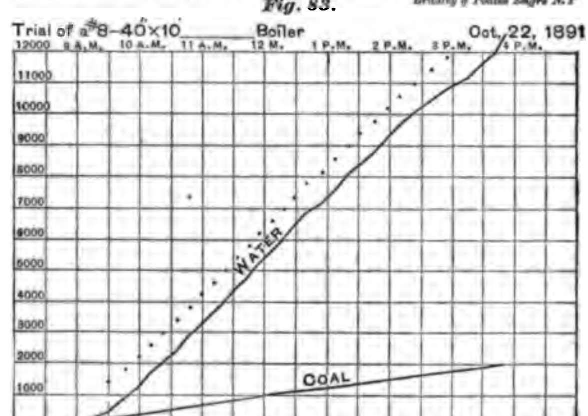

Fig. 84.

Bradley & Poates Engrs N.Y

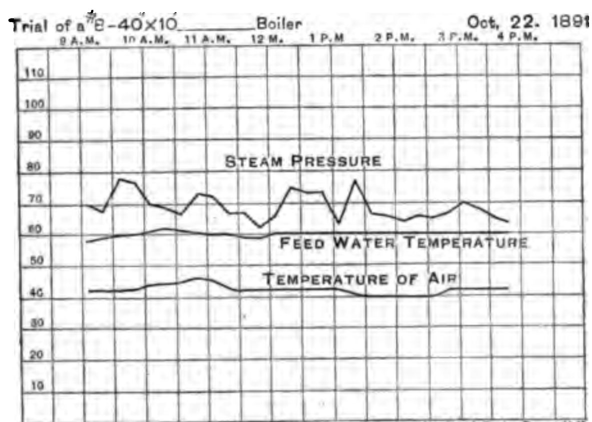

Fig. 85. Bradley & Poates Engrs N.Y

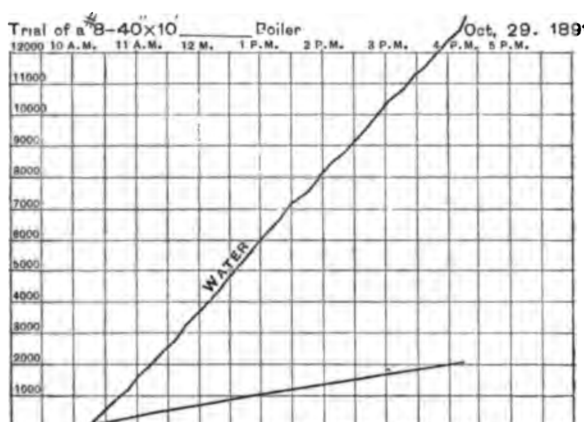

Fig. 86. Bradley & Poates Engrs N.Y

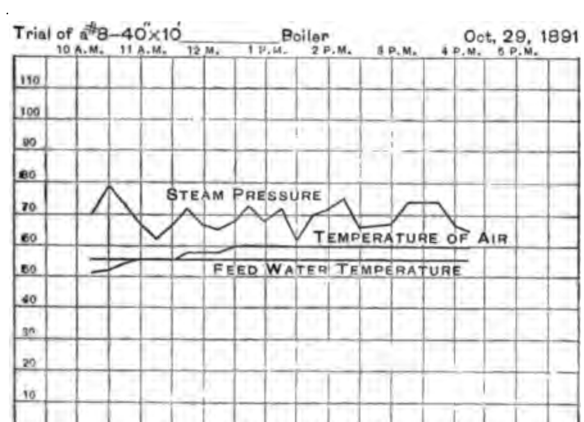

Fig. 87. Bradley & Poates Engrs N.Y

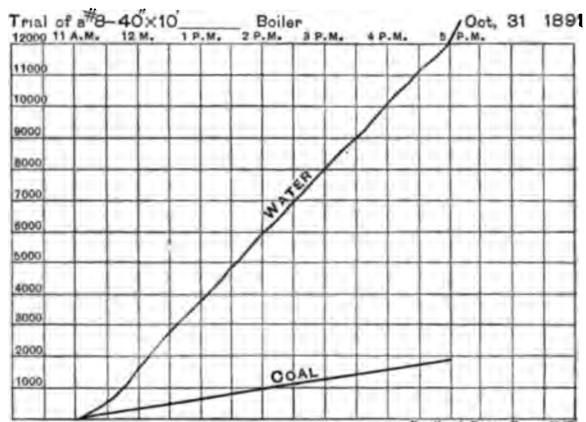

Fig. 88.

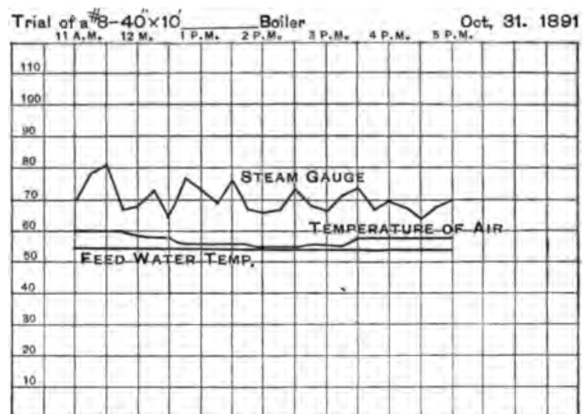

Fig. 89.

DISCUSSION.

Prof. D. S. Jacobus.—How is the line marked 33 extra calculated? This line is the equivalent of water evaporating per pound of combustible from and at 212° corrected for injector. By the ordinary method of calculation we have line 33 in the main table. 33 is equivalent to water evaporated from and at 212°, not corrected for injector. Taking the last four tests, we have 9.8 and 10.1 and so on. But on coming down to line 33 extra, the 9.8, after being corrected for the injector, becomes 10.4; 10.1 becomes 10.7; and 9.87 becomes 10.4. Line 33 extra is therefore about $\frac{1}{16}$ more than line 33, which is about the steam used by the injector. Line 33 is simply the actual water evaporated per pound of combustible, multiplied by the factor of evaporation for the temperature of water entering the injector. For example, in test No. 4 of the 40-H. P. boiler, the temperature of water entering an injector is 60°, and the steam pressure 69 lbs. The total heat of steam at 69 lbs. pressure is about 1210, so that the factor of evaporation is 1.19, which multiplied into the actual evaporation per pound of combustible, or 8.5, gives 10.1, which is the figure given in line 33. If my suppositions in regard to the method of conducting the tests is correct, we cannot take and afterward add the steam used by the injector and increase this figure to 10.7, but must stop at the figure given in line 33.

I wish to know if there were any peculiar conditions existing in the test which I have overlooked in making my remarks.

Mr. F. A. Scheffler.—Professor Jacobus has opened the same question which I wanted to bring up. I do not see how the equivalent evaporation of water per pound of combustible from and at 212° can be corrected for injector and show a higher evaporation than what it actually showed in the test. An explanation from Mr. Webber would be very satisfactory, I think. My idea of this is that it is a mistake. I think that this correction referred to under the head of "extra" evidently means a certain amount of water was caught at the injector at the time it was fed, which was supposed to have gone into the boiler, but did not enter, and should have been *deducted* from Nos. 33, 33½, and 43½. The test of No. 6 boiler, which I understand was made without the smoke-box being lined with fire-brick the same as it is lined in the fire-box, corresponds almost exactly with two tests which I made of the same boiler under the same conditions. The test on No. 8 boiler shows that lining the smoke-box adds considerably to the efficiency.

*Mr. Wm. O. Webber.**—Replying to Professor Jacobus and Mr. Scheffler, the extras Nos. 33, 33½, and 43½ were added to the test somewhat against the writer's judgment, and merely to check some tests made by other parties on other portable boilers, who figured their results in this manner and claimed it was the proper way to do. In the writer's opinion, all of the steam used by the injector is returned to the boiler in the way of heat and water, and that very little of the heat is lost by radiation.

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* Author's closure.

The amount of steam used by the injector was calculated from the pressure of steam and the size of the steam orifice in the injector by the regular formulæ.

There were no peculiar conditions existing at these tests, except that both boilers stood in the open air and were exposed to our lake wind all the time ; hence the final tests of the No. 8, with the boiler covered, and the corrections made in 33½ for radiation, which the last tests show to be substantially correct. Items Nos. 33 and 33½ are the gist of the whole series of tests.

CCCCXC.*

AN EXPERIMENTAL LOCOMOTIVE.

BY W. F. M. GOSS, LAFAYETTE, INDIANA.

(Member of the Society.)

In the fall of 1890, the authorities of Purdue University decided to erect and equip a new building for advanced work in engineering, and soon after it was determined to purchase a locomotive of a size which should well represent its class, and to mount it in the laboratory for experimental purposes.

The plan of mounting, in its inception, involved (1) supporting wheels carried by shafts running in fixed bearings, to receive the locomotive drivers and to turn with them; (2) brakes which should have sufficient capacity to absorb continuously the maximum power of the locomotive, and which should be mounted on the shafts of the supporting wheels; and (3) a traction dynamometer of such form as would serve to indicate the horizontal moving force and at the same time allow but a slight horizontal motion of the engine on the supporting wheels. It was believed that a locomotive thus mounted could be run either ahead or aback under any desired load and at any speed; that while thus run, its performance could be determined with a degree of accuracy and completeness far excelling that which it is possible to secure under ordinary conditions of the road; and that the whole apparatus would be extremely valuable to students in steam engineering. It was not thought that every condition of the track would be perfectly met, but it was expected that the results obtained would prove valuable in extending a knowledge of locomotive performance.

The locomotive was ordered of the Schenectady Locomotive Works, in May, 1891. It is of the eight-wheel type, with 17 x 24-inch cylinders and 63-inch drivers. Its weight is 85,000 pounds, 56,000 pounds being on the drivers.

The details of the mounting were designed during the summer

* Presented at the San Francisco Meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the Transactions.

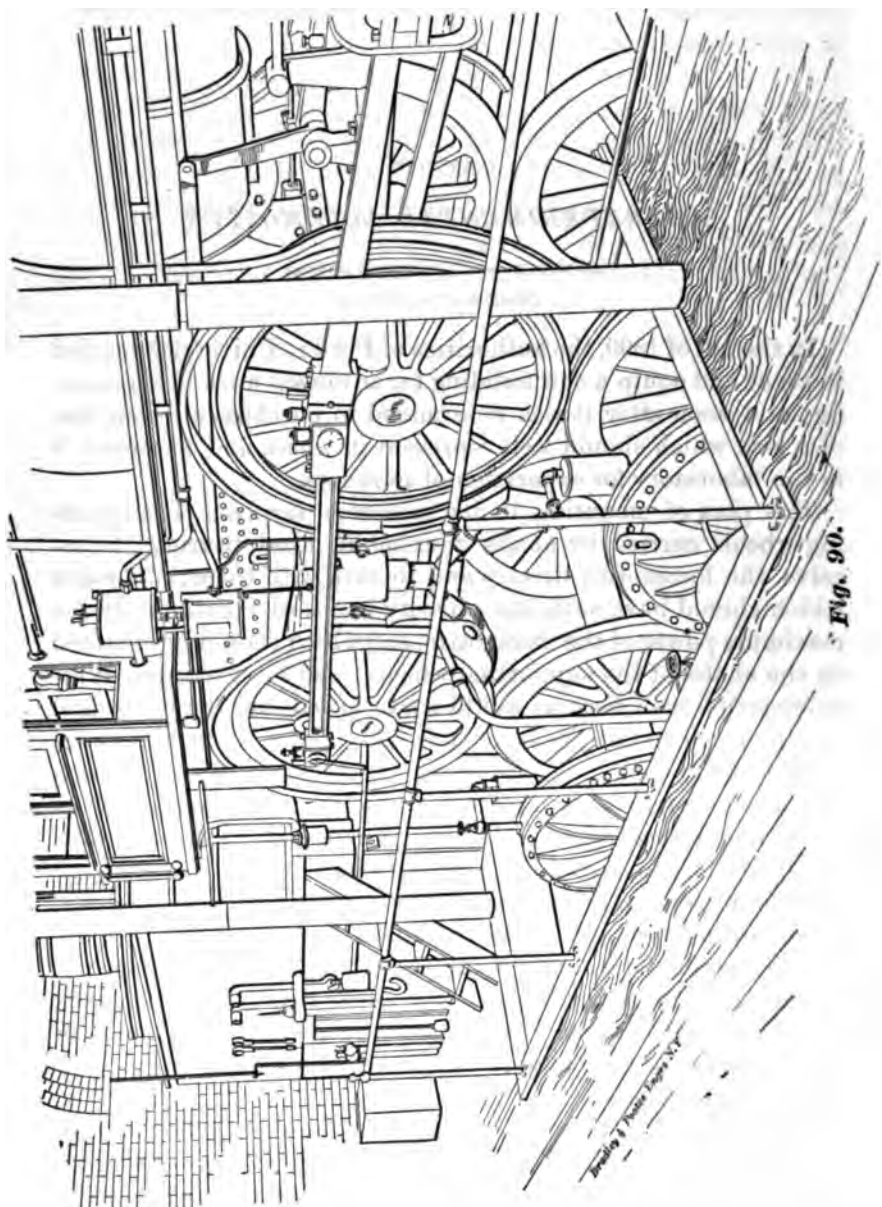


Fig. 90.

of 1891, and by the following September, when the engine was received, the supporting wheels were in place. The mounting is now (January, 1892) practically complete. Fig. 90 is from a photograph giving a partial view of the drivers, supporting wheels, and brakes; Fig. 91 shows a complete general view, in elevation, of the locomotive and its mounting machinery, and Fig. 92 shows in plan, the mounting machinery only.

Reference to these figures will show that there is a heavy rubble foundation, capped at convenient points with cut stones rising 10 inches above grade line. Upon these stones are placed well-seasoned oak timbers arranged in two lines, each composed of three lengths, 4 x 14 inches. The timbers of each line are well bolted to each other and are securely anchored to the foundation. Upon the timbers rest the bearings of the supporting shafts, which are thus given 14 inches of oak to constitute an element of elasticity between them and the foundation.

The supporting wheels are of the same diameter with the locomotive drivers, and are in other respects similar, save that the cranks and counter-weights are omitted. Their faces are turned flat, with the inside edge rounded as in a rail.

The four friction brakes which provide the load for the supporting shafts, and which are shown in position in Figs. 91 and 92, were designed on the principle developed by Professor George I. Alden, and already described by him.* The details of the brake design under consideration, will be given further on, but it is important to state here that the principle as developed by Professor Alden provides extensive rubbing surfaces of cast-iron and copper. Excessive wear is prevented by thorough lubrication. The intensity of the brake action is controlled by water pressure, by which means the rubbing surfaces are brought into contact more or less intimate; and the heat evolved is carried off by water circulation.

By referring to the plan and elevation, Figs. 91 and 92, it will be seen that there is no provision for weighing the load at the brakes, where, instead of a weighted lever, anchor rods are used to secure the case of the brakes to the foundation. The entire load shows itself at the dynamometer connected with the draw-bar of the locomotive. The water supply for the brakes is given by a 3-inch pipe. It passes first a balanced valve *A*, around which there is a by-pass controlled by valve *B*. From the tee *C* the pipe is branched for the several brakes, 2½-inch piping serving for two brakes, and 2-inch for each individual brake.

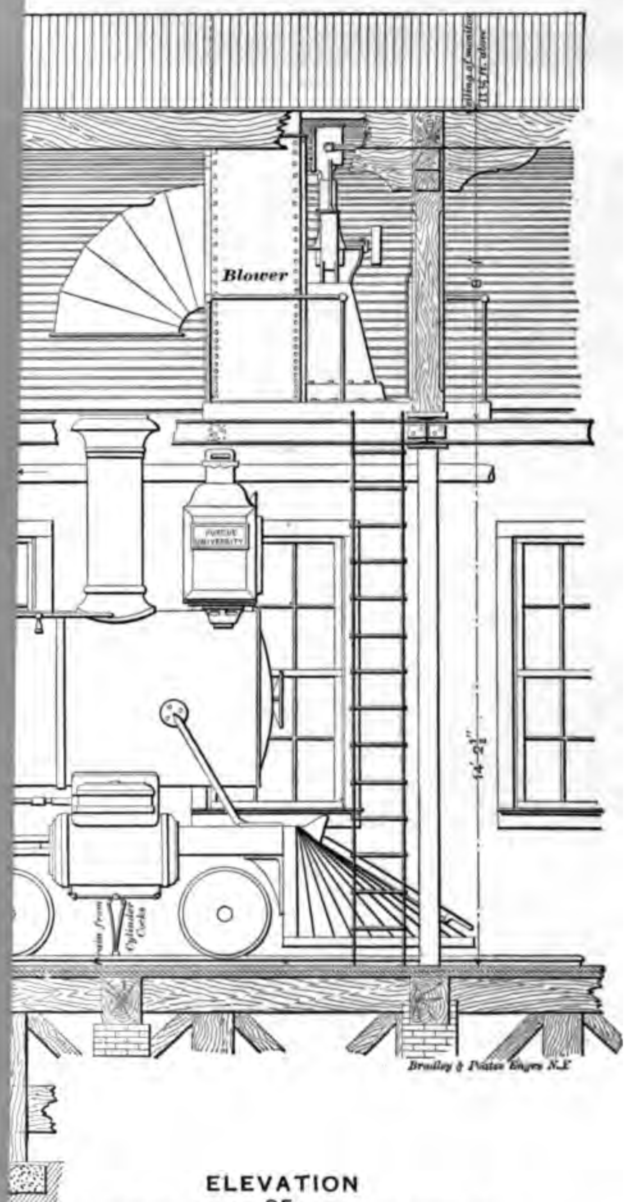
* Vol. XI., p. 939 *et seq.* Transactions of the Society.

Valves D are provided in the supply pipe for each brake, so that any one may be entirely cut out or have its action modified to any desired extent. The water from each brake is returned by a separate pipe to a point E, where valves are provided by which the amount of water allowed to pass each brake is regulated. The water flows from these valves in an open stream and is finally discharged into the sewer. The water pressure within each brake is indicated by one of the four gauges at F (Fig. 91).

The balanced valve A has its spindle connected with one of the levers of the dynamometer in such a way that its position is controlled by the pull, or if backing, by the push exerted by the locomotive. Thus, suppose the locomotive is in motion and the outlet valves at E adjusted to allow the passage of water enough to keep down the temperature of the brakes, and suppose the pull of the locomotive is such as will bring the weighted lever of the dynamometer to its mid-position, then there will be a definite opening of the balanced valve, and a definite water pressure within the brakes will result. If, now, for any reason the weighted lever falls, there will be a corresponding increase in the opening of the balanced valve, and, hence, an increase of water pressure within the brakes. The greater pressure will result in greater resistance to the moving of the supporting wheels, and, hence, in a stronger pull of the locomotive on the dynamometer, and this increased pull will tend to lift the weighted lever again. Similarly, if for any reason the pull of the locomotive is sufficient to raise the lever beyond its central position, the balanced valve will respond by reducing the water pressure within the brakes; the tractive force of the engine will decrease and the dynamometer lever will fall. When, therefore, it is desired to increase the load on the locomotive it is necessary only to place the additional weight upon the lever of the dynamometer, and the corresponding increase in the load is furnished automatically by the brakes.

By a proper adjustment of the by-pass valve B the lever may be made to stand exactly in its central position.

The traction dynamometer is made up of a system of levers. The first lever in the system, shown by dotted outline at G in Figs. 91 and 92, has a direct connection with the locomotive draw-bar. The last lever, shown at H, carries an ordinary weight-holder. The whole arrangement is such that, whether



ELEVATION
OF
LOCOMOTIVE MOUNTING

ineering Laboratory Purdue University.
Lafayette, Ind.

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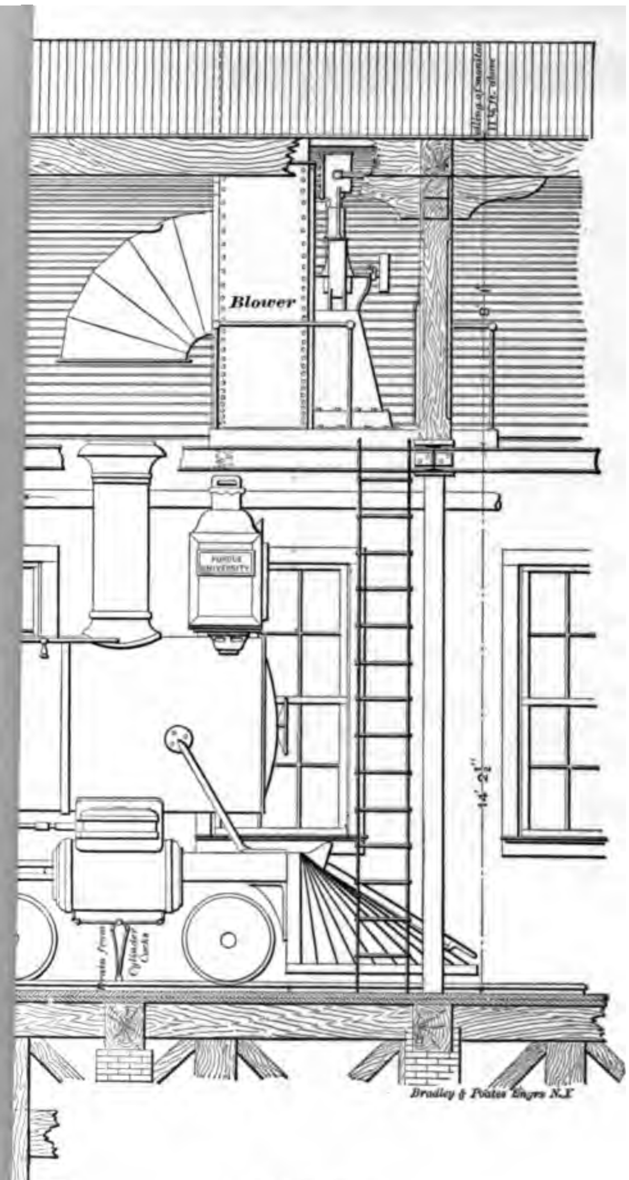

ELEVATION
OF
LOCOMOTIVE MOUNTING

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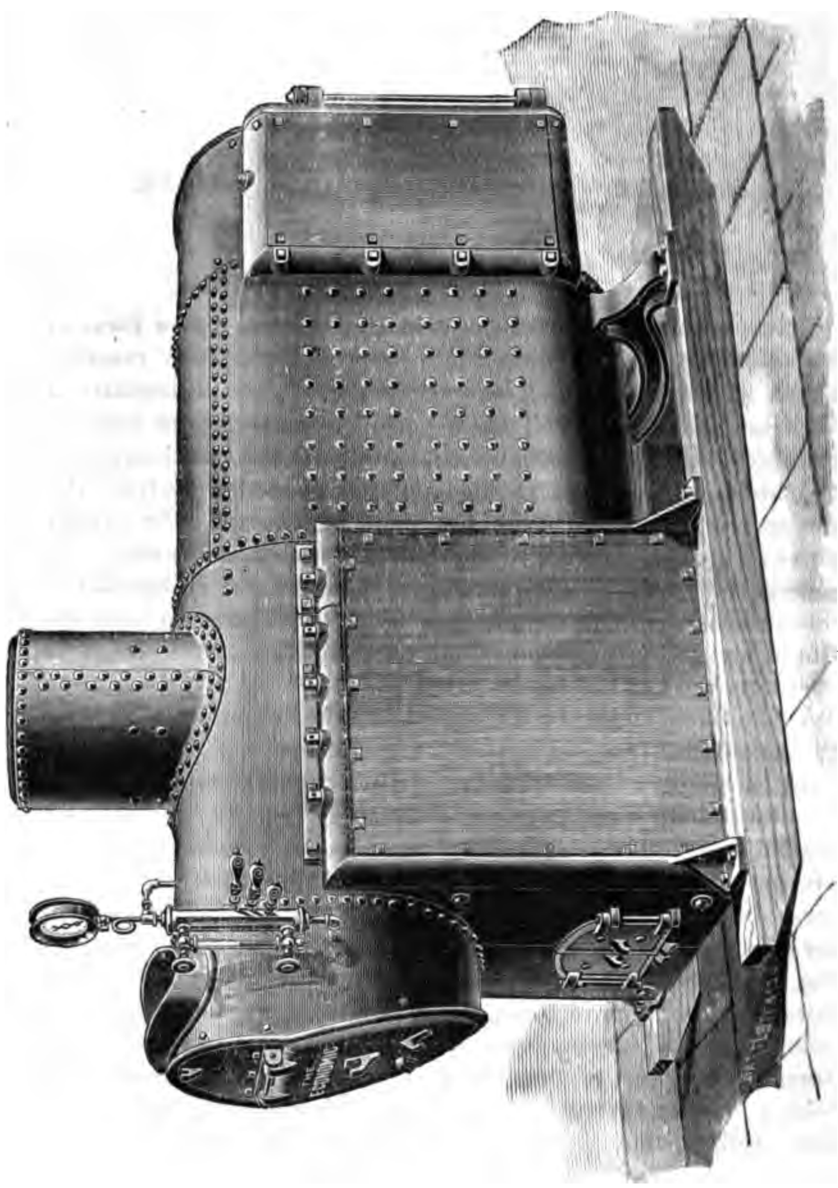
Lafayette, Ind.

brick being held in place by vertical iron rods which are protected from the fire and can be removed and replaced when neces-



sary. These iron rods are square in cross section, and are set at an angle to fit the V-notch in the ends of the fire brick, and the

the engine moves ahead or aback, the stress is transmitted by the draw-bar and its value is shown by the weight necessary to balance the lever *H*.

The dynamometer levers are carried by a heavy framework, which is well secured to the locomotive foundation and to surrounding parts of the building. The character of the framing is but imperfectly shown by the drawings.

While the draw-bar is the only active agent by which the horizontal movement of the locomotive is controlled, there is ample provision of chains and buffers to check any excessive movement which may chance to occur.

Above the levers of the dynamometer, a floor is laid which chiefly serves the purposes of a tender. It gives room for a tank from which the locomotive injectors draw their supply and for the storage of a limited quantity of coal. Connected with the water tank is a glass gauge *I*, and above the tank is a weighing barrel through which the tank receives its supply. Scales for weighing fuel are also given a place on the "tender floor."

The three counters at *J* are connected, respectively, with the rear driving axle and to each of the two supporting shafts. These give a ready means for determining the speed of the engine, and the per cent. of slip between the drivers and their supporting wheels.

The telltale at *K* shows the position of the locomotive relative to the supporting wheels. The board *a* is fastened to the locomotive and consequently moves with it; the rod *b* is connected at one end to an iron column as a fixed point, and at the other end to the pointer *d*. This pointer is pivoted to the board *a* at *e*, so that any backward or forward movement of the locomotive is greatly multiplied by a similar movement of the lower end of the pointer *d*.

A tangent-wheel and screw are provided at *L* for the purpose of turning by hand the forward supporting shaft, and hence the engine, whenever it may be desired to do so, as, for example, for convenience in valve-setting. When not in use, the screw may be disengaged.

The truck-wheels of the engine rest upon light rails which are fixed at the level of the laboratory floor and extend in front of the engine a distance sufficient to allow the whole machine to be moved forward off the supporting wheels, whenever the latter may need to be taken out for repairs.

A Sturtevant $4\frac{1}{4} \times 6\frac{3}{4}$ steam blower located above the engine (Fig. 91) but not in pipe connection with it, removes from the room everything that is given out of the locomotive stack, without changing, materially, the draft conditions under which the locomotive is worked.

The cylinder cocks and the over-flow pipes from the injectors, are all in loose connection with the sewer. The discharge from the over-flow pipes may be directed into weighing barrels.

The boiler is in pipe connection with the fixed boiler which supplies steam for general use in the laboratory, so that the locomotive may be used to supply steam to other apparatus, or the fixed boiler may be used to supply the locomotive. In the latter case the locomotive boiler is drained by the steam trap *M*; by this means the boiler is freed from water, and as

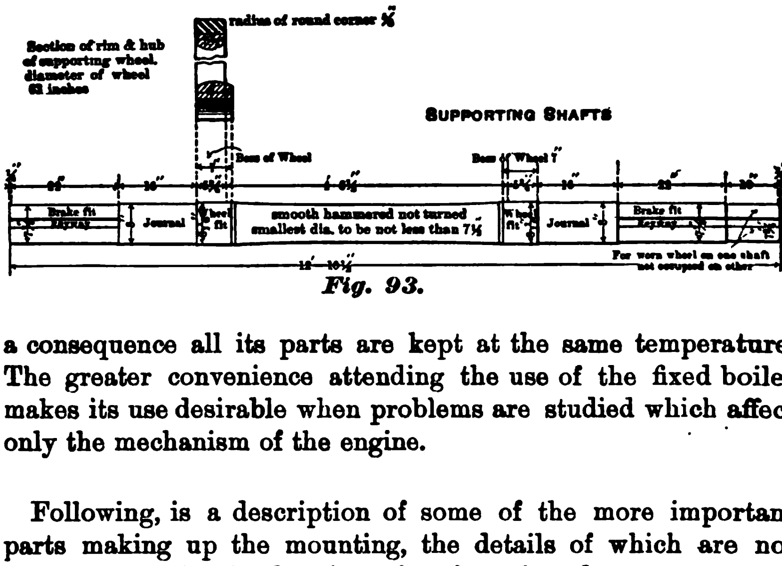


Fig. 93.

a consequence all its parts are kept at the same temperature. The greater convenience attending the use of the fixed boiler makes its use desirable when problems are studied which affect only the mechanism of the engine.

Following, is a description of some of the more important parts making up the mounting, the details of which are not clearly shown by the drawings thus far referred to.

THE SUPPORTING SHAFTS are of hammered iron and of the form and dimensions shown by Fig. 93. The bearings for these shafts are 8″ in diameter and 16″ long. Each bearing is fitted to a cast-iron plate 14″ x 36″ x 2″, which plate, in turn, is carefully bedded upon the oak timbers (Figs. 91 and 92). The whole is made secure by four bolts passing through the bearing, plate and timber.

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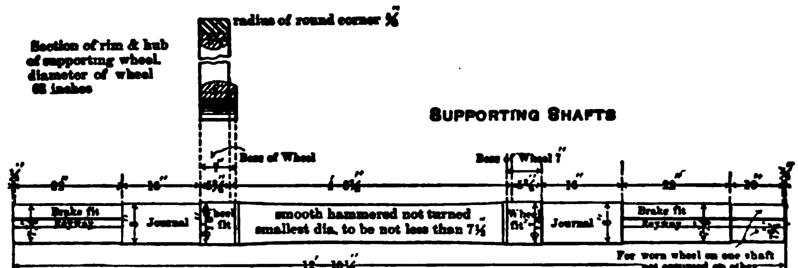

Fig. 93.

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29 Equivalent total heat derived from fuel in B. T. U.	6,733,947	4,623,788	6,745,668	6,752,028	16,816,632	14,833,675	15,062,028	14,093,781
30 Equivalent water evaporated into dry steam from and at 212° per hour	1886	1199.8	1401	1752.9	2184	2285	2606.5	2440.8
ECONOMIC EVAPORATIONS.								
31 Water actually evaporated per lb. of dry coal	6.	6.12	6.06	6.17	6.7	6.9	6.8	7.44
31½ " " " " " combustible	6.7	6.87	7.35	7.14	8.8	8.5	8.25	8.8
32 Equivalent water evaporated per lb. of dry coal from and at 212°	7.	7.44	7.21	7.88	8.	8.3	8.1	8.9
33 " " " " combustible from and at 212°	7.9	8.2	8.75	8.53	9.8	10.1	9.87	10.5
33½ Equivalent water evaporated per lb. of combustible from and at 212°, corrected for radiation	8.8	8.6	9.18	8.95	10.4	10.7	Boiler	covered.
COMMERCIAL EVAPORATION.								
34 Equivalent water evaporated per lb. of dry coal, with ⅙ refuse, at 70 lbs. and from temperature of 100°	5.74	5.98	6.34	6.18	7.1	7.3	7.1	7.6
RATE OF COMBUSTION.								
35 Dry coal actually consumed per sq. ft. of grate surface per hour	17.9	14.9	17.64	21.5	18.5	18.7	21.8	18.8
36 Consumption of dry coal per hour. Coal assumed with ⅙ refuse per sq. ft. of grate surface	19.	15.9	17.4	22.4	18.1	18.4	21.5	18.6
37 Consumption of dry coal per hour. Coal assumed with ⅙ refuse per sq. ft. of heating surface	.71	.6	.65	.84	.59	.6	.7	.61
38 Consumption of dry coal per hour. Coal assumed with ⅙ refuse per sq. ft. of least area for draught	140.1	126.	137.1	175.9	127.	129.	150.7	132.3
RATE OF EVAPORATION.								
39 Water evaporated from and at 212° per sq. ft. of heating surface per hour	4.73	4.09	4.78	5.98	4.8	5.	5.7	5.4
40 Water evaporated per hour from 100° into steam of 70 lbs. per sq. ft. of grate surface	109.6	94.8	101.68	138.6	129.	135.	154.8	144.4
41 Water evaporated per hour from 100° into steam of 70 lbs. per sq. ft. of heating surface	4.1	3.55	3.8	5.2	4.2	4.3	5.	4.6
42 Water evaporated per hour from 100° into steam of 70 lbs. per sq. ft. of least area for draught	861.5	745.4	798.9	1089.	904.	946.	1079.	1010.9
COMMERCIAL HORSE-POWER.								
43 On basis of 30 lbs. of water per hour evaporated from temperature of 100° into steam at 70 lbs.	40.1	34.7	40.6	50.8	63.	66.	75.3	70.4
43½ Horse-power corrected, allowing for radiation in boiler	42.	36.4	42.6	53.	67.	70.	....	....
44 Horse-power, builders' rating	25.	25.	25.	25.	40.	40.	40.	40.
45 Per cent. developed above rating	60%	39%	60%	100%	67½%	73%	88%	75%
Extra:								
33 Equivalent water evaporated per lb. of combustible from and at 212°, corrected for injector	....	....	....	....	10.4	10.7	10.4	11.1
33½ Equivalent water evaporated per lb. of combustible, corrected for radiation and injector	....	....	....	....	11.	11.3	....	....
43½ Horse-power, corrected, allowing for radiation and injector	....	....	....	....	67.5	70.5	80	75

the opening *R*, sufficient water being allowed to pass to carry away the heat resulting from the friction. To prevent the water pressure within the brake from spreading the sides of the case, the rings *S* are fitted over a feather, to the hub of the moving disc, and are held to their place by nuts which screw up to a shoulder.

When the brake is in use, all clearance space between the copper plates and about the moving cast-iron disc, is filled with oil. The distribution of the oil is secured by 32 radial grooves on each face of the cast-iron disc (Ele., Fig. 94), and by a spiral groove extending from the inner edge to the circumference of the rubbing surfaces of the disc, with a pitch of about 4 inches. The entire rubbing area of the cast-iron disc is thus

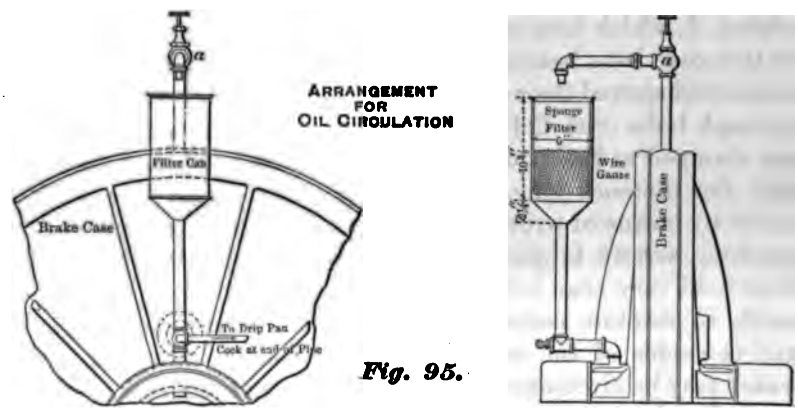

Fig. 95.

split up into surfaces, the length or breadth of which in no case greatly exceeds 4 inches. The spiral oil-way gives these surfaces a position such that in passing a given point on the copper plate their alignment is continually changing, which fact, it is believed, greatly assists in the distribution of the oil. The radial grooves also give rise to a slight pumping action, by means of which the oil may be kept in circulation between the circumference and center of the brake. Provision for this circulation is made as shown by Fig. 95; the oil passes the valve and piping *a* from the highest point in the brake, and is received by the filter-can, and is thence delivered to the center of the brake. The filter-can also serves the purpose of a supply reservoir, and always contains surplus oil when the brake is in action. This circulation helps to maintain the oil at a uniform

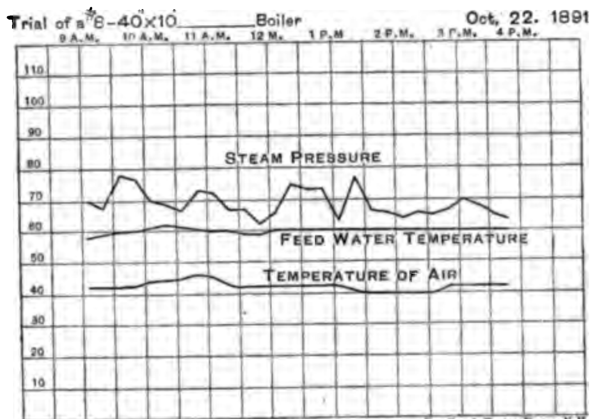

Fig. 85.

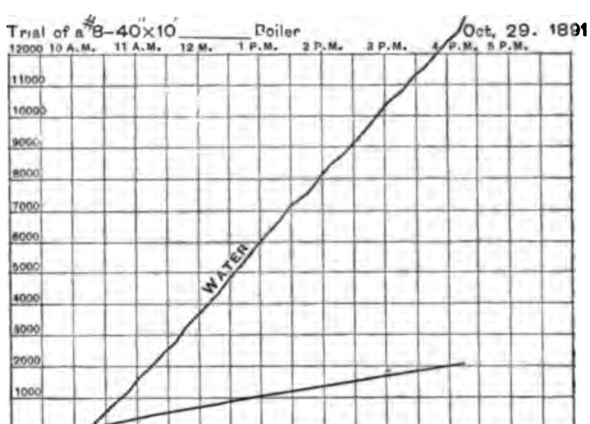

Fig. 86.

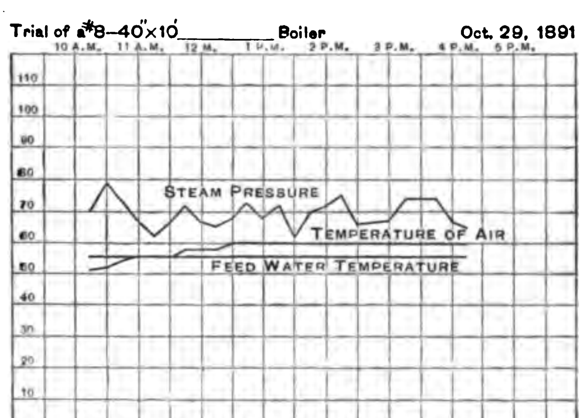

Fig. 87.

the opening *R*, sufficient water being allowed to pass to carry away the heat resulting from the friction. To prevent the water pressure within the brake from spreading the sides of the case, the rings *S* are fitted over a feather, to the hub of the moving disc, and are held to their place by nuts which screw up to a shoulder.

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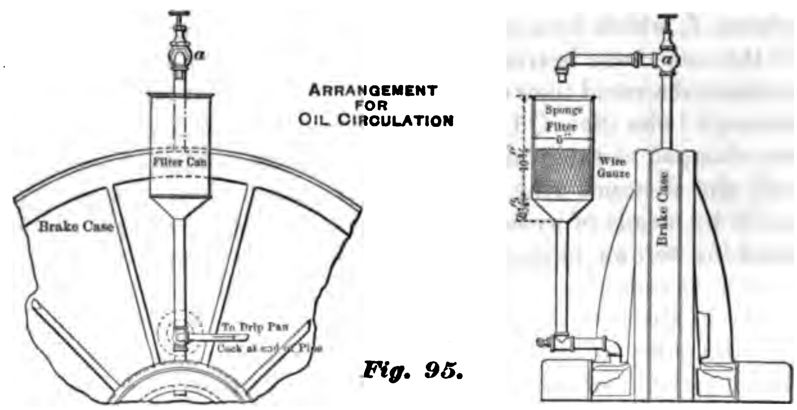

ARRANGEMENT FOR OIL CIRCULATION

Fig. 95.

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As preliminary to the design of these brakes, some experiments were made on a 21-inch disc brake while driven at speeds varying from about 300 to 450 revolutions per minute, from which experiments it appeared that lubrication could be maintained with certainty under a water pressure of 40 lbs. per square inch, and this pressure was adopted as the maximum to be used in the design of the larger brakes herein described. It also appeared that the *apparent coefficient of friction** varied from 2.7% to over 4%: depending largely on the viscosity of the oil used and upon the temperature of the brake. It was thought that 3.5% would be a safe coefficient for moderate speeds, and this factor was accordingly used. It was also decided to use four brakes in preference to two.

The locomotive drivers are 63 inches in diameter, and the sum of the moments about the two driving axles when the engine is exerting a tractive force of 16,000 lbs. (assumed to be maximum) is therefore $16{,}000 \frac{63}{2 \times 12} = 42{,}000$ foot pounds, which, since the supporting wheels are of the same diameter with the drivers, is the moment under which the brakes on the supporting shaft must work. Each of the four brakes, therefore, must be capable of acting under a moment of 10,500 foot pounds. The disc of the brakes constructed, and shown by Fig. 94, is 56 inches in diameter. The area in effective contact with the copper plates on either side is represented by an annulár surface having its outer radius equal to 28 inches, and its inner radius equal to 10 inches. These dimensions, in connection with the assumed coefficient of friction and the assumed maximum water pressure, give a calculated moment slightly in excess of the required 10,500 foot pounds.

THE TRACTION DYNAMOMETER is shown by Fig. 96. The outline plan and elevation in this figure show the relative position of the different levers when in place. The details of each lever are shown by the remainder of the plate.

*By apparent coefficient of friction is meant that factor which is obtained by assuming that the entire moment of the brake is due to the friction between the rubbing surfaces of the moving disc and the copper plates. Since there are other rubbing surfaces it is clear that the apparent coefficient of friction is larger than the actual coefficient.

By referring first to the main lever A, the following description will be seen to apply. This lever is supported by a round steel pin which connects it with the cast plate E, and which in turn is securely bolted to heavy timbers, forming a part of the frame-work behind the locomotive. The round rod F is an extension of the locomotive draw-bar, and the pull, or push, of the locomotive is exerted along the line of its axis. Stress is transmitted from this rod to the blocks f by nuts as shown, the fit between the nuts and the blocks being spherical (detailed view, Fig. 96) to allow slight changes in the direction of F. The blocks f are connected with the short arm of the lever A by the round steel pins g, and the long arm of this lever is engaged by the hook l, which is free to move within the link k. To illustrate the action of this part of the dynamometer, let it now be assumed that the pull of the locomotive on the draw-bar F is ahead, that is in the direction of the upper arrow; then the stress on the draw-bar will result in a tendency to raise the hook l, and the lever B with its weight will serve only as a counter-balance to the lever A. But, if it be assumed that the locomotive is working aback, that is, that the stress in the draw-bar is in the direction of the lower arrow, the long arm of the lever A with the hook l will tend to fall. Motion of l in this direction, however, is soon arrested by the link k, which by virtue of its connection with the lever B rises as l falls, until it engages the check-nuts i; these are thus made a means of transmitting the stress to the hook l, which as before will move upward, while the shackle at the upper end of the hook l is entirely free from stress. It will thus be seen that whether the locomotive is working in forward or in backward gear, its tractive force is made manifest by an upward movement of the hook l, from which point two simple levers complete the dynamometer. The amount of lost motion between the check-nuts i and the link k is very small.

The direction of the lever C, which connects with hook l, is at right angles with that of the lever A, its purpose being to bring the last lever D of the system out from behind the locomotive. The lever D carries a weight holder arranged to receive twenty 10-lb. weights. The ratio of the whole system is as 1 to 100; each weight, therefore, that is balanced at the weight holder represents a tractive force exerted by the locomotive of 1,000 lbs.

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AN EXPERIMENTAL LOCOMOTIVE.

BY W. F. M. GOSS, LAFAYETTE, INDIANA.
(Member of the Society.)

In the fall of 1890, the authorities of Purdue University dec
to erect and equip a new building for advanced work in engin
ing, and soon after it was determined to purchase a locom
of a size which should well represent its class, and to mou
in the laboratory for experimental purposes.

The plan of mounting, in its inception, involved (1) sup
ing wheels carried by shafts running in fixed bearings, to
ceive the locomotive drivers and to turn with them; (2) br
which should have sufficient capacity to absorb continuously
maximum power of the locomotive, and which should be mou
on the shafts of the supporting wheels; and (3) a traction d
nometer of such form as would serve to indicate the horizo
moving force and at the same time allow but a slight horizo
motion of the engine on the supporting wheels. It was beli
that a locomotive thus mounted could be run either ahea
back under any desired load and at any speed; that while
run, its performance could be determined with a degree of accu
and completeness far excelling that which it is possible to se
under ordinary conditions of the road; and that the whole
paratus would be extremely valuable to students in steam
gineering. It was not thought that every condition of the t
would be perfectly met, but it was expected that the results
tained would prove valuable in extending a knowledge of l
motive performance.

The locomotive was ordered of the Schenectady Locomo
Works, in May, 1891. It is of the eight-wheel type, with 17 x
inch cylinders and 63-inch drivers. Its weight is 85,000 pou
56,000 pounds being on the drivers.

The details of the mounting were designed during the sum

* Presented at the San Francisco Meeting, May, 1892, of the American Sc
of Mechanical Engineers, and forming part of Volume XIII. of the Transact

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A small dash-pot (not shown) is attached to the last lever, *D*. The lever *C* has depending from it a light rod, which controls the balanced valve *B* (Figs. 91 and 92) in the pipe supplying the brakes with water. By means of this valve, as previously described, the load on the brakes is made to vary automatically with the position of the last lever of the dynamometer.

The locomotive while in motion may be given a final adjustment to its place on the supporting wheels by means of the nuts on the draw-bar.

WORK ON GRADIENTS.—The matter of subjecting the locomotive to conditions similar to those of an inclined track has not been considered as in any way essential to the present scheme, and no complications have been added to the mounting mechanism with this end in view. It would appear, however, if the point of contact between each driver and its supporting wheel be such as to make their common tangents represent the desired grade, that then the conditions of gradient would be met. The nuts on the draw-bar readily allow such an adjustment.

CCCCXCI.*

THE UTILIZATION OF THE POWER OF OCEAN WAVES.

BY ALBERT W. STAHL, U. S. N., SAN FRANCISCO, CAL.
(Member of the Society.)

AN intelligent study of the possibility and practicability of utilizing the power of ocean waves presupposes a thorough knowledge of the geometry and mechanics of wave motion. It is proposed, in this paper, to set forth the modern and generally accepted theory of such motion, so far as it applies to the subject in hand, and to deduce therefrom, if possible, the logical and most efficient method of utilizing the power of the waves; giving at the same time a brief description and criticism of methods heretofore proposed and employed.

While the motion of ocean waves in nature is usually quite complex in character, there are certain simple typical forms of such motion which have been satisfactorily studied, and the geometry and mechanics of which are well understood; and by making suitable combinations of these simple type-waves, the condition of more or less complex and irregular seas can be approximated to and their mechanics investigated with sufficient exactness for most practical purposes.

Prominent among the simple types of waves just referred to, and forming usually by far the most important element of actual ocean waves in nature, are those known as the deep-sea wave and the shallow-water wave. These are the forms of wave motion which occur in nature in a long series of waves, in which each successive wave is an exact reproduction of the one just preceding it, so that the wave goes on repeating itself indefinitely. While these conditions are rarely complied with exactly in nature, yet they are often very nearly so; so much so, that from a study of these two types and of their combinations, we can draw conclusions which are practically applicable to nearly all the motions of the sea.

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* Presented at the San Francisco meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

The shallow-water wave is really the *general* case of regular trochoidal wave motion, the depth of the water entering as a factor in determining the shape and motion of the wave, while the deep-sea wave is only that *special* case of such motion in which the depth of water is so great that it no longer has any appreciable influence on the wave motion.

While it would thus be more logical to discuss first the more general case of the shallow-water wave, and to pass thence, by limitation, to the special case of the deep-sea wave, yet practically the much greater simplicity of the theory of the latter makes it preferable to reverse this order, taking up first the discussion of the deep-sea wave, and thence passing to the other and more general case. Our attention for the present will therefore be confined to the deep-sea wave, and the following discussion will be understood as referring to that wave except when otherwise specified.

Many and widely different theories of wave motion have been advanced from time to time, of which it is here only necessary to state that the older theories are now definitely set aside as erroneous; and that the modern or trochoidal theory is generally accepted as very closely representing the actual phenomena which occur in nature.

Before entering on the explanation and discussion of the trochoidal theory of wave motion, it will be well to note the conditions which must in all cases be satisfied by any correct theory:

(1) The Condition of Dynamical Equilibrium, which expresses the general law of the motion of liquids, that the effective force acting on each particle to produce acceleration is the resultant of its weight and the pressure of the surrounding liquid.

(2) The Condition of Continuity, which expresses the fact that the mass of each elementary volume fixed in space within the liquid is constant.

(3) The Boundary Conditions, which are in general the depth and conformation of the bottom, the extent of the surface of the liquid and the state of pressure on the same. For our purposes the depth is assumed as infinite or so great as to have no appreciable influence, and the same assumption is made as to the extent of surface. The pressure on the surface is assumed to be uniform.*

* Since the surface of the liquid is in contact with the atmosphere, it is not strictly a free surface, but rather the surface of separation of two fluids, the one

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the engine moves ahead or aback, the stress is transmitted by the draw-bar and its value is shown by the weight necessary to balance the lever *H*.

The dynamometer levers are carried by a heavy framework, which is well secured to the locomotive foundation and to surrounding parts of the building. The character of the framing is but imperfectly shown by the drawings.

While the draw-bar is the only active agent by which the horizontal movement of the locomotive is controlled, there is ample provision of chains and buffers to check any excessive movement which may chance to occur.

Above the levers of the dynamometer, a floor is laid which chiefly serves the purposes of a tender. It gives room for a tank from which the locomotive injectors draw their supply and for the storage of a limited quantity of coal. Connected with the water tank is a glass gauge *I*, and above the tank is a weighing barrel through which the tank receives its supply. Scales for weighing fuel are also given a place on the "tender floor."

The three counters at *J* are connected, respectively, with the rear driving axle and to each of the two supporting shafts. These give a ready means for determining the speed of the engine, and the per cent. of slip between the drivers and their supporting wheels.

The telltale at *K* shows the position of the locomotive relative to the supporting wheels. The board *a* is fastened to the locomotive and consequently moves with it; the rod *b* is connected at one end to an iron column as a fixed point, and at the other end to the pointer *d*. This pointer is pivoted to the board *a* at *e*, so that any backward or forward movement of the locomotive is greatly multiplied by a similar movement of the lower end of the pointer *d*.

A tangent-wheel and screw are provided at *L* for the purpose of turning by hand the forward supporting shaft, and hence the engine, whenever it may be desired to do so, as, for example, for convenience in valve-setting. When not in use, the screw may be disengaged.

The truck-wheels of the engine rest upon light rails which are fixed at the level of the laboratory floor and extend in front of the engine a distance sufficient to allow the whole machine to be moved forward off the supporting wheels, whenever the latter may need to be taken out for repairs.

A Sturtevant $4\frac{1}{2} \times 6\frac{3}{4}$ steam blower located above the engine (Fig. 91) but not in pipe connection with it, removes from the room everything that is given out of the locomotive stack, without changing, materially, the draft conditions under which the locomotive is worked.

The cylinder cocks and the over-flow pipes from the injectors, are all in loose connection with the sewer. The discharge from the over-flow pipes may be directed into weighing barrels.

The boiler is in pipe connection with the fixed boiler which supplies steam for general use in the laboratory, so that the locomotive may be used to supply steam to other apparatus, or the fixed boiler may be used to supply the locomotive. In the latter case the locomotive boiler is drained by the steam trap *M*; by this means the boiler is freed from water, and as

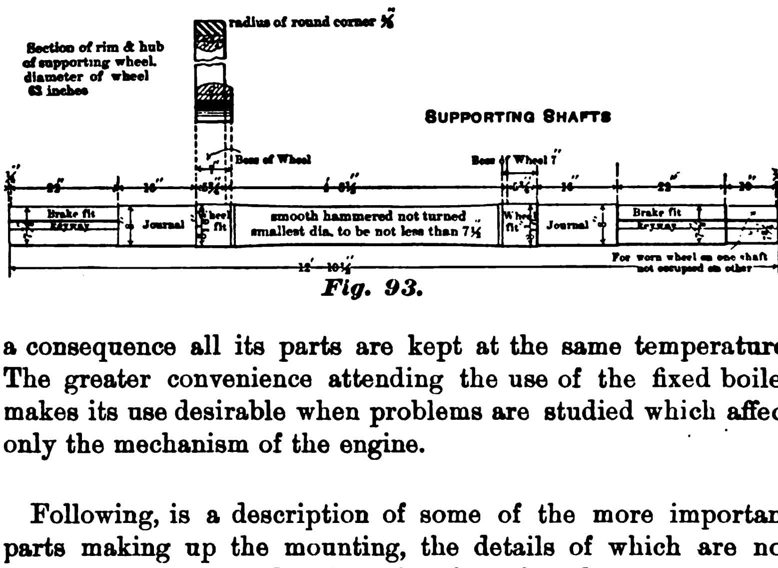

Fig. 93.

a consequence all its parts are kept at the same temperature. The greater convenience attending the use of the fixed boiler makes its use desirable when problems are studied which affect only the mechanism of the engine.

Following, is a description of some of the more important parts making up the mounting, the details of which are not clearly shown by the drawings thus far referred to.

THE SUPPORTING SHAFTS are of hammered iron and of the form and dimensions shown by Fig. 93. The bearings for these shafts are 8″ in diameter and 16″ long. Each bearing is fitted to a cast-iron plate 14″ x 36″ x 2″, which plate, in turn, is carefully bedded upon the oak timbers (Figs. 91 and 92). The whole is made secure by four bolts passing through the bearing, plate and timber.

THE ALDEN FRICTION BRAKES which supply the load to the

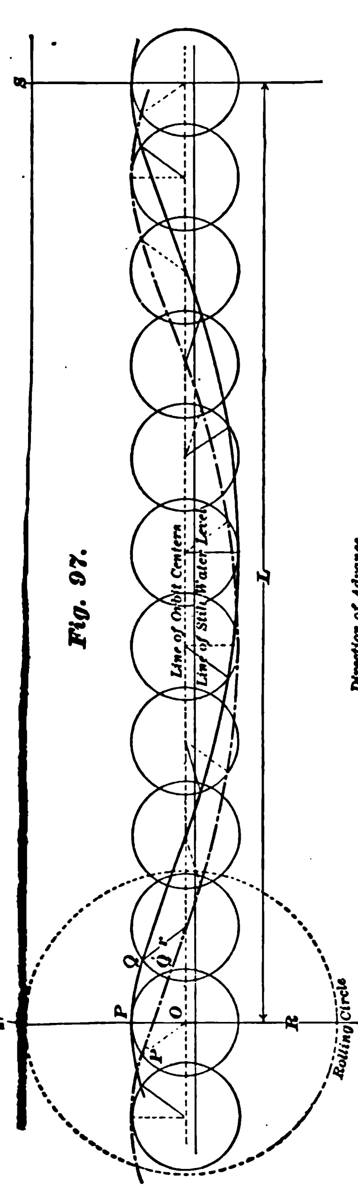

Fig. 97.

Direction of Advance

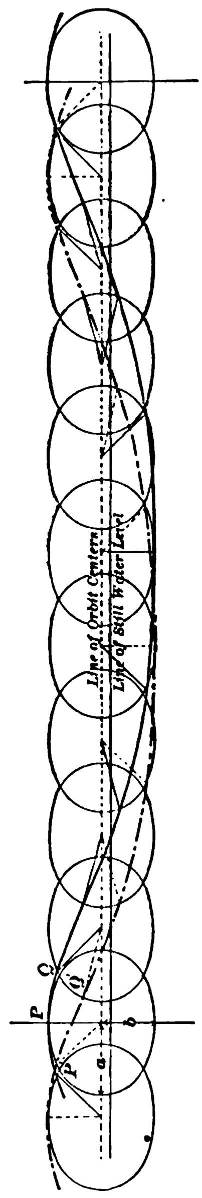

Fig. 98.

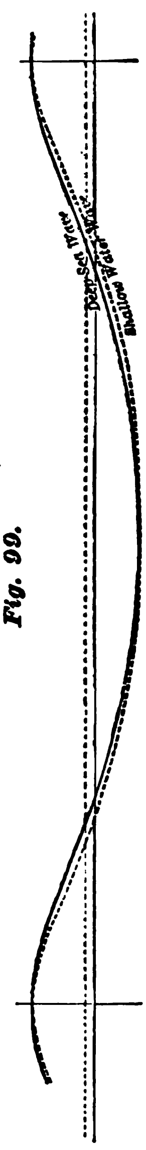

Fig. 99.

into any number of equal parts, say 10, and with each point of subdivision as a centre describe a circle of radius r. At ON, assumed as the crest plane, the particle is at the highest point of its orbit and moving to the left *with* the wave, as at P. As we have divided the wave length into ten parts, the particle in the first circle to the right of ON must be $\frac{360°}{10} = 36°$ further advanced in its orbit than the one at P, and is thus shown at Q. Similarly, the particle in each successive circle to the right must be advanced 36° further than in the circle just preceding. Thus we find that in the fifth circle the particle is at 180° from crest position and moving to the right, being opposite to the direction of propagation of the wave, while in the tenth circle the particle has again the same phase and motion as at P.

Drawing a curved line through the particles in successive circles, we obtain the profile of the wave, which is the curve known as the trochoid; hence the name trochoidal theory. If with O as a centre we describe a circle whose circumference is equal to the length of the wave, *i.e.*, its radius $ON = R = \frac{L}{2\pi}$, and roll this circle along the horizontal line NS, then the point P of the radius ON will describe the same trochoid previously constructed.

The trochoidal curve is markedly more peaked at the crest than at the trough, and the more so the higher the wave in proportion to its length. As the point P is taken further out along the radius ON, the height of the trochoidal curve increases, the limiting case being evidently when the point P is taken at the extremity of the radius $ON = R$. The cycloidal curve produced in the latter case is the theoretical breaking wave, because if the point P be further from O than the distance ON, the curve would become looped, involving discontinuity in the water. As a matter of fact, waves break long before this condition is reached, and there are theoretical reasons for believing that, in nature, trochoidal waves never reach a sharper angle than 30° of slope at the breaking cusp.

One of the most noticeable features of such waves is their apparently rapid advance, even when their dimensions are moderate. Thus a wave 200 feet long has a velocity of 19 knots per hour; waves of 400 and 600 feet in length have speeds of 27 and 32 knots per hour respectively.

But it is most important to note that in all wave motions it is the wave *form* that travels at these high speeds, and not the particles of water. To illustrate the rapid advance of the wave form consequent on a small motion of the individual particles, let us in Fig. 97 cause each of the particles to revolve in its respective orbit circle through an additional angle of 36°. Drawing a curve through the new positions of the particles, we have the wave profile in its new position. The wave form has thus advanced from its original position a distance of $\frac{1}{10}$ of the wave length, while each particle has only moved over $\frac{1}{10}$ of its orbit circumference, the latter distance for ordinary waves being only about one-sixth to one-seventh of the former.

It should be noticed that the orbital motion of the particles may be regarded at every instant as the resultant of two motions, one vertical and the other horizontal, the value of each of these component motions depending on the phase of the particle considered. At certain points one or other of these motions reduces to zero. Thus at the crest or trough the particle instantaneously moves horizontally and has no vertical motion, while at mid-height it moves vertically and has no horizontal motion. Its maximum horizontal velocity will be at the crest or trough; its maximum vertical velocity at mid-height. Thus, uniform motion along the circular orbit involves continuously varying accelerations and retardations of the component velocities in the horizontal and vertical directions.

We will now ascertain whether the trochoidal theory, as above set forth, will satisfy the condition of dynamical equilibrium. In Fig. 100 let P represent a particle of unit mass, with orbit centre O and orbit radius $OP = r$. Let the linear speed of advance of the wave form be expressed by V, and let ω represent the angular velocity of the particle in its orbit, the units being feet and seconds. The particle P is acted on by, (1) gravity, acting vertically downwards; (2) the acceleration of the centrifugal force acting from O along the radius r; (3) the pressure of the adjacent fluid particles, which must be normal to the surface of the liquid at P. Now in order that the particle may remain in dynamical equilibrium—*i.e.*, in order that it may continue moving at a uniform speed in its orbit circle – the acceleration of the centrifugal force must evidently be counterbalanced by an equal acceleration in the opposite direction; and this latter acceleration must be the resultant of the weight of the particle and the pressure of the adjacent fluid. But this simply amounts to saying that the three forces first above mentioned as acting on the particle P must be in equilibrium. Let the line OP represent in amount, as it does in direction, the centrifugal force $\omega^2 r$, and from O lay off vertically ON of such a length that it may represent gravity g on the same scale as that on which OP represents $\omega^2 r$. Join NP. In accordance with the principle of the triangle of forces, NP must represent, both in direction and magnitude, the third and remaining force, *i.e.*, the fluid pressure; and hence NP must be normal to the surface of the liquid at P. Now this can only be true if $ON = R$ the radius of the rolling circle of a trochoid passing through P; for in that case N will be the instantaneous centre of motion of this rolling circle, and hence of all points, such as P, within that circle, and consequently the instantaneous

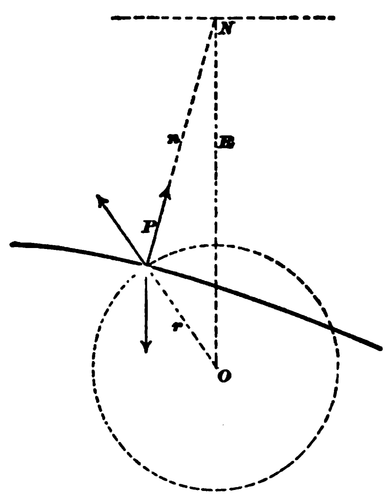

Fig. 100.

motion of P (and hence the direction of the surface of the liquid) will be normal to NP. In the triangle above constructed, we have

$$\frac{ON}{OP} = \frac{g}{\omega^2 r}$$

$$\therefore ON = \frac{g}{\omega^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

But we have just seen that, in order that NP shall be normal to the surface at P, we must have $ON = R$. We thus have

$ON = R = \frac{g}{\omega^2}$ or $\omega = \sqrt{\frac{g}{R}}$, as a necessary condition for dynamical equilibrium.

The velocity, V, of the wave form is evidently equal to the velocity, $R\omega$, of the circumference of the rolling circle; and as ω must equal $\sqrt{\frac{g}{R}}$, we have

$$V = R\omega = \sqrt{gR};$$

and letting L, as before, represent the length of the wave, we have

$$V = \sqrt{\frac{gL}{2\pi}} = \sqrt{5.123L}. \quad . \quad . \quad . \quad . \quad (2)$$

It follows that the condition of dynamical equilibrium can only be satisfied provided the speed of the wave depends on its length alone, and that their relation is as shown by this equation. We shall see later that this relation between speed and length is practically identical with that found to exist in waves in nature; and that hence this condition is satisfied by the trochoidal theory.

We have seen that the line NP represents the resultant pressure on the particle on the same scale as ON represents the force of gravity, and is thus for wave motion the analogue of gravity in still water; for evidently the rate of increase of the fluid pressure in a direction normal to the surface is measured at any point by the corresponding length of this line NP, whereas in still water it would be measured by the length of ON, or gravity. It is hence called the *virtual gravity* at the point P. Thus for a sub-surface of uniform pressure infinitesimally near the surface, the normal thickness of the layer of liquid at any point must be inversely proportional to the corresponding virtual gravity; or if z be this normal thickness and n the value of virtual gravity, we must have all along the layer

$$n.z. = \text{Constant}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

the second condition necessary for dynamical equilibrium. We shall see later that this condition is likewise fulfilled by our theory.

Taking up next the consideration of the condition of continuity, we may state the same very simply by saying that, although any given mass of water changes its shape during the passage of the wave form, yet the continuity of the water must remain unbroken.

In Fig. 101, let P represent any particle with orbit centre O, and orbit radius r. Let $ON = R$ be the radius of the rolling circle. If we now imagine this whole system to be given a velocity $V = R\omega$, equal to the actual velocity of the wave, but in the opposite direction, it is evident that the wave form will no longer appear to move along bodily, but that the particles will move along an apparently fixed trochoidal stream. The

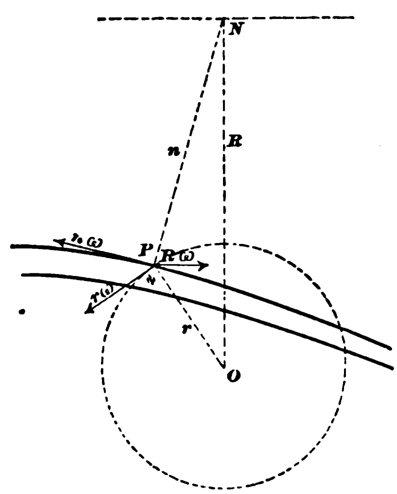

Fig. 101.

velocity of the particle P in such stream is the resultant of its orbital velocity $r\omega$ and of the velocity $R\omega$, supposed to be imparted to it and all other parts of the system in a direction opposite to that of the propagation of the wave. These two velocities are normal respectively to the radii $OP = r$ and $ON = R$: and as the final velocity of the particle in the stream must be normal to $NP = n$, it follows that the triangle NOP is the triangle of velocities for this point and that this stream velocity must be $n\omega$. Considering a small elementary stream of thickness z, bounded by two vertical planes parallel to the direction of wave propagation, we have a kind of trochoidal pipe or tube. For continuity, this pipe must always run full, and the volume passing any section in the unit of time must be constant. We must therefore have

$$n.\omega.z = \text{Constant},$$

which for any assumed constant value of ω, reduces to

$$n.z = \text{Constant}, \quad \ldots\ldots\ldots \quad (4)$$

this being then the expression of the condition of continuity, which is thus seen to be identical with that of the second condition for dynamical equilibrium.

The investigation of the equilibrium of the trocho dal wave form applies unchanged to any sub-surface of uniform pressure, so far as its equilibrium *per se* is concerned. We will now examine what trochoidal sub-surface, if any, ultimately very near the surface, of the same length as the wave, will fulfil the purely geometrical condition $n.z =$ Constant, imposed by the conditions of both continuity and dynamical equilibrium.

In Fig. 102, let the orbit centres of the upper trochoidal surface lie on a horizontal line through the point O. Conceive a trochoidal sub-surface, whose line of orbit centres passes through O', indefinitely near O. Then, since the two trochoidal surfaces are to lie crest under crest, L and hence R must be the same for both surfaces. NN' will be equal to OO'; and $N'P' = n' = n + \delta n$ will be the consecutive value and direction of virtual gravity as we go downward to the lower trochoid. The orbit radius of the upper trochoid being r, that of the lower trochoid will similarly be $r' = r + \delta r$.

Produce NP, which is normal to the surface at P, to meet the sub-surface in P'', so that ultimately, when the two surfaces are infinitesimally close together, $PP'' = z$. Project NN' and $N'P'$ on NP'', the point P' projecting on P'' with an error of the second order which may be neglected. Then we have

$$NP + PP'' = NN'.\cos.\,\phi + N'P'.\cos.\,\delta\phi \quad . \; . \quad (5)$$

that is, $n + z = NN'.\cos.\,\phi + (n + \delta n)\cos.\,\delta\phi. \quad . \; . \; . \; . \quad (6)$

As the trochoidal surfaces approach more and more closely, $\delta\phi$ approaches the value zero and cos. $\delta\phi$ approaches the value 1. In the limit, when these values are reached, we have therefore

$$n + z = NN'.\cos.\,\phi + n + \delta n \quad . \; . \; . \; . \; . \quad (7)$$

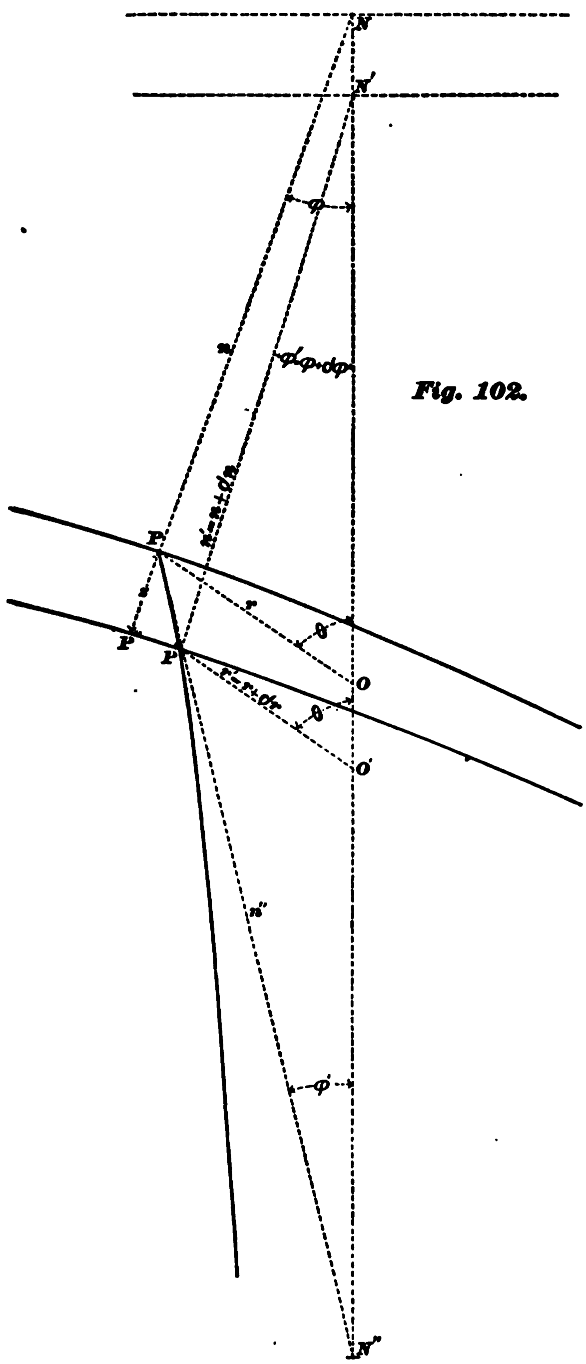

Fig. 102.

Letting h = depth of line of orbit centres of any sub-surface trochoid below line of surface orbit centres, we have

$NN' = OO' = \delta h$, and consequently $n + z = \cos.\ \phi.\ \delta h + n + \delta n.$

Transposing and multiplying through by n we get

$$n.z = n.\ \delta n + n.\ \cos.\ \delta\phi.\ \delta h \quad . \quad . \quad . \quad . \quad . \quad (8)$$

But from the triangle NPO, we have

$$n\ \cos.\ \phi = R - r\ \cos.\ \theta; \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

and this value of n cos. ϕ being substituted in Eq. 8, we have

$$n.z = n.\ \delta n + (R - r\ \cos.\ \theta).\ \delta h. \quad . \quad . \quad . \quad . \quad . \quad (10)$$

From the triangle NPO, we also have

$$n^2 = R^2 + r^2 - 2\ R\ r\ \cos.\ \theta. \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Differentiating this expression in terms of n and r, the angle θ being constant, or, in other words, differentiating for an adjacent lower particle in the same phase, we have

$$n.\ \delta n = r.\ \delta r - R.\ \cos.\ \theta.\ \delta r \quad . \quad . \quad . \quad . \quad . \quad (12)$$

and substituting this value of $n.\ \delta n$ in Eq. 10, we get

$$n.z = r.\ \delta r + R.\ \delta h - (R.\ \delta r + r.\ \delta h)\ \cos.\ \theta. \quad . \quad . \quad (13)$$

But since $n.z$ must be *constant* all along the elementary stream layer, its value must be independent of θ, and hence the term containing θ must vanish. As cos. θ is not, in general, equal to zero, we must therefore have

$$R.\ \delta r + r.\ \delta h = O, \text{ whence } \delta h = -R.\frac{\delta r}{r}, \quad . \quad . \quad (14)$$

or ultimately,

$$dh = -\ R.\frac{dr}{r} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Letting r_o = surface orbit radius, and integrating between the limits of the surface and the depth h, we have

$$\int_o^h dh = \int_{r_o}^{r} - R.\frac{dr}{r} \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

$$h = R.\ \log_e \frac{r_o}{r} \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

$$\therefore r = r_o \, . \, \varepsilon^{\frac{-h}{R}} = r_o \, . \, \varepsilon^{\frac{-2\pi h}{L}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

This is the law which the orbit radii at successive depths must fulfil, in order that the conditions of continuity and dynamical equilibrium may be satisfied ; or in other words, as the depth increases in arithmetical progression, the orbit radii must diminish in geometrical progression.

Assuming a surface orbit radius $r_o = 1$, this expression becomes

$$r = \varepsilon^{\frac{-2\pi h}{L}} = 2.7183^{\frac{-2\pi h}{L}}$$

$$\text{and } \log. r = -\frac{2\pi h}{L} . \log. 2.7183 = -2.7287 \frac{h}{L} . \quad . \quad . \quad (19)$$

By giving successive values to $\frac{h}{L}$ in this expression, we obtain the results given in the following table :

$\frac{h}{L}$	r	$\frac{h}{L}$	r
0.00	1.0000	.50	.0432
.05	.7304	.60	.0231
.10	.5835	.70	.0123
.15	.3897	.80	.0066
.20	.2846	.90	.0035
.25	.2079	1.00	.0019
.30	.1518	1.10	.0010
.35	.1109	1.20	.00053
.40	.0810	1.50	.00008
.45	.0592	2.00	.0000035

This table illustrates the great difference between the amplitude of motion of the surface particles as compared with those at greater depth. Assuming thus, for instance, a storm wave 600 feet long and 30 feet high, we see that at a depth of 150 feet the height of the surface wave will be 30 × .2079 = 6.237 feet = 6 feet 2.84 inches ; at 300 feet depth its height will be 30 × .0432 = 1.296 feet = 1 foot 2½ inches ; while at 600 feet depth its height will only be 30 × .0019 = .057 feet = 0.68 inch.

We have thus succeeded in finding a mathematical scheme of motion fulfilling the first three conditions, and it only remains to develop the qualities of the motion, and finally to apply the somewhat complex conditions of formation as far as we understand them, in order to discover the initial circumstances necessary for the generation of such motion, and see how far they obtain in nature.

The body of water under the trochoidal surface divides itself into an indefinite number of trochoidal sub-surfaces of uniform pressure; and since the energy of each particle in a sub-surface is the same, they must originally have formed a horizontal sub-

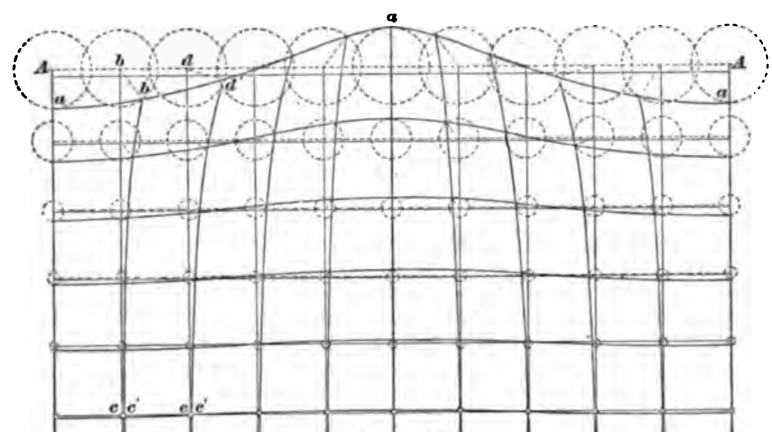

Fig. 103.

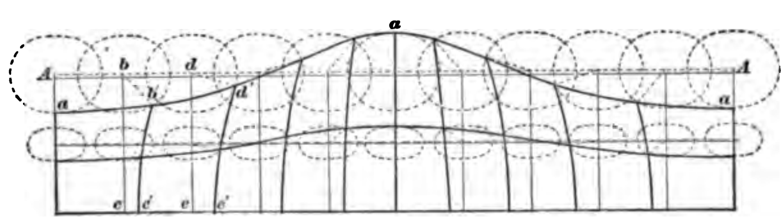

Fig. 104.

surface in still water at the corresponding level, or the originally horizontal layers have by the passage of the wave system become trochoidal surfaces. Also the originally vertical layers have become distorted according to the laws of the orbit radii.

The internal structure of the trochoidal deep-sea wave is graphically shown in Fig. 103. The originally horizontal and vertical layers are drawn to intersect so as to form squares, the displacements and distortions of which, with constant area, are shown by the quasi-parallelograms included between the trochoidal surfaces and sub-surfaces and the distorted verticals.

The dotted horizontal lines are the lines of orbit centres corresponding to the still-water levels shown by the full lines close below them. The rapid diminution of the motion with increasing depth is clearly shown by the rapid decrease in size of the orbit circles.

A notable feature of the trochoidal deep-sea wave is that it is more peaked at crest than at trough. Thus while $AB'C$ (Fig. 106) represents the curve of sines, the crest and trough of which are perfectly symmetrical, ABC represents the trochoid of same length and height. Considering the area of half the advancing heap of water $ABCD$, it follows that the corresponding surface of still water must fall below the mid-height or line of orbit centres OO, the amount of the difference of level

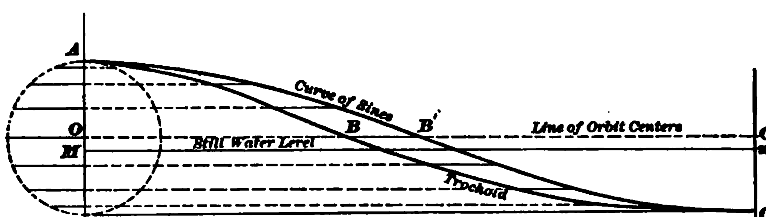

Fig. 106.

being most simply obtained by comparing the corresponding horizontal elements of the areas of the curve of sines and of the trochoid. From the methods of constructing these two curves, it follows that the difference at each height is equal to the corresponding horizontal element of the orbit semicircle, so that the difference in the areas from crest to trough is equal to the area of this semicircle $= \frac{\pi r^2}{2}$. But the area from crest to trough of the curve of sines, from its symmetry, is the half product of its length by its height, or $\frac{rL}{2}$, whence the corresponding area of the trochoidal half wave is $\frac{rL}{2} - \frac{\pi r}{2}$. The corresponding level of still water is therefore the line Mm, such that the area of the rectangle $MmCD$ is

$$MD \times \frac{L}{2} = \frac{rL}{2} - \frac{\pi r^2}{2},$$

whence

$$MD = r - \frac{\pi r^2}{L}; \text{ and } OM = r - MD = \frac{\pi r^2}{L} = \frac{r^2}{2R}. \quad (20)$$

Thus the mean elevation of the particles of any trochoidal surface, throughout the wave length, above their corresponding still-water level, is $\frac{r^2}{2R}$, which for the particles of the upper surface becomes $\frac{r_0^2}{2R}$.

It follows that the mean *potential* energy of each particle in a complete revolution is $mg\frac{r^2}{2R}$. Its *kinetic* energy in its orbit is $\frac{m\omega^2 r^2}{2} = mg\frac{r^2}{2R}$, the same as its *potential* energy. Accordingly, the whole energy of the wave is half kinetic and half potential.

Considering any elementary layer $L.\delta k$ of undisturbed water, it appears from the above that its centre of gravity is by the passage of the wave form raised through a height $\frac{r^2}{2R}$.

Letting $W =$ the weight of the liquid per unit of volume, we have for the potential energy of an elementary layer of unit breadth

$$L.\,\delta k.\,W.\,\frac{r^2}{2R} = L.\,\delta k.\,W.\,\frac{r_0^2}{2R}.\;\varepsilon^{\frac{-2h}{R}}. \quad . \quad . \quad . \quad (21)$$

But, substituting in equation (13) the value of δr obtained from equation (14), we get

$$n.z = \frac{R^2 - r^2}{R}.\,\delta h = R.\delta k \quad . \quad . \quad . \quad . \quad . \quad (22)$$

where δk is the increment of head in still water corresponding to δh in the wave motion; whence

$$\delta k = \left(1 - \frac{r^2}{R^2}\right)\delta h = \left(1 - \frac{r_0^2}{R^2}.\,\varepsilon^{\frac{-2h}{R}}\right)\delta h \quad . \quad . \quad . \quad (23)$$

Substituting, we have for the potential energy of the elementary layer

$$W.L.\frac{r_0^2}{2R}\left(1 - \frac{r_0^2}{R^2}.\,\varepsilon^{\frac{-2h}{R}}\right)\varepsilon^{\frac{-2h}{R}}\,\delta h. \quad . \quad . \quad . \quad (24)$$

Integrating this expression between the limits $h = \infty$ and

$h = 0$, we get for the potential energy of the whole wave length = kinetic energy of the whole wave length

$$= W.L.\frac{r_0^2}{2R}\left(\frac{R}{2} - \frac{r_0^2}{4R}\right)$$

$$= W.L.\frac{r_0^2}{4}\left(1 - \frac{r_0^2}{2R^2}\right)$$

$$= W.L.\frac{H^2}{16}\left(1 - \frac{\pi^2 H^2}{2L^2}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

The total energy for the whole wave length = the sum of the kinetic and potential energy for the whole wave length, and is thus

$$W.L.\frac{H^2}{8}\left(1 - \frac{\pi^2 H^2}{2L^2}\right) = W.L.\frac{H^2}{8}\left(1 - 4.935\frac{H^2}{L^2}\right). \quad . \quad (26)$$

For sea water $W = 64$ lbs. per cubic foot.

Hence the total energy of one whole wave length of a wave H feet high, L feet long, and one foot in breadth is

$$E = 8LH^2\left(1 - 4.935\frac{H^2}{L^2}\right) \text{ foot pounds.} \quad . \quad . \quad (27)$$

The time required for each wave to travel through a distance equal to its own length being $P = \sqrt{\frac{L}{5.123}}$ seconds, it follows that the number of waves passing any given point in one minute is

$$N = \frac{60}{P} = 60\sqrt{\frac{5.123}{L}}.$$

Hence the total energy of an indefinite series of such waves expressed in horse-power per foot of breadth, is

$$\frac{E \times N}{33000} = .0329H^2L\left(1 - 4.935\frac{H^2}{L^2}\right). \quad . \quad . \quad . \quad (28)$$

By substituting various values for $\frac{H}{L}$, within the limits of such values actually occurring in nature, we obtain the following table of

TOTAL ENERGY OF DEEP-SEA WAVES IN TERMS OF HORSE-POWER PER FOOT OF BREADTH.

Ratio of length of waves to height of waves.	Length of Waves in Feet.							
	25	50	75	100	150	200	300	400
50	.04	.23	.64	1.31	3.62	7.43	20.46	42.01
45	.05	.29	.79	1.62	4.47	9.18	25.30	51.94
40	.06	.36	1.00	2.05	5.65	11.59	31.95	65.58
35	.08	.47	1.30	2.68	7.37	15.14	41.72	85.63
30	.12	.64	1.77	3.64	10.02	20.57	56.70	116.38
25	.16	.90	2.49	5.23	14.40	29.56	80.85	167.22
20	.25	1.44	3.96	8.13	21.79	45.98	126.70	260.08
15	.42	2.83	6.97	14.31	39.43	80.94	223.06	457.89
10	93	5.58	15.24	31.29	86.22	177.00	487.75	1001.25
5	3.30	18.68	51.48	105.68	291.20	597.78	1647.31	3381.60

The above table gives the *total* energy, including both kinetic and potential, inherent in a regular series of waves. The figures are strictly correct for the trochoidal deep-sea wave with circular orbits only, though they give a close approximation for

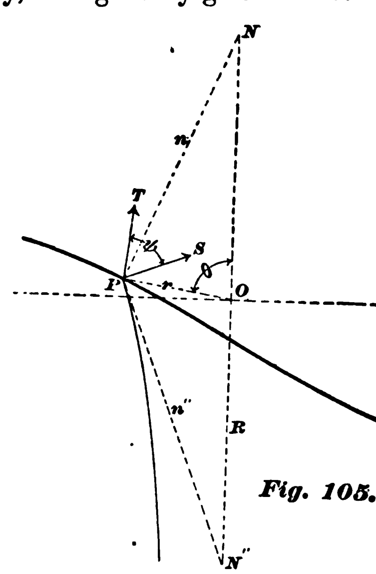

Fig. 105.

any nearly regular series of waves in deep water, and a fair approximation for waves in shallow water.

Let us now examine the transmission of energy through a distorted vertical column of unit breadth. In Fig. 105, let p be the intensity of the normal pressure at any point of the distorted vertical, being directly proportional to the original still-water depth of the point, so that calling W the weight of the liquid per unit volume, we have

$$dp = W.dk = W.\frac{r^2 - R}{R.r}\,dr. \quad . \quad . \quad . \quad . \quad . \qquad (29)$$

At the surface $r = r_0$ and (neglecting atmospheric pressure) $p = O$. Integrating between the limits of the surface and any desired depth, we get

$$\int_0^p dp = W\int_{r_0}^{r} \frac{r^2 - R^2}{R.r}\,dr$$

$$p = W.\left[R.\log_e \frac{r}{r_0} - \frac{r^2 - r_0^2}{2R}\right] \quad . \quad . \quad . \quad . \qquad (30)$$

Draw PT perpendicular to OP, and PS perpendicular to PN'' and let ψ represent the angle TPS.

The length of a short element of the distorted vertical is

$$ds = \frac{n''}{r}\cdot dr = \frac{r + R\cos.\theta}{r\cos.\psi}\cdot dr,$$

and the velocity of the point P in the direction PS of the normal pressure is $\omega r\cos.\psi$.

Then the energy exerted in time dt by the normal pressure p on a length ds moving with a velocity $\omega r\cos.\psi$, is

$$p.ds.\omega r\cos.\psi.dt$$

$$= W\left(R.\log_e \frac{r}{r_0} - \frac{r^2 - r_0^2}{2R}\right)\left(\frac{r + R\cos.\theta}{r\cos.\psi}\right)\omega r\cos.\psi.dr.dt$$

$$= W.R\omega\left\{\left(\log_e \frac{r}{r_0} - \frac{r^2 - r_0^2}{2R}\right)\left(\frac{r}{R} + \cos.\theta\right)\right\}dr.dt \quad (31)$$

Hence the energy exerted by the whole length of the distorted vertical in time dt

$$= \int_{r_0}^{0} W.R\omega\left\{\left(\log_e \frac{r}{r_0} - \frac{r^2 - r_0^2}{2R}\right)\left(\frac{r}{R} + \cos.\theta\right)\right\}dr.dt$$

$$= W.R\omega\left\{R\cos.\theta\left(r_0 - \frac{r_0^3}{3R^2}\right) + \frac{r_0^2}{4}\left(1 - \frac{r_0^2}{2R^2}\right)\right\}dt \quad . \quad (32)$$

From the form of this expression it is evident that during a portion of the wave period the column exerts energy forward or in the direction of propagation, and during another portion of the period it exerts energy backward. The energy exerted forward is in excess of that exerted backward, and the total resultant forward energy exerted by the column during the wave period

$$=\int_o^T W.R\omega \left\{ R\cos.\theta\left(r_o - \frac{r_o^3}{3R^2}\right) + \frac{r_o^2}{4}\left(1 - \frac{r_o^2}{2R^2}\right)\right\} dt$$

$$=\int_o^{2\pi} W.R^2\cos.\theta\left(r_o - \frac{r_o^3}{3R^2}\right)d\theta + \int_o^{2\pi}\frac{R.\,r_o^2}{4}\left(1 - \frac{r_o^2}{2R_2}\right)d\theta \quad (33)$$

the first member of which vanishes between the limits, and the second evaluates to

$$W.\frac{\pi R r_o^2}{2}\left(1 - \frac{r_o^2}{2R^2}\right) = W L\frac{r_o^2}{4}\left(1 - \frac{r_o^2}{2R^2}\right) \quad . \; . \; (34)$$

which by comparison with equation (25) is seen to be one-half the total energy of the wave. During each complete wave period, then, an amount of energy equal to one-half the total energy of the wave passes through every column in the direction of propagation of the wave; in other words, one-half the total energy of the wave is transmitted forward with the wave form.

The above includes the main and important points in connection with the structure and movement of the deep-sea wave, according to the trochoidal theory. Many interesting deductions could be made therefrom, but it seems preferable in this paper to confine ourselves to those fundamental principles which have a direct bearing on the *energy* of the waves and its utilization.

We have found above that this theory satisfies the conditions of dynamical equilibrium, of continuity, and of boundary; it now remains to glance at the conditions of formation, and see whether our theoretical waves can be generated by the action of observed forces on water at rest. In this connection the initial condition of the still water as to "molecular rotation" must also be considered. This molecular rotation is a somewhat complex quality, involving a sliding over one another of successive layers, a grinding action, so to speak. A fairly good idea of the physical nature of "molecular rotation" may be obtained by considering a small sphere anywhere in the liquid to become suddenly solidified and removed from the remaining fluid, when if the sphere retains only a motion of translation, such motion is said to be *irrotational.* If, however, a rotation about its centre of gravity appears, the motion is *rotational.* Such motion evidently requires for its generation a force equivalent to internal friction, and hence cannot be generated or destroyed in a *perfect* fluid by natural forces. Thus if fluid motion is once irrotational, it is always so. We have then as a consequence that no wave motion involving molecular rotation can be generated by natural forces from a perfect fluid at rest (and therefore irrotational), and this is one of the tests which the conditions of formation imply.

Without entering into the mathematical demonstration, it may be stated that the theoretical trochoidal wave does involve such molecular rotation, which increases from the bottom up, at a rate increasing very rapidly near the surface. Our theory is then at fault in this respect, as no such wave can be formed from still water by natural forces. Also if such a wave be in existence, and then by the reverse action of natural forces be reduced to still water, the molecular rotation must remain constant, which can readily be demonstrated to require that the water shall have an initial velocity in a direction opposite to that of the wave propagation, decreasing with the depth and being at all depths equal to $V.\frac{r^2}{R^2}$. In the generation of waves from water at rest, the action of the wind undoubtedly exerts a forward force on the water at the surface, imparting a forward momentum to it, which by any lack of fluidity is imparted gradually, at a rate diminishing with the depth, to the water beneath. A current is thus generated somewhat of the *nature* required by the trochoidal motion, but in the opposite *direction,* the natural wave thus departing still more widely from the theoretical form. From these considerations, it does not seem possible that a *truly* trochoidal wave is ever formed except in accidental cases; and yet the modifications rendered necessary by the above, though important in *character,* are not important in *extent.* The motion actually generated from water at rest very nearly approaches trochoidal motion so long as the height of the wave is not great; and the smaller the height, the more nearly irrotational is trochoidal motion. As long as the height is not more than $\frac{1}{10}$ the wave length (and this is practically the case in the great majority of waves), the trochoidal wave may be regarded as practically irrotational and identical with the wave which can in nature be produced from water at rest. Beyond that height, the difference increases rapidly. Prof. Stokes has formulated a refinement of the trochoidal theory, based on the condition that the motion must be susceptible of being produced from water at rest, *i.e.*, no molecular rotation must be involved. According to his theory, the particles describe circular orbits about centres moving forward with the proper velocity $V.\frac{r^2}{R^2}$, the motion of oscillation being thus combined with a slow motion of translation. There is a slight deviation from the truly trochoidal form, the actual wave falling a little within the trochoid of same height and length. The deviation for a wave of height $\frac{1}{10}$ the length is two per cent. and varies as the cube of the height. Although this motion is of some theoretical importance, and may be a *trifle* nearer the truth than the trochoidal theory, it is still so much more complicated of expression and involves so small a departure from the simpler theory in most cases, that it has not been adopted as a working hypothesis.

When a previously calm sea is disturbed by the action of the wind, a large variety of oscillatory motions is produced; simple translation, rotation, and these two combined in many different proportions. But not all these motions are sufficiently stable to render them capable of indefinite propagation; most of them, in fact, have only a very slight margin of stability. The trochoid appears to be the only one of these possible motions whose margin of stability is sufficiently large to enable it to successfully withstand any considerable disturbance. The practical result thus seems to be that as the various motions thus produced come in contact so as to influence each other, the least stable forms are gradually destroyed, a portion of their energy going to increase the motion of the more stable systems. In general, it may be said that there seems a marked tendency for waves once formed to work down into the trochoidal form and characteristics.

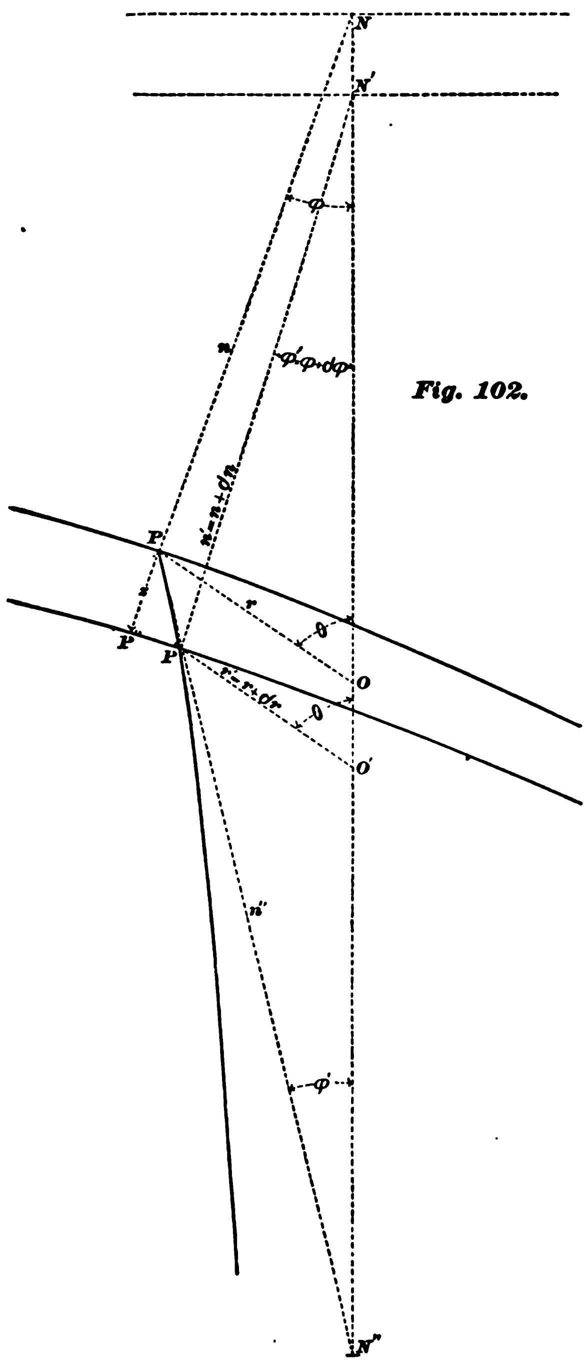

Fig. 102.

shallow-water wave motion in detail, it will be sufficient to note the points in which it differs from that of the deep-sea wave.

In Fig. 107, let $ABPCD$ be a common trochoid, and let us construct a new curve, $GFP'CK$, having the same abscissæ as the original trochoid, but having all its ordinates reduced in the proportion $\frac{SP'}{SP} = \frac{b}{a}$. This new curve is called a *reduced trochoid*, and is the path traced out by a point P', whose orbit is an *ellipse about the centre*, the semi-major and semi-minor axes of the ellipse being a and b, respectively, and the elliptical orbit itself having a uniform motion in the direction of its horizontal

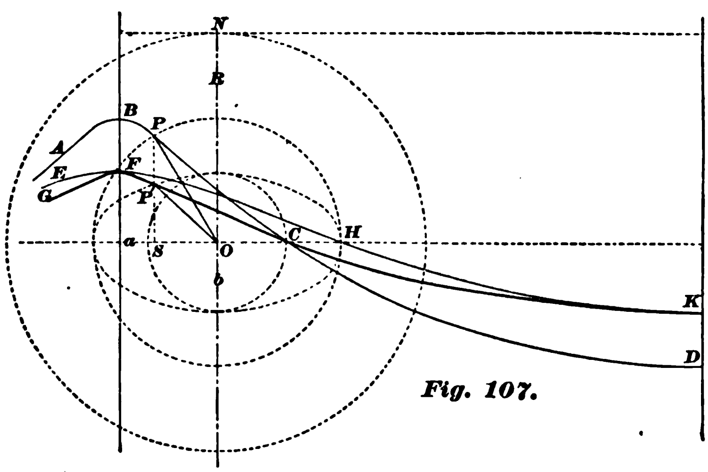

Fig. 107.

major axis. This curve is the theoretical oscillating wave in shallow water, the ratio $\frac{b}{a}$ depending on the depth of water. The character of the reduced trochoid is apparent from the figure. Constructing $EFHK$, the common trochoid having the same height as the reduced trochoid $GFP'CK$, we find that the latter is flatter at trough and steeper at crest than $EFHK$. Its highest and lowest points correspond with those of $EFHK$, but it crosses the line of orbit centres at the same point as $ABPCD$. The angular velocity of the particle in its elliptical orbit is not *constant*, as is the case in the circular orbits of the deep-sea wave, but is greater at and near crest and trough, and less at and near mid-height than would be the case if the orbit were circular. The velocity of the particle P' in its elliptical orbit is always such that the point P, vertically above it on the circumference of a circle having the major axis of the ellipse for its diameter, has a constant angular velocity.

In Fig. 98 is constructed the surface profile of a shallow-water wave for the case in which the total depth of water is 20% of the length of the wave, the actual length and height of this wave being the same as in the deep-sea wave, whose profile was constructed in Fig. 97. The difference in the two profiles is further illustrated in Fig. 99, where they are both drawn, the deep-sea wave being the full, and the shallow-water wave, the dotted curve.

This theoretical wave in shallow water does not satisfy the conditions of continuity and dynamical equilibrium so perfectly as does the theoretical deep-sea wave. Both these conditions are *perfectly* satisfied at the crest and at the trough of the shallow-water wave, but for intermediate points the theory is slightly at fault; so slightly, however, that it may be accepted as practically correct for our purposes.

Let a = horizontal semi-major axis of elliptical orbits at any depth h.

b = vertical semi-minor axis of elliptical orbits at any depth h.

a_0 = horizontal semi-major axis of elliptical orbits of surface particles.

b_0 = vertical semi-minor axis of elliptical orbits of surface particles.

h = depth from centre of orbits of surface particles to centre of orbits whose semi-axes are a and b.

h_0 = depth from centre of orbits of surface particles to bottom.

Then, for the surface orbits, it may be proved that

$$a_0 = b_0 \cdot \frac{\varepsilon^{\frac{4\pi h_0}{L}} + 1}{\varepsilon^{\frac{4\pi h_0}{L}} - 1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

The ratio $\frac{b_0}{a_0}$ of the axes of the surface orbits enters as a factor into all the expressions for velocity, period, energy, etc.

Thus the velocity of the shallow-water wave is

$$V = \sqrt{\frac{b_0}{a_0} \cdot \frac{gL}{2\pi}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (36)$$

which is seen to be less than for the deep-sea wave. As the depth of water increases, the ratio $\frac{b_0}{a_0}$ approaches unity, and when the theoretically infinite depth of the deep-sea wave is reached, $\frac{b_0}{a_0} = 1$, and all these formulæ reduce to those previously deduced for that wave.

To illustrate the practical relation existing between the shallow-water and deep-sea wave, the following table has been computed:

Depth in percentage of wave length. $\frac{h_0}{L}$	Ratio of axes of surface orbits. $\frac{b_0}{a_0}$	Velocity of shallow-water wave in percentage of velocity of deep-sea wave of same length.
.10	.557	.746
.15	.786	.858
.20	.847	.920
.25	.917	.958
.30	.955	.977
.35	.975	.987
.40	.987	.993
.45	.993	.996
.50	.996	.998
.55	.998	.999
.60	.999	.9999
.75	.9999	.99999
1.00	.99999	.999999

From this table it appears that there is practically no difference between the deep-sea and shallow-water wave, so long as the depth of water is not less than about one-half the length of the wave. For depths less than about one-third the wave length, the difference rapidly increases. The axes of the elliptical orbits whose centre is at any depth below the centres of the surface orbits, are given by the following formulæ:

$$b = b_0 \cdot \frac{\varepsilon^{\frac{2\pi(h_0 - h)}{L}} - \varepsilon^{\frac{-2\pi(h_0 - h)}{L}}}{\varepsilon^{\frac{2\pi h_0}{L}} - \varepsilon^{\frac{-2\pi h_0}{L}}} \quad . \quad . \quad . \quad (37)$$

$$a = b_0 \cdot \frac{\varepsilon^{\frac{2\pi(h_0 - h)}{L}} + \varepsilon^{\frac{-2\pi(h_0 - h}{L}}}{\varepsilon^{\frac{2\pi h_0}{L}} - \varepsilon^{\frac{-2\pi h}{L}}} . \quad . \quad . \quad . \quad (38)$$

To illustrate the difference in the rate of decrease of the two axes of the elliptical orbits as the particles considered are further below the surface of the water, the following table has been computed for various shallow-water waves, giving the lengths of the two axes of the orbits for the particles at the surface, at the bottom, and at a point half-way down:

Depth of water in percentage of wave length.	Axes of elliptical orbits at					
	Surface.		Half-depth.		Bottom.	
	Minor Axis.	Major Axis.	Minor Axis.	Major Axis.	Minor Axis.	Major Axis.
.10	1.000	1.796	.475	1.566	0	1.493
.15	1.000	1.858	.448	1.022	0	.919
.20	1.000	1.176	.415	.745	0	.626
.25	1.000	1.091	.877	.576	0	.436
.80	1.000	1.047	.888	.459	0	.811
.85	1.000	1.026	.800	.480	0	.2[illegible]9
.40	1.000	1.018	.263	.809	0	.168

In Fig. 104 is shown the internal structure of a shallow-water wave, whose height and length are the same as those of the deep-sea wave in Fig. 103; but the depth of the water is only 20% of the length of the wave. This figure illustrates the change in eccentricity of the elliptical orbits at various depths. The distorted verticals are seen to have a greater displacement as a whole than those for the corresponding deep-sea wave, accompanied by a slight decrease in their curvature.

The mean elevation of the particles of any surface or subsurface throughout the wave length, above the corresponding still-water level, is $\frac{a\,b}{2R}$, which, as the water becomes deeper in proportion to the wave length, approximates to the previously found value for the deep-sea wave $\frac{r^2}{2R}$.

The energy of this wave is somewhat less than that of the deep-sea wave of the same length and height, the amount of difference depending on the depth of water; and this energy, as in the deep-sea wave, is half kinetic and half potential, the potential half being transmitted forward through the distorted verticals, thus travelling onward with the wave form.

We have now examined into the main facts of trochoidal wave motion, and it only remains to see how far waves in nature agree with those of theory.

Many observations of ocean waves have been made, especially by officers of the French and English navies; and, without entering on the details of their results, it may be broadly stated that the theoretical relation between velocity and length, which was shown to be one of the test conditions of the trochoidal theory, has been found to practically agree with that actually observed in ocean waves in nature.

Many experiments have likewise been made on artificially formed waves; and among such experiments those of the Weber brothers are best known and specially important, on account of the great care and ingenuity displayed and the large number of observations made.

The experimental tanks used by the Webers were between 5 and 6 feet long, 8 to 30 inches deep, and half an inch to an inch wide, the sides being wholly or partly made of glass. They experimented with various fluids, but the results of greatest interest to us are those obtained with water containing a great number of floating particles of the same specific gravity as water. By observing the movements of these particles with the naked eye, or, when necessary, with the microscope, the motion of the particles of water throughout the whole depth was successfully studied. The waves were generated by plunging a glass tube into the water, raising the latter to a certain level in the tube by suction, and then allowing it suddenly to drop.

The profile of the front of the wave was satisfactorily obtained by placing a slate sprinkled with flour in advance of the coming wave, and then suddenly withdrawing the slate as the wave was sweeping along it, the water removing the flour from the immersed portion of the slate. Attempts to obtain the profile of the back of the wave by suddenly plunging the prepared slate into it were not so successful.

The results of their many experiments were embodied by them in the following conclusions:

When a crest is followed by an equal trough, every particle moves in a curve which, as near as the eye can judge, is an ellipse with its major axis horizontal; the motion of the particle when in the highest part of the ellipse being in the same direction as the motion of the wave, and in the opposite direction when at the lowest part of the ellipse.

At different depths the motion was found to be different, the horizontal motion being diminished in some degree for the deeper particles, and the vertical motion being diminished to a greater extent, so that in approaching the bottom the ellipses became very flat, being almost indistinguishable from a horizontal straight line. It was also found that different particles in the same originally vertical line described corresponding parts of their orbits at the same instant of time.

The general velocity of the wave was found to increase with the depth of the fluid and with the size of the wave.

Their work seems to have been done without regard to any particular theory, and the verifications by their experiments of our theoretical results already obtained is thus specially valuable. The contrivance of using a vessel with glass sides and observing the motions of floating particles is one so admirably adapted to overcome the greatest of all the difficulties attending the comparison of a wave theory with experiment—namely, that of ascertaining the laws of movement of individual particles—that these experiments must be accorded a deservedly very great importance.

Taking up now the question of the practical utilization of the energy which we have found to exist in ocean waves, and which for large waves reaches an enormous figure, the subject naturally divides itself into several parts:

1. The various motions of the water which may be utilized for power purposes.

2. The wave-motor proper—that is, the portion of the apparatus in direct contact with the water, and receiving and transmitting the energy thereof; together with the mechanism for transmitting this energy to the pumping or other suitable machinery for utilizing the same.

3. Regulating devices, for obtaining a uniform motion from the more or less irregular and variable action of the waves, as well as for adjusting the apparatus to the state of the tide and condition of the sea.

4. Storage arrangements for ensuring a continuous and uniform output of power during a calm or when the waves are comparatively small.

Taking up first the consideration of the motions that may be utilized for power purposes, we find the following:

1. Vertical rise and fall of particles at and near the surface.

2. Horizontal to-and-fro motion of particles at and near the surface.

3. Varying slope of surface of wave.

4. Impetus of waves rolling up the beach in the form of breakers.

5. Motion of distorted verticals.

All of these motions, except the last one mentioned, have at various times been proposed to be utilized for power purposes; while no attempts seem to have been made to utilize this last-mentioned motion, that of the distorted verticals, which seems

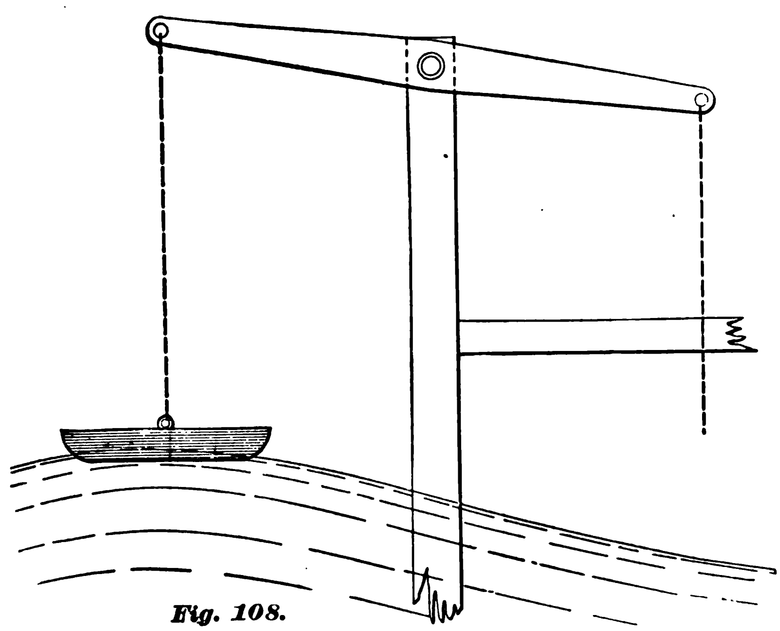

Fig. 108.

the one by far most likely to give efficient results, as will be presently explained.

The wave-motor proper, that is, the portion of the apparatus in direct contact with the water, together with its mechanism for transmitting the energy to the machinery provided for the utilization thereof, may be best examined at the same time with the particular motion of the wave which it is employed to utilize.

The first motion we have mentioned is that of the vertical rise and fall of the particles at and near the surface. The most rational way of utilizing this motion, and the one almost invariably proposed, is by means of a heavy float. The float is either permitted to have a small amount of side motion, the extent of such motion being limited by the length of chains connecting the float to anchors, piles, or other fixed structures, or it is guided in a vertical straight line, or in the arc of a vertical circle. The float is usually hollow, being ballasted if necessary with rock or other heavy materials. It is usually rectangular and flat, or in the shape of a sphere, an ellipsoid, or a cylinder. In Fig. 108 is shown a simple case, the float being secured to a rope which is attached to one end of a walking beam, the other end of the latter being connected by a second rope to suitable

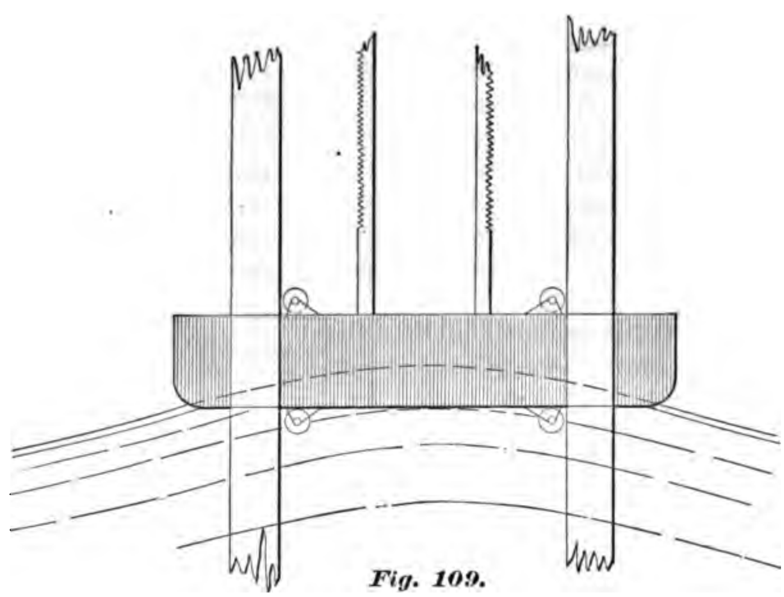

Fig. 109.

mechanism for utilizing the power. As the float rises on the wave, the slack of its rope is taken up by suitable ratchet arrangements, and when the wave falls, the weight of the float is available for the production of useful work. In Fig. 109 the float is guided by guide rollers travelling along vertical piles. This float is provided with vertical straight racks which transmit their motion to pinions located on the structure above. The racks being rigid, power can be taken off on both the up and down strokes, though of course no increase of power is thereby gained over the simple scheme of taking off the power during the down stroke only. In Fig. 110 is shown a very simple device, a cylindrical float rising and falling on a central guide pile, the power being transmitted by means of vertical racks. A modification of this has the pump, which is operated by the rise and fall of the float, in the upper part of the guide pile itself, the object being to get all the principal parts as nearly as possible in line. In Fig. 111 is shown an ellipsoidal float, held at one end of a frame, the other end of which is pivoted to a rigid structure at some point above the water. The motion of the float is transmitted by a rope to suitable pumping mechanism. This device is practically employed at some points

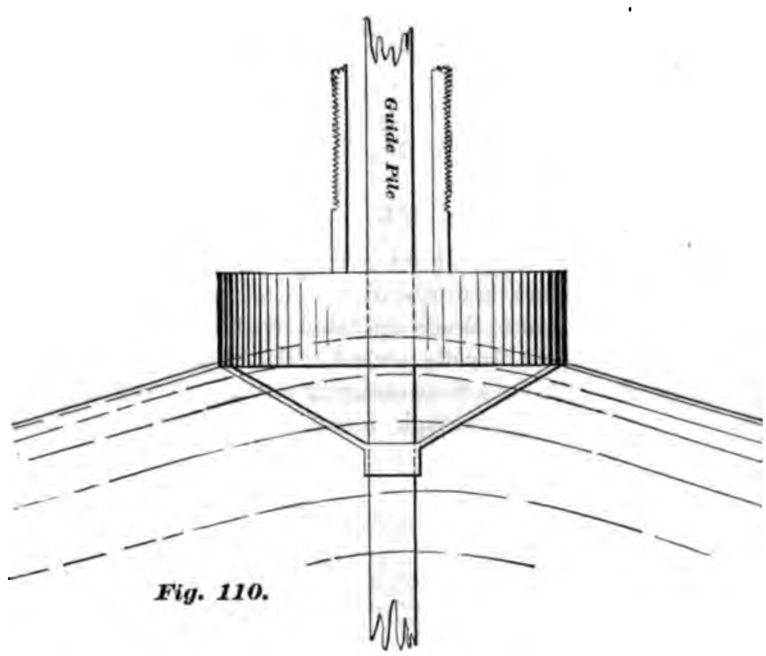

Fig. 110.

on the Eastern coast to pump salt water for street sprinkling purposes. Fig. 112 shows a somewhat similar arrangement, the frame carrying the cylindrical float being pivoted below the surface of the water, and the shaft to which the frame is attached actuating suitable mechanism by means of a geared sector. This float is, in addition, provided with a curved lip to somewhat confine the water and thus get the full benefit of its momentum. In Fig. 113 is shown a spherical float attached to a rope leading downward. The actual rise and fall of the float causes a corresponding motion of the rope, which leads through a sheave below to proper mechanism on shore. The spherical float shown in Fig. 114 is anchored by means of a rope leading downward and through a sheave to the shore. As the float is forced upward by the rise of the wave, it is also compelled to approach the vertical line passing through the lower sheave, and thus causes a pull on the power rope, the power being thus trans-

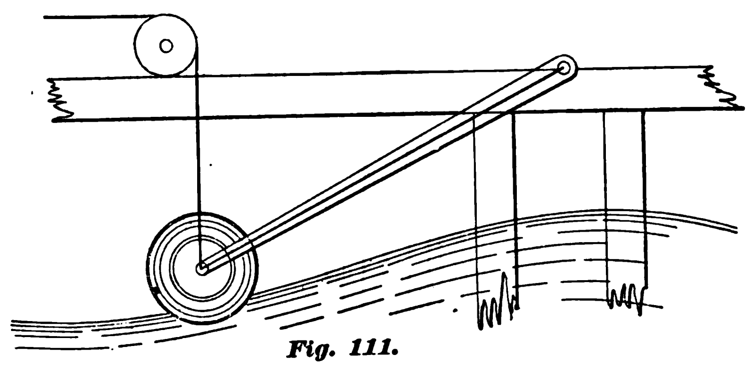

Fig. 111.

mitted by a sort of toggle-joint arrangement, the efficiency of which, to say the least, is doubtful.

The main objection to floats operated by this vertical motion of the water may be briefly stated. The quantity of power that can be obtained by a float covering any given area of water, and rising and falling through a certain distance, is directly

Fig. 112.

proportional to the weight of such float, as the number of foot-pounds of energy for each wave is simply the weight of the float in pounds multiplied by its rise or fall in feet. With a wave of given height, then, the amount of power obtainable from such float can only be increased by increasing the weight of the latter. This weight may be increased in either of two ways: (1) By making the float heavier per square foot of water area covered, as by using heavier material, or by increasing the amount of ballast carried; (2) by keeping the weight per square foot unaltered, but increasing the area of water covered by the float. Either of these methods is, however, attended with a loss of efficiency. An increase in weight of float means

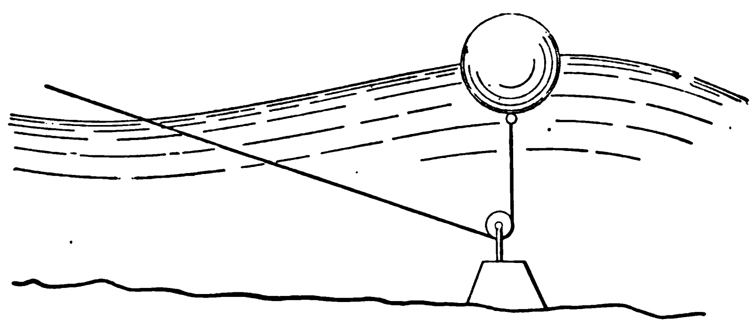

Fig. 113.

an increase in submerged depth and in inertia; and the inertia of such heavy float could not be overcome with sufficient rapidity to cause it to rise to the whole height of the wave, thus decreasing its efficiency. If, on the other hand, the float be made light to reduce its enertia, the possible amount of power to be transmitted thereby would again be correspondingly decreased.

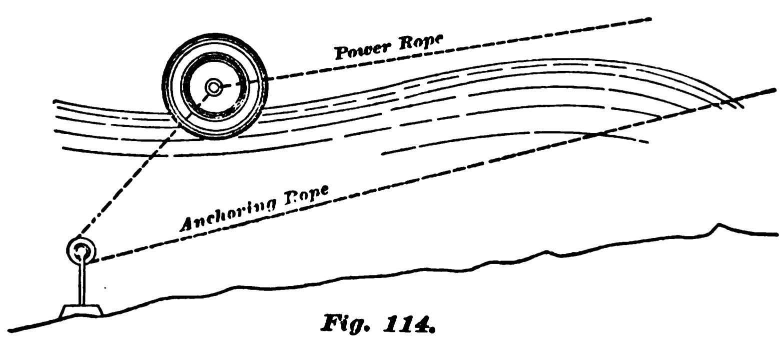

Fig. 114.

Furthermore, as the area of water covered by such float is made greater, the particles of water in contact with different points of its length would be in different phases, so that the *mean* rise of such water would be less. Thus it is evident that if the float were just half as long as the wave, it would have a tilting motion, but no rise or fall as a whole whatever; and if it were longer than the wave, practically no power at all could be obtained thereby. But as the length of the waves varies from day to day, a float which would be fairly efficient on one day with a certain series of waves might be utterly inefficient with a series of waves of a different length some other day.

The next motion to be considered is the horizontal to and fro motion of particles at and near the surface. The simplest arrangement for utilizing this motion consists in suspending a vertical flat vane from some point above the water, the lower end of such vane dipping into the water to a certain limited depth and being actuated by the horizontal component of the motion of the particles at and near the surface. Such arrange-

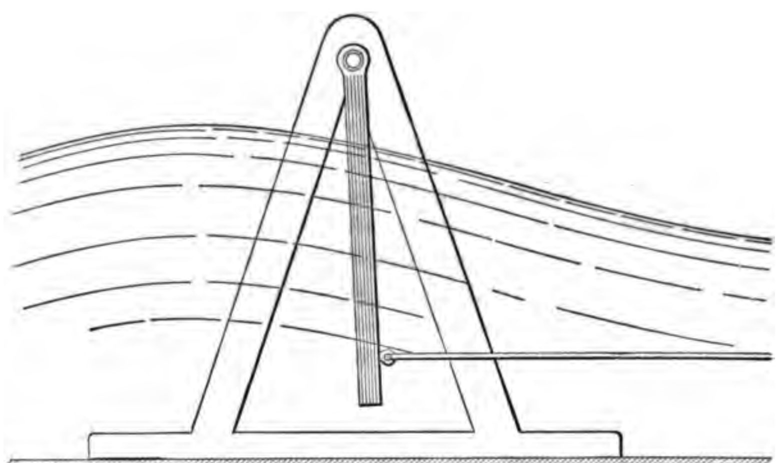

Fig. 115.

ments are shown in Figs. 115 and 116, the vane in the latter actuating suitable mechanism by means of a connecting rod attached to its prolongation above the point of support, while in the former it operates a submerged pump directly, by means of a connecting rod attached to its lower end.

That some power can be obtained by these devices is beyond question, but their efficiency is hampered by the fact that while at first sight it would seem that the amount of power thereby obtained would, with a given breadth of vane, be directly proportional to the depth of immersion of the lower end of the vane, yet it is evident that the deeper the immersion of the vane the less efficient the apparatus would be. For the vane has an angular motion about its point of support, its lower immersed end moving through a greater distance than the portion at the surface of the water. But we have above seen that the horizontal motion of the particles is greatest at the surface and becomes less as the respective particles are further below the surface. The relative amounts of motion of the upper and lower portions of the vane are thus the reverse of the natural relative motions of the particles of water in contact with those portions; so that as such vane extends deeper into the water, it becomes less efficient. If the vane be extended to the bottom, its motion would be much reduced in the case of a shallow-water wave, and it would come to an absolute standstill in the

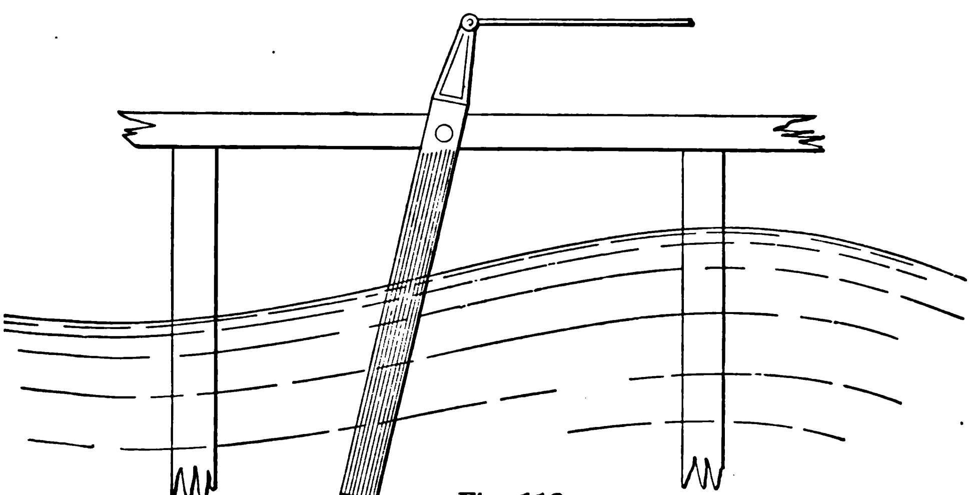

Fig. 116.

case of the deep-sea wave. Instead of the vanes just described, it has also been proposed to employ a float, rigidly secured to a long arm extending to and pivoted at some point of a fixed structure overhead, or extending downward and pivoted at some point near the bottom of the water, but in each of these cases the utilization of the motion of the surface particles only is contemplated. A somewhat more complicated arrangement is shown in Fig. 117, in which a peculiarly shaped float is supported by crossed suspension rods. The longitudinal concavities of the surface of the float are alleged to be "of substantially the form the waves assume as they approach the shore, so that the full force thereof is utilized better than if sharp corners were presented."

In Fig. 118 is shown an arrangement of a cylindrical float, suspended by a number of ropes attached to the float at such points as to utilize not only the rise and fall of the float; but also the horizontal to and fro motion of the same. The extreme motion of the float is limited by chains attached thereto, as shown.

The next motion to consider is that due to the varying slope

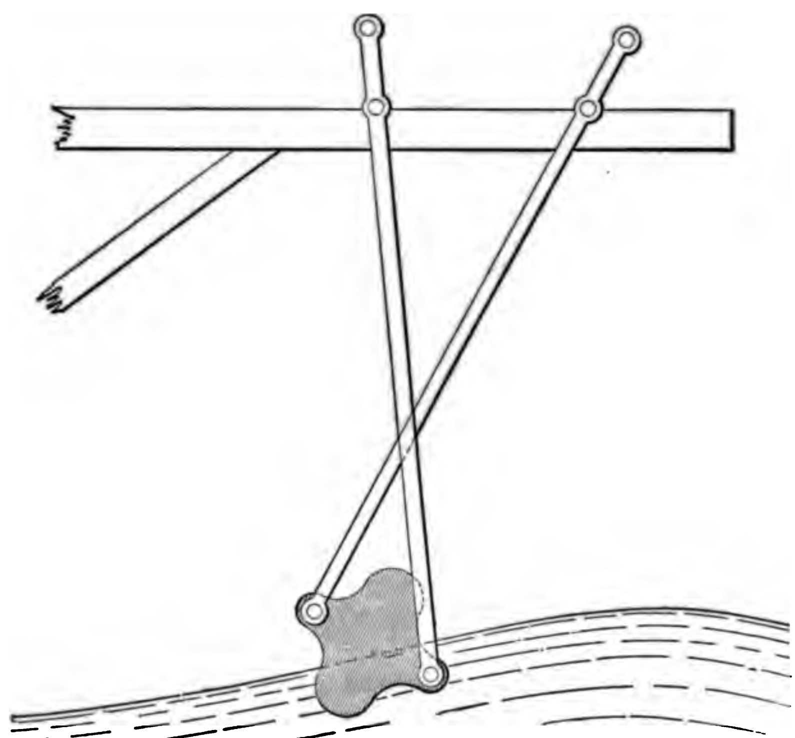

Fig. 117.

of the wave. In Fig. 119 is shown a float which lies on the wave and changes its inclination to the horizontal in accordance with the varying slope of the wave. A rigid arm is firmly secured to the float, extending across the latter; and ropes attached to the upper and lower ends of this arm transmit the power to mechanism on the shore or on a suitable structure erected in the water near the shore, the float itself being prevented from moving shoreward by an anchoring rope. The objections to this scheme are two-fold. In the first place, the curves in which the power ropes hang have different deflections according to the strain to which they are subjected. As the float becomes inclined by the action of the wave slope, the strain on one of the ropes increases, while that on the other decreases, the deflection of the former becoming less and that of the latter becoming greater; and the change in deflection corresponding to any increase in strain must be produced before the latter can be transmitted. The motion of the float must thus first take up a certain amount of slack in the rope, so as to decrease its deflection to that corresponding to the strain to be

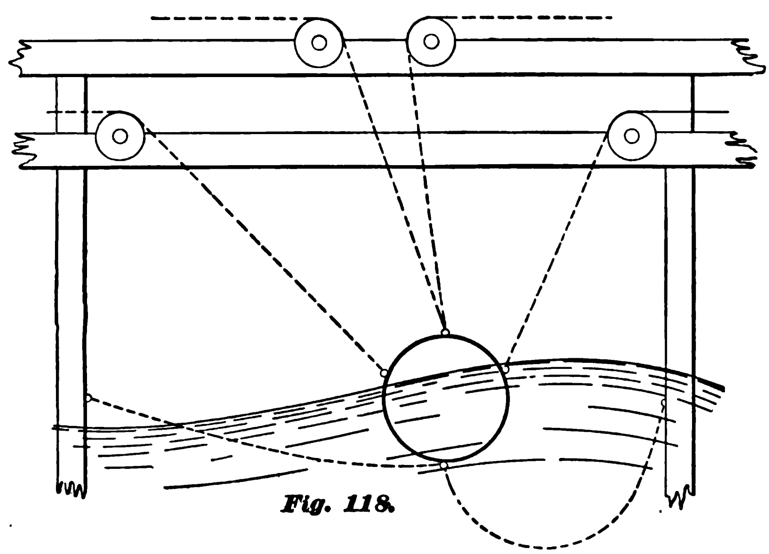

Fig. 118.

transmitted; but unless the waves were very steep, the motion of the float would probably hardly suffice to do more than take up this slack; and in such case no power, or at any rate very little power, would be transmitted. This defect in the arrangement could, however, be much lessened by placing the float much nearer the structure supporting the mechanism, so that shorter ropes or even rods could be employed for transmitting the power. But there is another and more vital objection. To obtain the largest amount of power from this arrangement, the float should be exactly half the length of the wave. But as the waves vary considerably in length from day to day, it follows that the float, while quite efficient in waves to which its length was suited, would lose efficiency among larger waves, and come almost to a standstill among much shorter waves.

Somewhat akin to the above is the device proposed for employing the varying angle between two portions of the surface some distance apart in the direction of the length of the wave. It consists (Fig. 120) of two floats, preferably pontoons containing the mechanism for utilizing the power, these floats

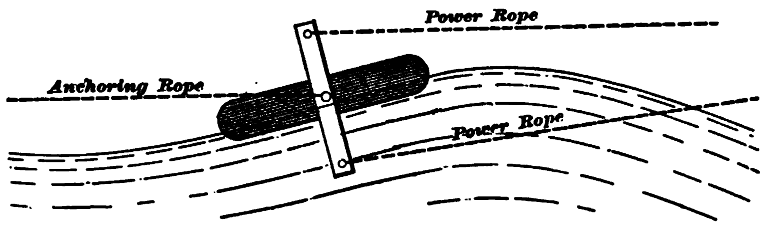

Fig. 119.

being hinged together so as to be capable of motion relative to each other about a horizontal axis. As the floats ride on opposite sides of the crest of a wave, their outer ends are lower than their inner ends, while in the trough of a wave this condition is reversed. The angular motion of the floats relative to each other is employed to operate a ratchet wheel as shown, from which the power is transmitted to any suitable mechanism. The difficulty with this device is also due to the varying

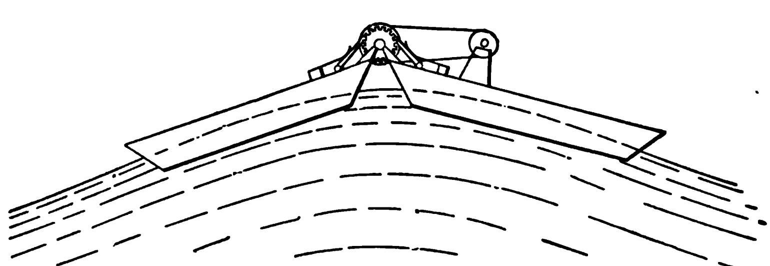

Fig. 120.

length of the waves from day to day. It is evident that if the length of the floats or pontoons be too short in comparison with the length of the wave, the angle between them will be very slight and but little power will be obtained. On the other hand, if they are too long, the angle is again diminished and the same difficulty presents itself. Thus a pair of such pontoons, of proper lengths to give their greatest efficiency for a certain length of wave, would be much less efficient for a wave one-half as long or double as long.

In Fig. 121 is shown an arrangement for utilizing all three of the motions above discussed. It consists of a rectangular float,

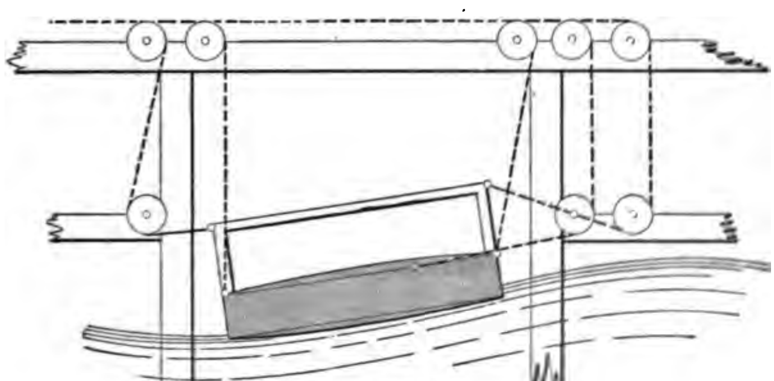

Fig. 121.

surmounted by a frame-work rigidly attached to the same. A number of ropes are attached to various points of the float and of the frame-work as shown, leading over sheaves so placed that power will be transmitted not only by the rise and fall of the float, but by its tilting action on the slope of the wave, and by its horizontal to and fro motion. While superior, in some

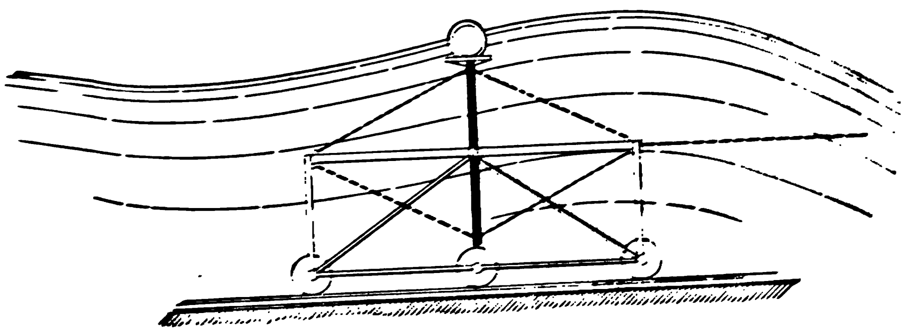

Fig. 122.

respects, to some of the simpler schemes above explained, it partakes of many of their disadvantages, and has the additional one of being somewhat more cumbersome and complicated.

Several radically distinct methods have been proposed for utilizing the impetus of breakers rolling up the beach. In Fig.

122 is shown a nearly vertical vane held in a frame-work which is provided with wheels at the bottom and with ropes leading to suitable mechanism on shore. These wheels run on rails laid down the slope of the beach, or may even run on the slope of the beach itself, if the latter be hard and smooth. As the breaker strikes this vane, it drives it rapidly up the slope of the beach, and the slack of the rope is taken up by suitable ratchets or other contrivances. As the water recedes, the weight of the vane and carriage causes the latter to run down the slope of the beach, and useful work may be performed by utilizing the pull on the rope. Some little difficulty would probably be experienced with this apparatus on account of the sand filling in along the track, which would probably finally result in throwing the

Fig. 123.

apparatus off the track. It labors under the disadvantage that much of the inherent energy of the smooth waves is dissipated in the eddies of the breakers before it reaches the vane, and that the delivery of the energy of the breakers to the vane by what would practically be a sharp blow is naturally much less efficient than in cases where this transfer of energy is more gradual.

An entirely different method is shown in Fig. 123. The mass of water is allowed to enter a large, strongly built chamber, which decreases in both height and width toward its shoreward end. At this end a non-return valve *a* is provided, beyond which is a closed chamber. In the top of the latter is the non-return valve *b*, above which is an air chamber and a pipe leading to the water reservoir. When the wave enters the receiving chamber it has a certain velocity; and as the cross-sectional area of this chamber becomes less, the velocity of the water must increase until at the inner end it becomes sufficient to pass through the non-return valves *a* and *b*, against the pressure due to the height of the water in the reservoir. While this apparatus has the advantage of simplicity and few working parts, it labors under the disadvantage that much of the energy of the waves is dissipated by the time they reach the apparatus. There is also much internal friction and great liability to filling up with sand. It has the further disadvantage of not being adjustable to rise and fall of the tide or the condition of the sea.

Having now briefly considered the principal methods hitherto proposed for utilizing the power of the waves, and examined into the causes of any inefficiency that may have been found to exist, we will next take up the consideration of the utilization of the motion of the distorted verticals; and as this motion seems the one most likely to give efficient results, we shall discuss the same at length.

It may be well, in the first place, to summarize here the main features of the motion of the individual particles of water, as already investigated and determined in the foregoing pages.

Each individual particle of water in a trochoidal wave moves in an elliptical orbit, whose major axis is horizontal and the plane of which is vertical and perpendicular to the wave ridge or crest, the motion of the particle in the upper portion of its orbit being in the direction of advance or propagation of the wave itself, and in the lower portion of its orbit, in the opposite direction. The eccentricity of these ellipses depends on the relation between the length of the wave and the total depth of the water. As this depth of water increases in proportion to the length of the wave, the eccentricity of the ellipses becomes less; and they finally become circles when the total depth of water becomes infinite; practically they cannot be distinguished from circles when the depth of water exceeds about one-half the length of the wave. On the other hand, as the total depth of the water becomes less, the eccentricity of the ellipses increases, their horizontal axes becoming much greater than their vertical axes as the water becomes very shallow.

In any given wave, all the particles which were originally in the same vertical straight line while the water was at rest describe their respective orbits in the same time and occupy the same phase in these orbits at the same instant; but the orbits themselves become smaller as the distance of the respective particles below the surface increases. In the case of the deep-sea wave with circular orbits, the circles thus decrease in diameter with increase of depth of the respective particles. In the case of the shallow-water wave with elliptical orbits, the ellipses also decrease in size with increase of depth, but their focal distance remains constant. Their vertical minor axes, therefore, decrease faster than their horizontal major axes, thus rendering the ellipses flatter, as well as smaller, with increased depth of particles. At the bottom the vertical axis disappears entirely and the particles there move in horizontal straight lines, whose length is equal to the focal distance of the elliptical orbits of the higher particles. It is to be noted that the above is strictly true only for a perfect liquid with a horizontal and frictionless bottom; practically the motion is somewhat modified by the actual slope and roughness of the latter.

The horizontal component of the motion of a particle at any specified depth is equal to the diameter of the respective orbit circle or the major axis of the respective orbit ellipse, as the case may be, and is thus not the same at all depths, but is less as the particle considered is further from the surface. Hence it follows that a set of particles, originally in the same vertical straight line when the water is at rest, does not remain in a vertical line during the passage of the wave; so that the line connecting a set of such particles, while vertical and straight in still water, becomes distorted, as well as displaced, during the passage of the wave, its upper portion moving further and more rapidly than its lower portion.

Referring to Fig. 103, the line AA is the original still-water level, aaa is the profile of an assumed deep-sea wave, and the dotted line above AA is the centre line of the orbit circles for the surface particles. The circular orbits of some of these particles are shown dotted, the position of the orbit radius in each case showing the phase of the respective particle. Similar elements are shown for virtual sub-surfaces at successively equal depths below the surface, thus illustrating the gradual decrease in motion with increasing depth below the surface. The lines *bc*, *de*, etc., are the originally vertical straight lines connecting a set of particles in still water, previously referred to, which become displaced and distorted during the passage of the wave. In the particular phase represented in Fig. 103, they occupy the positions $b'c'$, $d'e'$, etc.; and during the passage of one complete wave, each such originally vertical line passes by a continuous motion through the positions and shapes shown in the diagram, and through all intermediate positions and shapes. Each such originally vertical line moves as a whole in alternately opposite directions, its upper end having the greater and more rapid motion and always curving toward the nearest wave crest, be the same approaching or receding, the line as a whole passing through a vertical phase at the passage of each successive crest and trough of the wave across such line.

By substituting for the orbit circles in Fig. 103, the orbit ellipses corresponding to any particular shallow-water wave, we obtain the corresponding diagram for that wave. In Fig. 104 is thus constructed the corresponding diagram for a shallow-water wave of the same length and height as the deep-sea wave of Fig. 103, but the depth of water being only 20% of the length of the wave.

Considering now the motion of one of these distorted verticals in a deep-sea or shallow-water wave, its extreme positions are evidently found by drawing curves passing through and connecting the ends of the horizontal axes of the orbits at different depths, the line passing from one of these extreme positions to the other with its upper end describing the upper half of the surface orbit during the forward vibration and the lower half of the surface orbit during the backward vibration. Figs. 124 to 130 show the range of motion and the extreme positions of the distorted verticals corresponding to a number of waves of the same height and length as those shown in Figs. 103 and 104, the depth of water being, however, successively less in the several figures. Thus Fig. 124 corresponds to the deep-sea wave of Fig. 103; Fig. 125 is for a depth of water of 40% of the length of the wave; Fig. 126 for 35%; Fig. 127 for 30%; Fig. 128 for 25%; Fig. 129 for 20% (corresponding to the wave of Fig. 104); and Fig. 130 for 15%. The left-hand diagram of each of these figures corresponds to the forward vibration, and the right-hand diagram to the backward vibration. It will be noticed that, as the depth of water diminishes in proportion to the length of the wave, the maximum curvature of the distorted verticals gradually decreases, which is accompanied by an increase in their translation or displacement as a whole. Considering the motion of each such distorted vertical in any of the waves corresponding to the length, height, and depth of water assumed in Figs. 124 to 130, the entire area swept over by the distorted vertical during the forward vibration is that shown hatched in the left-hand diagram of the corresponding figure, having for its upper boundary the upper half of the surface orbit, while the area swept over by each such vertical during the backward vibration is that shown hatched in the right-hand diagram of the corresponding figure, being a somewhat similar figure, but having for its upper boundary a curve practically agreeing with the lower half of the surface orbit.

Considering, in each of the waves to which Figs. 124 to 130 respectively refer, a theoretical surface which was originally in still water a vertical plane perpendicular to the direction of advance of the assumed wave, each such surface during the passage of the wave will move in alternately opposite directions, but its upper end will have the greater and more rapid motion, curving in alternately opposite directions toward the nearest wave crest. The surface as a whole will, during the forward vibration, sweep through a volume whose section is shown by the hatched portion of the left-hand diagram of the corresponding figure, and during the backward vibration, through a volume whose section is shown by the hatched portion of the left-hand diagram of the same figure.

If we conceive this theoretical surface, in any case, to be replaced by an extremely thin and flexible metallic vane, the latter will evidently, during the passage of each wave, be subjected to a similar displacement as a whole in alternately opposite directicns, and to a bending in alternately opposite directions toward the nearest wave crest. If such vane be placed in still water and caused by suitable mechanism to assume and pass through all the successive positions and shapes of a distorted vertical corresponding to any particular trochoidal deep or shallow-water wave, a certain expenditure of power will be required and the corresponding wave will be produced. Conversely, it follows that, by means of suitable mechanism, a certain amount of power can be derived from a vane of this kind which, by the passage of a wave, is caused to undergo the motions described.

Should such vane be made rigid instead of flexible, it would still have approximately the same motion of translation as the flexible vane, but instead of bending in detail to conform to the

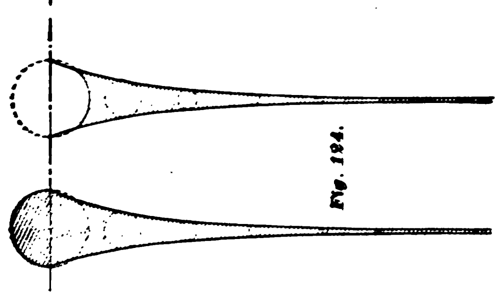

Fig. 124.

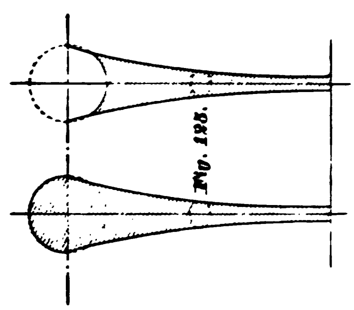

Fig. 125.

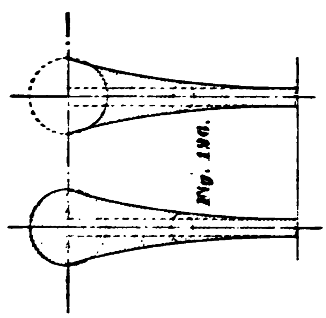

Fig. 126.

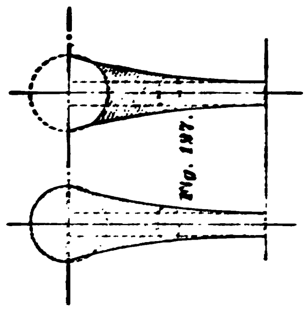

Fig. 127.

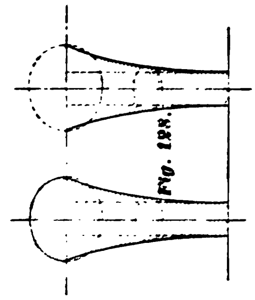

Fig. 128.

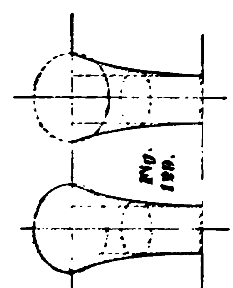

Fig. 129.

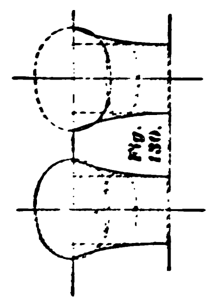

Fig. 130.

than the wave, practically no power at all could be obtained thereby. But as the length of the waves varies from day to day, a float which would be fairly efficient on one day with a certain series of waves might be utterly inefficient with a series of waves of a different length some other day.

The next motion to be considered is the horizontal to and fro motion of particles at and near the surface. The simplest arrangement for utilizing this motion consists in suspending a vertical flat vane from some point above the water, the lower end of such vane dipping into the water to a certain limited depth and being actuated by the horizontal component of the motion of the particles at and near the surface. Such arrange-

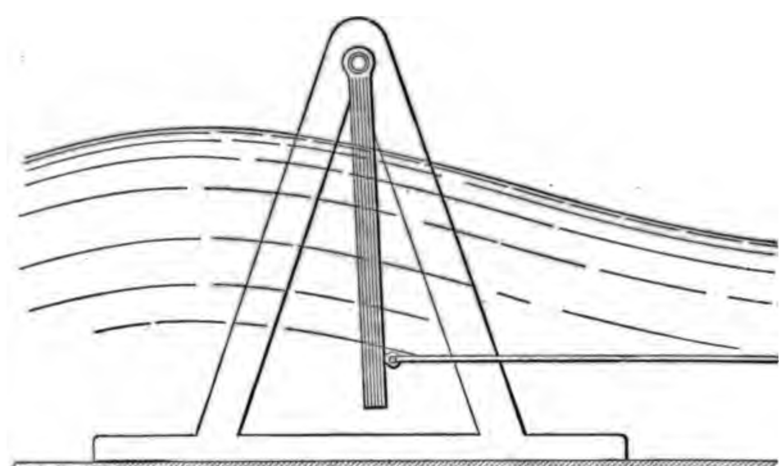

Fig. 115.

ments are shown in Figs. 115 and 116, the vane in the latter actuating suitable mechanism by means of a connecting rod attached to its prolongation above the point of support, while in the former it operates a submerged pump directly, by means of a connecting rod attached to its lower end.

That some power can be obtained by these devices is beyond question, but their efficiency is hampered by the fact that while at first sight it would seem that the amount of power thereby obtained would, with a given breadth of vane, be directly proportional to the depth of immersion of the lower end of the vane, yet it is evident that the deeper the immersion of the vane the less efficient the apparatus would be. For the vane has an angular motion about its point of support, its lower immersed end moving through a greater distance than the portion at the surface of the water. But we have above seen that the horizontal motion of the particles is greatest at the surface and becomes less as the respective particles are further below the surface. The relative amounts of motion of the upper and lower portions of the vane are thus the reverse of the natural relative motions of the particles of water in contact with those portions; so that as such vane extends deeper into the water, it becomes less efficient. If the vane be extended to the bottom, its motion would be much reduced in the case of a shallow-water wave, and it would come to an absolute standstill in the

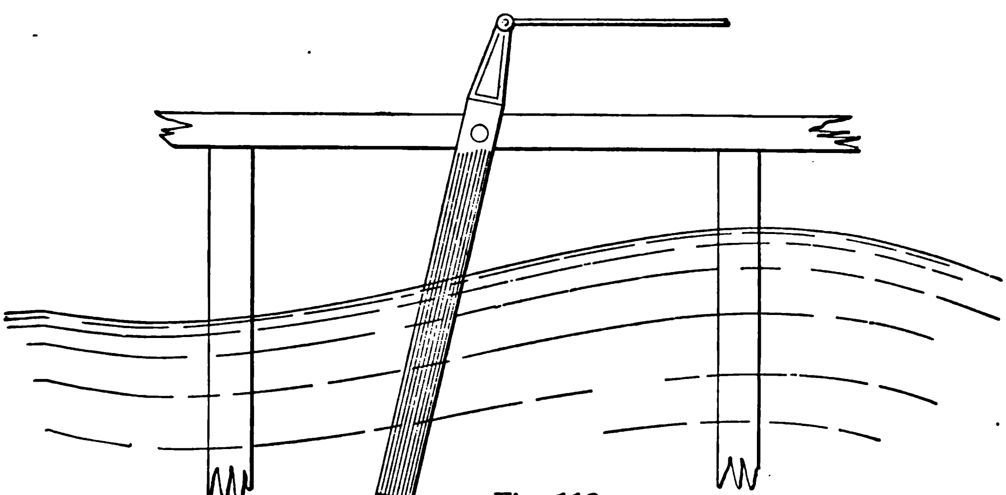

Fig. 116.

case of the deep-sea wave. Instead of the vanes just described, it has also been proposed to employ a float, rigidly secured to a long arm extending to and pivoted at some point of a fixed structure overhead, or extending downward and pivoted at some point near the bottom of the water, but in each of these cases the utilization of the motion of the surface particles only is contemplated. A somewhat more complicated arrangement is shown in Fig. 117, in which a peculiarly shaped float is supported by crossed suspension rods. The longitudinal concavities of the surface of the float are alleged to be "of substantially the form the waves assume as they approach the shore, so that the full force thereof is utilized better than if sharp corners were presented."

In Fig. 118 is shown an arrangement of a cylindrical float, suspended by a number of ropes attached to the float at such points as to utilize not only the rise and fall of the float; but also the horizontal to and fro motion of the same. The extreme motion of the float is limited by chains attached thereto, as shown.

The next motion to consider is that due to the varying slope

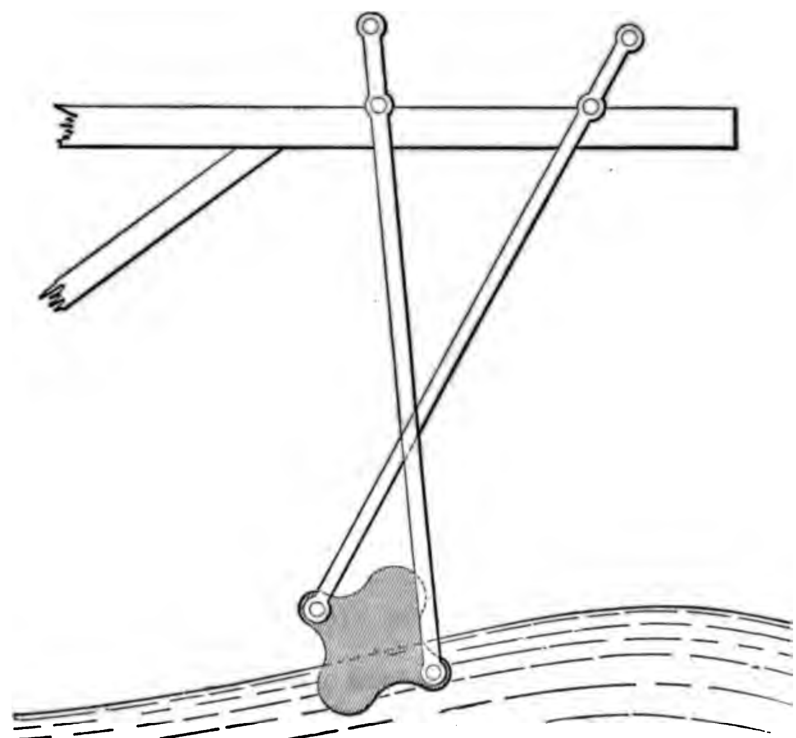

Fig. 117.

of the wave. In Fig. 119 is shown a float which lies on the wave and changes its inclination to the horizontal in accordance with the varying slope of the wave. A rigid arm is firmly secured to the float, extending across the latter; and ropes attached to the upper and lower ends of this arm transmit the power to mechanism on the shore or on a suitable structure erected in the water near the shore, the float itself being prevented from moving shoreward by an anchoring rope. The objections to this scheme are two-fold. In the first place, the curves in which the power ropes hang have different deflections according to the strain to which they are subjected. As the float becomes inclined by the action of the wave slope, the strain on one of the ropes increases, while that on the other decreases, the deflection of the former becoming less and that of the latter becoming greater; and the change in deflection corresponding to any increase in strain must be produced before the latter can be transmitted. The motion of the float must thus first take up a certain amount of slack in the rope, so as to decrease its deflection to that corresponding to the strain to be

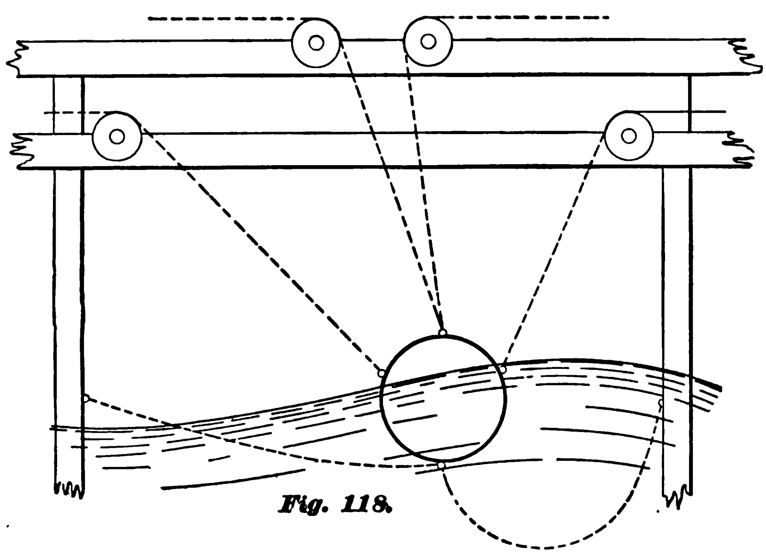

Fig. 118.

transmitted; but unless the waves were very steep, the motion of the float would probably hardly suffice to do more than take up this slack; and in such case no power, or at any rate very little power, would be transmitted. This defect in the arrangement could, however, be much lessened by placing the float much nearer the structure supporting the mechanism, so that shorter ropes or even rods could be employed for transmitting the power. But there is another and more vital objection. To obtain the largest amount of power from this arrangement, the float should be exactly half the length of the wave. But as the waves vary considerably in length from day to day, it follows that the float, while quite efficient in waves to which its length was suited, would lose efficiency among larger waves, and come almost to a standstill among much shorter waves.

Somewhat akin to the above is the device proposed for employing the varying angle between two portions of the surface some distance apart in the direction of the length of the wave. It consists (Fig. 120) of two floats, preferably pontoons containing the mechanism for utilizing the power, these floats

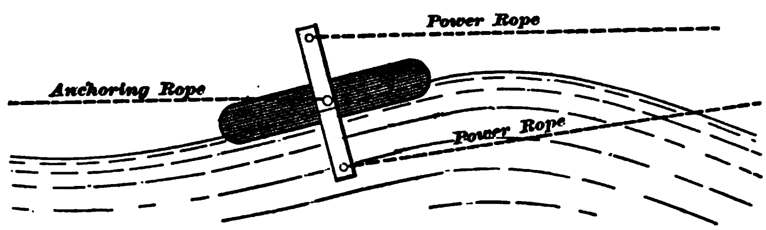

Fig. 119.

being hinged together so as to be capable of motion relative to each other about a horizontal axis. As the floats ride on opposite sides of the crest of a wave, their outer ends are lower than their inner ends, while in the trough of a wave this condition is reversed. The angular motion of the floats relative to each other is employed to operate a ratchet wheel as shown, from which the power is transmitted to any suitable mechanism. The difficulty with this device is also due to the varying

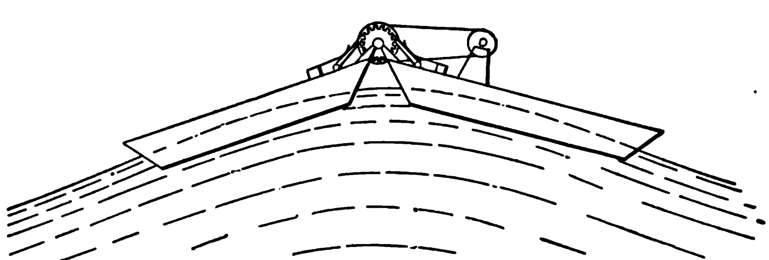

Fig. 120.

length of the waves from day to day. It is evident that if the length of the floats or pontoons be too short in comparison with the length of the wave, the angle between them will be very slight and but little power will be obtained. On the other hand, if they are too long, the angle is again diminished and the same difficulty presents itself. Thus a pair of such pontoons, of proper lengths to give their greatest efficiency for a certain length of wave, would be much less efficient for a wave one-half as long or double as long.

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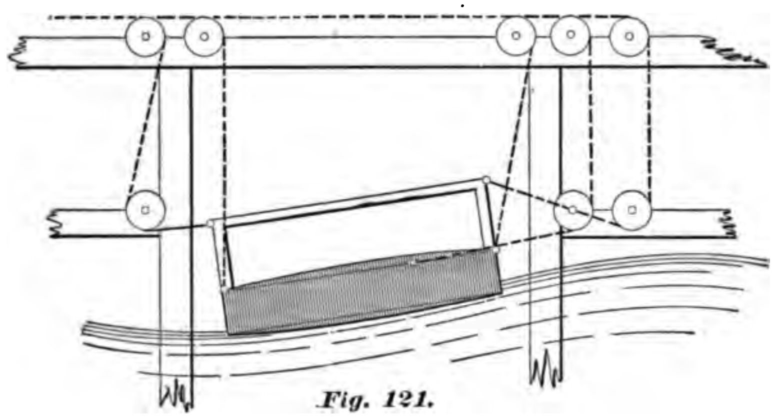

Fig. 121.

surmounted by a frame-work rigidly attached to the same. A number of ropes are attached to various points of the float and of the frame-work as shown, leading over sheaves so placed that power will be transmitted not only by the rise and fall of the float, but by its tilting action on the slope of the wave, and by its horizontal to and fro motion. While superior, in some

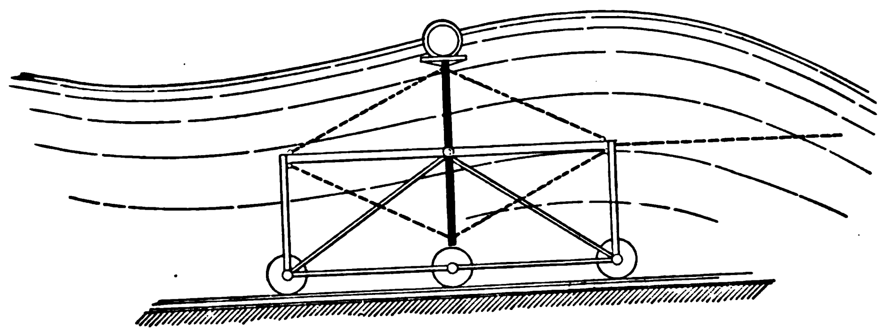

Fig. 122.

respects, to some of the simpler schemes above explained, it partakes of many of their disadvantages, and has the additional one of being somewhat more cumbersome and complicated.

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themselves become smaller as the distance of the respective particles below the surface increases. In the case of the deep-sea wave with circular orbits, the circles thus decrease in diameter with increase of depth of the respective particles. In the case of the shallow-water wave with elliptical orbits, the ellipses also decrease in size with increase of depth, but their focal distance remains constant. Their vertical minor axes, therefore, decrease faster than their horizontal major axes, thus rendering the ellipses flatter, as well as smaller, with increased depth of particles. At the bottom the vertical axis disappears entirely and the particles there move in horizontal straight lines, whose length is equal to the focal distance of the elliptical orbits of the higher particles. It is to be noted that the above is strictly true only for a perfect liquid with a horizontal and frictionless bottom; practically the motion is somewhat modified by the actual slope and roughness of the latter.

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Having now briefly considered the principal methods hitherto proposed for utilizing the power of the waves, and examined into the causes of any inefficiency that may have been found to exist, we will next take up the consideration of the utilization of the motion of the distorted verticals; and as this motion seems the one most likely to give efficient results, we shall discuss the same at length.

It may be well, in the first place, to summarize here the main features of the motion of the individual particles of water, as already investigated and determined in the foregoing pages.

Each individual particle of water in a trochoidal wave moves in an elliptical orbit, whose major axis is horizontal and the plane of which is vertical and perpendicular to the wave ridge or crest, the motion of the particle in the upper portion of its orbit being in the direction of advance or propagation of the wave itself, and in the lower portion of its orbit, in the opposite direction. The eccentricity of these ellipses depends on the relation between the length of the wave and the total depth of the water. As this depth of water increases in proportion to the length of the wave, the eccentricity of the ellipses becomes less; and they finally become circles when the total depth of water becomes infinite; practically they cannot be distinguished from circles when the depth of water exceeds about one-half the length of the wave. On the other hand, as the total depth of the water becomes less, the eccentricity of the ellipses increases, their horizontal axes becoming much greater than their vertical axes as the water becomes very shallow.

In any given wave, all the particles which were originally in the same vertical straight line while the water was at rest describe their respective orbits in the same time and occupy the same phase in these orbits at the same instant; but the orbits to the length, height, and depth of water assumed in Figs. 124 to 130, the entire area swept over by the distorted vertical during the forward vibration is that shown hatched in the left-hand diagram of the corresponding figure, having for its upper boundary the upper half of the surface orbit, while the area swept over by each such vertical during the backward vibration is that shown hatched in the right-hand diagram of the corresponding figure, being a somewhat similar figure, but having for its upper boundary a curve practically agreeing with the lower half of the surface orbit.

Considering, in each of the waves to which Figs. 124 to 130 respectively refer, a theoretical surface which was originally in still water a vertical plane perpendicular to the direction of advance of the assumed wave, each such surface during the passage of the wave will move in alternately opposite directions, but its upper end will have the greater and more rapid motion, curving in alternately opposite directions toward the nearest wave crest. The surface as a whole will, during the forward vibration, sweep through a volume whose section is shown by the hatched portion of the left-hand diagram of the corresponding figure, and during the backward vibration, through a volume whose section is shown by the hatched portion of the left-hand diagram of the same figure.

If we conceive this theoretical surface, in any case, to be replaced by an extremely thin and flexible metallic vane, the latter will evidently, during the passage of each wave, be subjected to a similar displacement as a whole in alternately opposite directicns, and to a bending in alternately opposite directions toward the nearest wave crest. If such vane be placed in still water and caused by suitable mechanism to assume and pass through all the successive positions and shapes of a distorted vertical corresponding to any particular trochoidal deep or shallow-water wave, a certain expenditure of power will be required and the corresponding wave will be produced. Conversely, it follows that, by means of suitable mechanism, a certain amount of power can be derived from a vane of this kind which, by the passage of a wave, is caused to undergo the motions described.

Should such vane be made rigid instead of flexible, it would still have approximately the same motion of translation as the flexible vane, but instead of bending in detail to conform to the

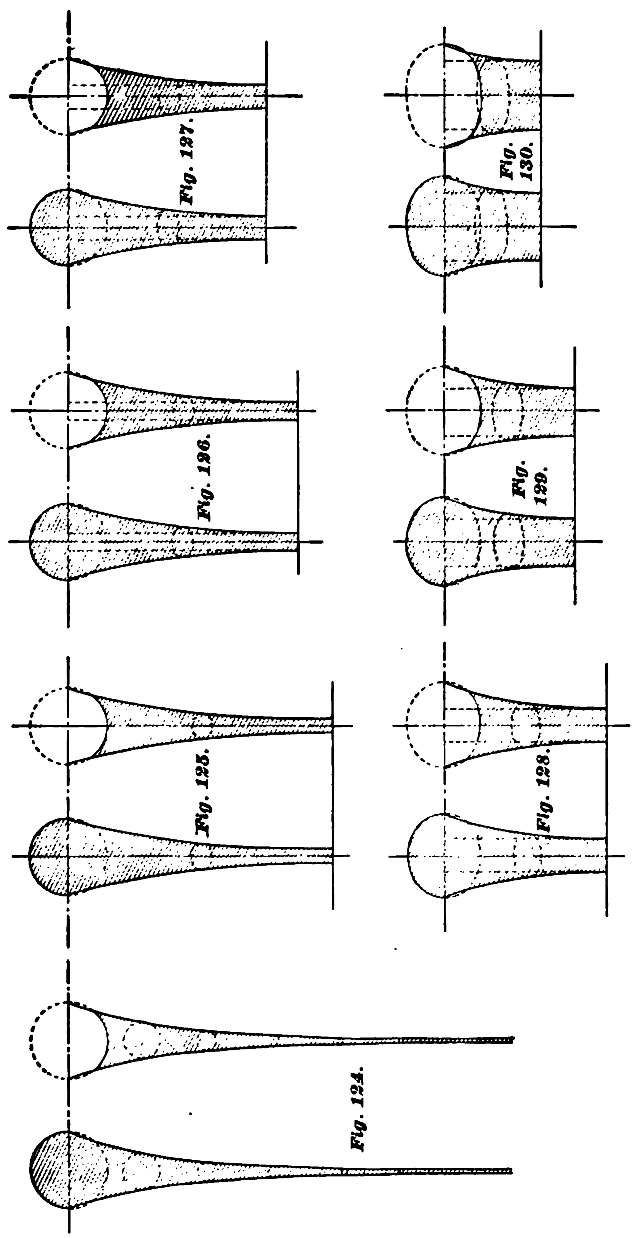

Fig. 124.
Fig. 125.
Fig. 126.
Fig. 127.
Fig. 128.
Fig. 129.
Fig. 130.

exact shape of the distorted verticals, it would slant as a whole toward the nearest wave crest, taking at any instant the average direction of their curvature along its length. While a flexible vane would theoretically be best as interfering least with the normal structure and movement of the wave, yet the increased complexity in mechanism required would more than counterbalance the gain in power, so that for practical purposes a rigid vane is to be preferred. Such vane should, in general, be made as light and thin as is consistent with proper strength and rigidity, and should extend from somewhat above the surface of the water to near the bottom. It is also to be noted that the above description of the movements of the wave applies strictly only to the more or less regular swell outside the breakers; and it is in these waves outside the breakers that, for greatest efficiency, such vane with its mechanism should be located.

In accordance with the principles just set forth it has been proposed jointly by the author and the late Mr. Richard Gatewood, U. S. N., to utilize the movements of the distorted verticals by opposing thereto movable vanes pivoted or supported either at their ends or other points of their lengths, so as to move in general accordance with the motion of the distorted verticals, and thus to receive and transmit the maximum effect of the wave movement.

The preferable method of suspending the vane, in any actual case, depends on the depth of the water and on the range of variation in the usual magnitude and direction of the waves.

When the water is sufficiently deep to permit of nearly circular orbits for the particles of water, in which case the motion near the bottom is very slight, the vane may be supported by and allowed to vibrate about, a single horizontal axis passing through or near its lower edge, as the movement and general direction of the distorted verticals will, in such case, be sufficiently well followed by a vane suspended in this manner. In Fig. 131, *aaa* is the wave profile, and *B* is a vane arranged as just described. By the passage of each wave, the vane is caused to vibrate in alternately opposite directions, following the movement and general direction of curvature of the distorted verticals. The power developed by these motions of the vane may be transmitted to the pumping or other mechanism for utilizing the same by means of connecting rods attached to the upper portion of the vane itself, or by crank arms attached to the horizontal shaft N, or by any other mechanically equivalent arrangement.

When, however, as is usually the case, the water is so shallow in proportion to the length of the wave that the orbits of the particles are quite elliptical, in which case there is considerable horizontal motion at the bottom, the vane is preferably suspended from above, at a point of its length near the mean height of the centre of pressure.

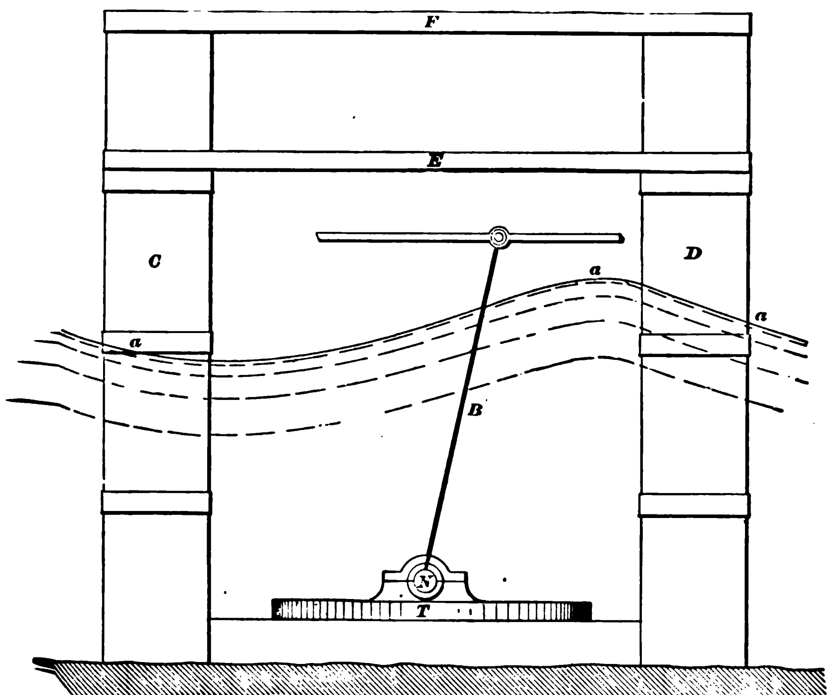

Fig. 131.

In Fig. 132, *aaa* is the wave profile, and B is a vane of the kind just referred to. The vane is suspended at the points O by crank arms P, at each end, these crank arms having an axis of rotation h at their upper ends, and the vane being free to revolve in bearings, O, at their lower ends. Thus, as the vane is caused to move in alternately opposite directions by the passage of a wave, that portion of the total power which is due to the translation or displacement of the vane as a whole, causes the crank arms P to vibrate from side to side, the amplitude and exact shape of the distorted verticals, it would slant as a whole toward the nearest wave crest, taking at any instant the average direction of their curvature along its length. While a flexible vane would theoretically be best as interfering least with the normal structure and movement of the wave, yet the increased complexity in mechanism required would more than counterbalance the gain in power, so that for practical purposes a rigid vane is to be preferred. Such vane should, in general, be made as light and thin as is consistent with proper strength and rigidity, and should extend from somewhat above the surface of the water to near the bottom. It is also to be noted that the above description of the movements of the wave applies strictly only to the more or less regular swell outside the breakers; and it is in these waves outside the breakers that, for greatest efficiency, such vane with its mechanism should be located.

In accordance with the principles just set forth it has been proposed jointly by the author and the late Mr. Richard Gatewood, U. S. N., to utilize the movements of the distorted verticals by opposing thereto movable vanes pivoted or supported either at their ends or other points of their lengths, so as to move in general accordance with the motion of the distorted verticals, and thus to receive and transmit the maximum effect of the wave movement.

The preferable method of suspending the vane, in any actual case, depends on the depth of the water and on the range of variation in the usual magnitude and direction of the waves.

When the water is sufficiently deep to permit of nearly circular orbits for the particles of water, in which case the motion near the bottom is very slight, the vane may be supported by and allowed to vibrate about, a single horizontal axis passing through or near its lower edge, as the movement and general direction of the distorted verticals will, in such case, be sufficiently well followed by a vane suspended in this manner. In Fig. 131, *aaa* is the wave profile, and *B* is a vane arranged as just described. By the passage of each wave, the vane is caused to vibrate in alternately opposite directions, following the movement and general direction of curvature of the distorted verticals. The power developed by these motions of the vane may be transmitted to the pumping or other mechanism for utilizing the same by means of connecting rods attached to the upper portion of the vane itself, or by crank arms attached to the horizontal shaft N, or by any other mechanically equivalent arrangement.

When, however, as is usually the case, the water is so shallow in proportion to the length of the wave that the orbits of the particles are quite elliptical, in which case there is considerable horizontal motion at the bottom, the vane is preferably suspended from above, at a point of its length near the mean height of the centre of pressure.

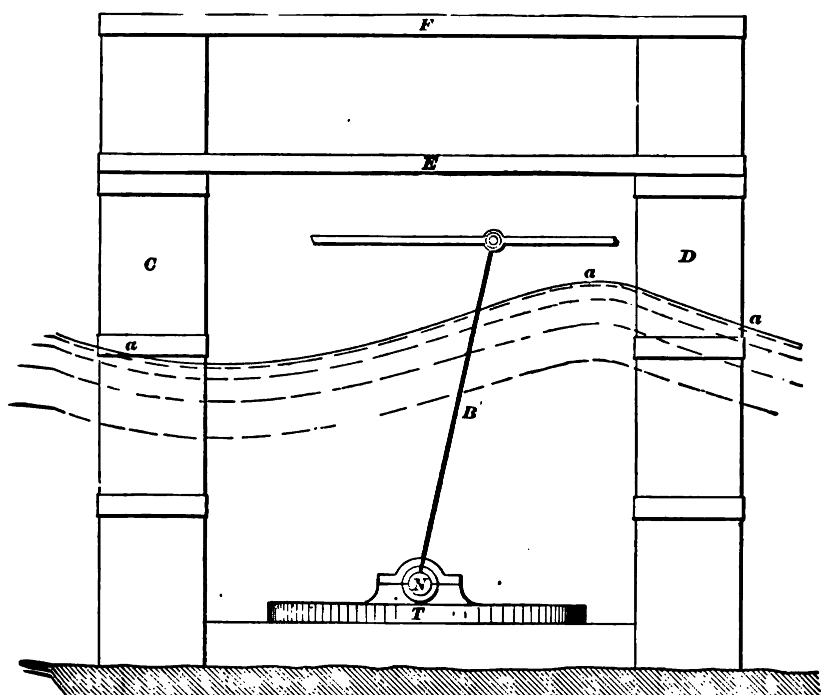

Fig. 131.

In Fig. 132, *aaa* is the wave profile, and B is a vane of the kind just referred to. The vane is suspended at the points O by crank arms P, at each end, these crank arms having an axis of rotation h at their upper ends, and the vane being free to revolve in bearings, O, at their lower ends. Thus, as the vane is caused to move in alternately opposite directions by the passage of a wave, that portion of the total power which is due to the translation or displacement of the vane as a whole, causes the crank arms P to vibrate from side to side, the amplitude and energy of such vibration depending on the size of the wave. But, in addition to this motion of translation as a whole, the vane also rotates about its axis *O*, in order to adjust itself to the general direction of the distorted verticals. The cranks *U*, *V*, are therefore rigidly secured to the vane at the point *O*. To the end of each of these cranks is jointed a link, *W* and *Y* respectively, of the same length as the crank arm *P*, these

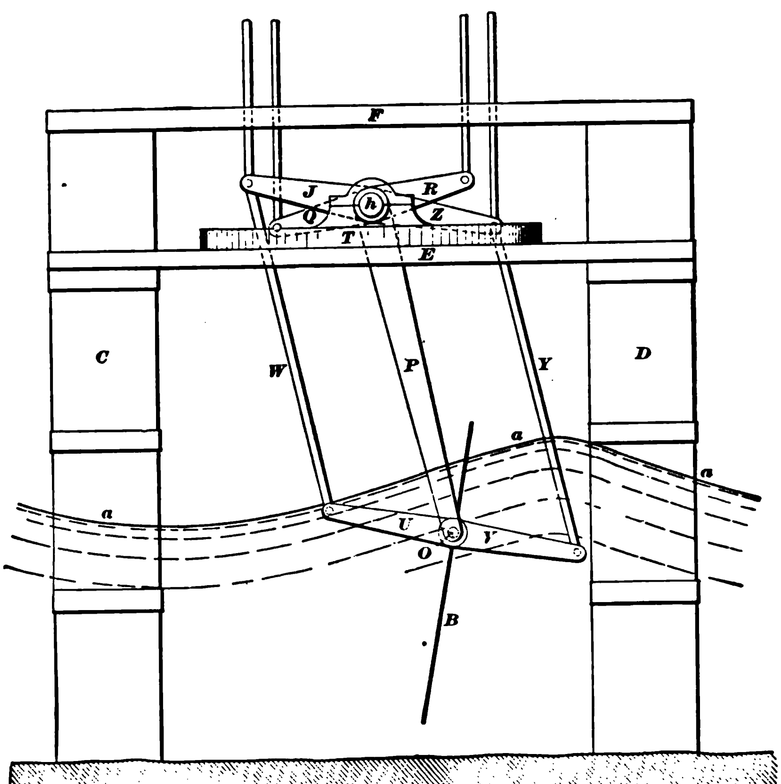

Fig. 132.

links being jointed at their upper ends to the cranks *J* and *Z* respectively. These cranks *J* and *Z* are the same length as the cranks *U* and *V*, and are free to turn about the axis *h*. Thus, as the vane is caused to move in alternately opposite directions by the passage of the wave, that portion of the total power which is due to the rotation of the vane about a horizontal axis to adjust itself to the general direction of the distorted verticals causes an equal rotation to be imparted to the cranks *J* and *Z*.

The *total* power due to the motion of the vane is thus employed to give motion to the crank arms P, and to the cranks J and Z. These motions may then be transmitted to suitable pumping or other mechanism for utilizing or storing this power by any of the ordinary suitable mechanical connections. Thus, in the figure the cranks Q and R, each provided with a connecting rod, are shown rigidly secured to the crank arms P, and connecting

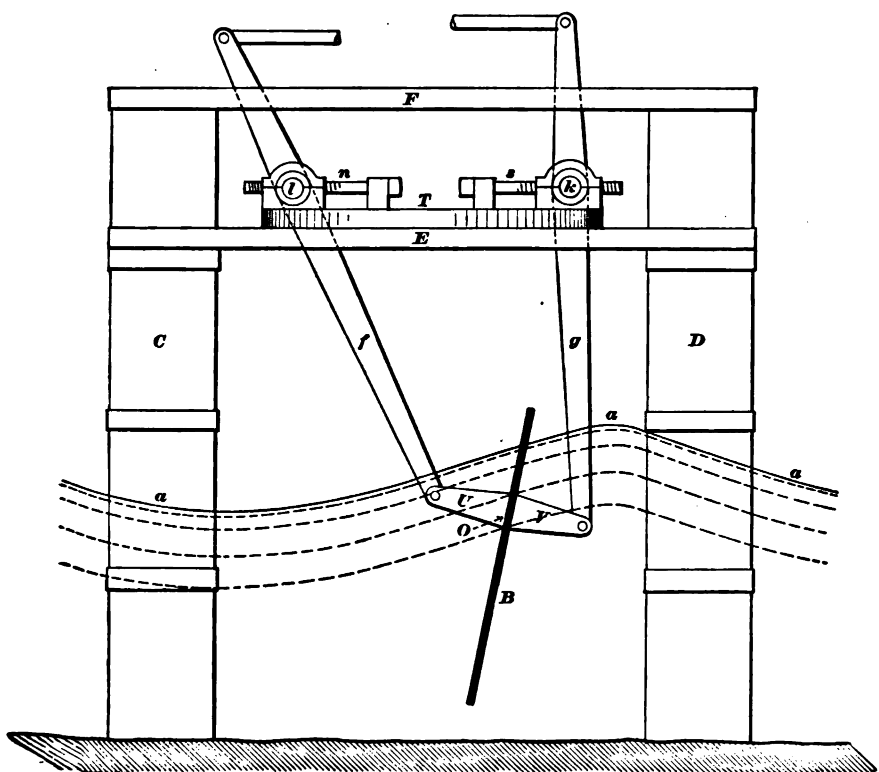

Fig. 133.

rods are jointed to the cranks U and V; but any other mechanical equivalents may be substituted, if preferred.

For any given depth of water the curvature of the distorted verticals, as well as their translation or displacement as a whole, depend on the dimensions of the wave; and for any given wave, the curvature and the translation of such distorted verticals bear a definite proportion to each other. These two motions can be computed or determined experimentally for any particular contour of bottom, depth of water, and size of wave; and the vane can, by suitable mechanism, be caused to rotate about a horizontal axis while undergoing the motion of translation imparted to it by the wave, this motion of rotation being regulated to bear, at all times, the proper proportion to the motion of translation.

In Fig. 133, *aaa* is the wave profile, and *B* is a vane of the kind just referred to. To the vane are rigidly secured the crank arms *U* and *V*, each of which is supported by and pivoted to another crank, *f* and *g* respectively, the latter being supported and pivoted at the points *l* and *k*. By means of this mechanism any translation or displacement of the vane as a whole is accompanied by a proportional rotation of the same about the axis *O*, the proper ratio between these two motions, for any particular series of waves, being secured by suitably adjusting the distance between the bearings *l* and *k*, by means of the screws *n* and *s*. The power due to both translation and rotation is taken off by the cranks *f* and *g*, and transmitted by connecting rods at their upper ends, or by any other equivalent method, to suitable pumping or other mechanism for utilizing or storing this power.

While other arrangements of mechanism may be devised to permit the vane to approximate as nearly as possible to the motion of the distorted verticals, the three arrangements just described seem the most simple and efficient of those that have suggested themselves to us; and of these three, the one shown in Fig. 133 is of most general application, and usually to be preferred.

In order to adjust the vane *B* about a vertical axis to suit the direction of the waves, there is provided in each of these arrangements a turn-table, *T*, which, by means of suitable mechanism, is turned and adjusted to any desired position and properly secured after adjustment. The turn-table supports and carries with it the bearings, the vane, and all other necessary parts. This turn-table may be dispensed with when the contour of the coast or other influences cause a practically constant direction of the waves.

C, *D*, etc., are firmly braced structures supporting the platforms or floors *E* and *F*, the turn-table *T*, and the vane *B*. On these platforms or floors, or within the structures *C*, *D*, etc., can be placed the pumping or other mechanism for transmitting, utilizing, or storing the power derived from the waves by the vane, and the mechanism for adjusting and securing the turn-table. In practically building these structures it would probably be found best to make their lower portions closed, so as to be filled with concrete or other heavy material, while their upper portions would be open lattice-work, so as to interfere as little as possible with the movement of the wave.

The energy of the forward vibration of the distorted verticals is greater than that of their backward vibration; and the vane will thus not vibrate equally in both directions if the same resistance is opposed to it during both vibrations. The proper and regular working of the vane can be ensured either by opposing a suitable smaller effective resistance to the motion of the vane during the backward than during the forward vibration, or by opposing to the forward vibration of the vane, in addition to the equal effective resistance opposing both vibrations, a suitable additional resistance so arranged as to yield up, during the backward vibration, the potential energy thereby stored up during the forward vibration. The former method simply requires that the amount of power derived from the vane on the forward and backward vibrations, respectively, is to be adjusted to correspond to the normal energy of the distorted verticals during these respective vibrations. The latter method is most simply carried out by the raising of a free weight or the extension or compression of a spring during the forward vibration, the potential energy of which is then allowed to assist the backward vibration. The irregularity of the vibrations which would be induced by a current can be rectified in a similar manner.

As the motion of these vanes, arranged as above proposed, conforms very closely to the natural movement of the distorted verticals, the elements of inefficiency which have been pointed out with reference to some of the previously described methods are very much reduced or even altogether absent. The action of the vane is more nearly akin to that of the piston of a steam-engine, being driven forward and backward by the variations of the fluid pressure on its faces. Its weight does not affect the amount of energy, except as a slightly prejudicial factor, which should be kept as low as possible. The vane is therefore made as light and thin as considerations of strength and rigidity will permit, thus reducing the loss of efficiency due to its inertia, and at the same time reducing to the least possible amount its interference with the normal structure and movement of the wave. A notable feature about these proposed vanes is that increase of immersion is accompanied by increase of efficiency; and this is a very important point of difference between them and certain other apparatus already described. The greatest efficiency for any particular series of waves will be obtained by letting the vane extend from the surface of the wave crest to the bottom of the water; and, provided the vane is long enough to reach to the top of the highest wave crest which it is desired to utilize, the efficiency of the apparatus will not be affected by variations in the magnitude of the waves.

To accommodate themselves to variation in height of tide, most of the arrangements previously discussed are raised and lowered bodily the required amount. Some of the floats are thus adjusted by simply shortening or lengthening the power ropes, which can readily be done; the floats that describe vertical arcs of circles adjust themselves by simply describing an arc which is higher up on their circle of motion; the floats describing horizontal arcs of circles and most of the vanes require their points of suspension or support to be adjustable in a vertical direction. The vanes proposed by the author require no adjusting, being made of such length that they are long enough to reach the wave crests at high tide, while at low tide their upper ends simply project above the water.

To prevent damage to the apparatus in a heavy storm, some of the motors are located in enclosures, the gates of which may be closed in bad weather, and others are arranged to hoist out of the water altogether. But as the energy of the waves is very much greater during a storm than with an ordinary swell (see Table of Horse-power, above given), it hardly seems wise to throw the motors out of action at a time when they could develop the most power. We propose, therefore, to make our vanes of such length that they will reach to the top of the largest waves that can be faced with the strength of the vanes as constructed. When the size of the waves increases beyond this limit, and huge breakers come rolling in, the crests of the latter will be considerably above the upper ends of the vanes, so that the vanes will not be injured by the breaking of these crests; and as the motion of the water rapidly decreases below the surface, we consider that our proposed vanes may safely work in any weather, being entirely below the reach of the breaking crests of the waves in a heavy storm, while they project up through the wave crest in ordinary weather.

The action of successive waves is usually more or less irregular, and the action of even a perfectly regular series of waves imparts to any wave-motor a variable velocity ranging from an instant of absolute rest at each end of its stroke to a maximum velocity about midway between these points of rest. As the practical utilization of any available power usually demands that such power shall be furnished at a fairly uniform rate, it becomes essential to employ some means to reduce this inherent irregularity to the smallest possible extent so far as the transmission of the power is concerned. Most of the apparatus referred to in the foregoing pages is provided with some sort of a ratchet arrangement, so arranged that a main shaft is caused to turn continuously in the same direction by each of the alternately opposite motions of the motor proper. To ensure a fair degree of uniformity of motion in the shaft, it is likewise provided with a heavy fly-wheel, and this object is still further aimed at by combining on the same shaft the motions of a number of such motors located at a little distance from each other, in the direction of advance of the waves, or the motions of different parts of the same large float or other motor. Another and better arrangement consists in the use of a fluid accumulator, either air or water under pressure being used, the motor keeping up the pressure in the accumulator by means of pumps or air-compressors, and the main shaft being driven by an air or water engine connected with the accumulator.

But in addition to this regulation of the amount of power during consecutive revolutions of the shaft, there is required another and more important regulation. In order that the power to be derived from the waves shall have any high marketable value for most purposes, we must be able to guarantee a certain uniform supply from day to day without regard to wind or weather. Now, as the waves vary from day to day, throughout all the changes from a perfect calm to a heavy storm, and as, furthermore, the power will usually only be required in the daytime, while the wave-motor may work day and night, any scheme for utilizing this power on a large scale for industrial purposes must involve a system of power storage.

There are four principal methods of storage which may be considered in this connection, viz.: (1) compressed air; (2) water under pressure; (3) electric storage by means of secondary batteries; (4) water in an elevated reservoir. In the storage of power by means of compressed air the volume of air is constant, so that the pressure must vary very greatly from day to day. In addition to the great expense of storage tanks, there exists thus the strong objection that the air-compressors for pumping into the tanks and the air engines which are driven by the compressed air are required to work under very great variations of pressure. With the hydraulic accumulators the latter objection need not exist, but the large quantity of liquid required renders the tanks themselves still more expensive. The best and most perfect system of storage for many purposes is to be found in the use of secondary electric batteries, and they have the additional advantage that the power is by them furnished in a convenient form for transmission to the point where it is to be finally used. But their cost is as yet, unfortunately, so great that their employment seems entirely out of the question for our purposes. The storage of water in an elevated reservoir offers many advantages, and seems on the whole to be the best adapted for use in connection with a wave-motor. When this plan is adopted it becomes unnecessary to regulate the power at the motor, as the latter may simply drive a system of pumps for delivering the water into the reservoir, the pumps always working at a practically constant pressure, and the number of such pumps in action at any particular time being regulated by the magnitude of the waves, and the consequent amount of energy in the same. The water from the reservoir may be used to drive a water-engine or water-wheel, and the power of the latter may then be further transmitted as seems best for each particular case.

The use of a water-storage reservoir would have an incidental advantage in this city (San Francisco), where the question of periodically flushing the sewers has become one of recognized importance. As this latter work does not require to be performed with absolute regularity, it could be done very cheaply by using the excess of water from such storage reservoir, at such times as might prove most convenient.

The size of the storage reservoir depends on the amount of power to be continuously supplied, and on the extremes of variation in the power of the waves from day to day. In default of positive knowledge on the latter point, we may assume such an amount of variation as seems likely to occur; any error in our assumption, as revealed by practical experience, will of course modify the amount of power that can be depended upon con tinuously.

As an illustration, let us assume that the reservoir is so located that the average level of the water in the same will be 340 feet above the water-wheel or other engine that it is intended to employ. This corresponds to a theoretical head of water of 150 lbs. per square inch = 21,600 lbs. per square foot. Practically we can only count on about 90% of this head, or 19,440 lbs. per square foot. Assuming a perfect efficiency for the water-wheel, the reservoir must discharge $\frac{33000}{19440} = 1.697$ cubic feet per minute per horse-power. Supposing a water-wheel to be used which can be depended on for an efficiency of 80%, the quantity of water to be discharged is thus $\frac{1.6975}{.80} = 2.122$ cubic feet per minute per horse-power. Assuming further, that the power is required for 10 hours each day, the discharge must be $2.122 \times 60 \times 10 = 1273.2$ cubic feet per horse-power per day of 10 hours. The next most important assumption is the variation in the supply of water pumped into the reservoir by the motor. To be on what seems the safe side, let us assume that the reservoir must contain sufficient water to supply the power at the assumed rate for a period of 10 days, without in that time receiving any water from the pumps of the motor. This would require a storage capacity of 12,732 cubic feet per horse-power. If we were supplying 100 H. P. this would require 1,273,200 cubic feet, which could be stored in a reservoir 20 feet high and 252 feet square, or 30 feet high and 205 feet square; similarly for 1,000 H. P. there would be required a reservoir 30 feethigh and 651 feet square, or 40 feet high and 564 feet square.

In any actual installation, the power of the water-wheels would be transmitted to the points at which it was to be finally used by such means as might be most economical and convenient, taking into account both the amount of power and the distance to which it is to be transmitted. In the case of this city, it would be found best to convert the power at once into electricity by means of dynamos driven by the water-wheels, transmitting the same by overhead wires to a central station in the city, whence it could be distributed to individual consumers in the usual manner.

In conclusion, it may be interesting to note that of the 32 patents which have been granted in this country for the improvements in wave-motors, just one-half have been granted to residents of California. This is probably largely due to the high price of coal in this State, which naturally causes men to look favorably on any scheme that seems likely to reduce the cost of power; while another and important reason is to be found in the fact that the ocean waves as they reach this shore are usually much more regular in size from day to day than on the Atlantic coast, thus helping to reduce the commercial difficulties of the problem. The circumstances are thus, on the whole, considerably more favorable to the success of a wave-motor on the Pacific than on the Atlantic coast, as a less degree of efficiency in such wave-motor would be required here than there to enable it to successfully compete against coal and the steam-engine.

DISCUSSION.

*Mr. E. D. Stodder.**—With your kind permission, I would like to say a few words in regard to this interesting subject on which Mr. Stahl has just addressed you. It is one which I have studied for some years and as some of my observations lead me to entertain at least a different opinion in some respects, I will endeavor, as briefly as possible, to give a few of many reasons I have for my conclusions.

I find that the principal difference between Mr. Stahl's ideas and mine are, that, generally speaking, he designs to obtain the power of the waves in a horizontal direction, acting on a vane suspended in a vertical position; while I design to obtain the power of the wave in every direction, acting on a float and moving it in what may be called a natural wave motion, or such as the waves give to ships and all floating objects with which they come in contact.

As each style of apparatus is designed to work in deep water, or beyond the breakers, the question arises: How would the waves act on each? A great deal of evidence can be produced as argument to show the advantages and disadvantages of each system. The principal objection I have to the

* Of San Francisco; by invitation.

horizontal motion is that in ordinary weather I don't think the waves have much power in that direction, for the following reasons, to wit:

First. Waves can be seen washing over some rocks, and rising and falling alongside of some others that are in deep water, without breaking against them; while the same waves can be seen dashing with great force against other rocks in shallow water nearer the shore. This can be seen any day in the year out near the "Cliff House" and in many other places; but the conditions under which it occurs vary with the size of the waves and the stage of the tide.

Second. I have seen small boats and large vessels rise and fall and rock and roll near piers in the ocean, without striking against the piers or drawing the lines tight that were intended to hold them. I have seen this occur when there were waves, and when there was only a swell; and heard people say at the time, "There is not a wave to be seen."

Third. I have noticed, when swimming in the ocean, that in starting out from the beach it required considerable exertion to get out beyond the breakers, as they have a tendency to drive one toward the shore; but once beyond the breakers and in deep water, it is as easy to swim in one direction as another.

Fourth. In the *Transactions of the Society of Engineers* (vol. x. p. 149), among other interesting information is the following: "It has been observed at Portland that, when the waves meet the circular heads of the passage between the two breakwaters (which consists of vertical walls twenty-four feet below the level of low-water mark), they rise and fall in a perfectly gentle manner, while some two hundred yards off, where the long slope occurs, the sea runs up and down with great violence and fury."

I do not say that where the water is deep there is no wave power in a horizontal direction, although these examples prove that there could not have been much when the observations were made, because numerous instances can be cited to the contrary and are well known from the destruction it has done in some cases; but I think they will be found the exception and not a general rule applicable to every wave; but I do believe the most available power can be obtained from a float arranged according to the principles of my invention.

As to what action wave motion has on a float, every one has noticed the ease with which the waves move a large vessel, but I don't think there is one of you that can correctly describe every motion given to a vessel by the waves; that if it were possible for every one of you to have a different idea of the way a vessel would be moved by the waves, every one of you could move, according to your own idea, a float that is part of my model; and every one of you would keep a shaft revolving in one direction, as long as you moved it; and there is not one of you but knows that the power of a revolving shaft can be converted into light, heat, and other forms of power, and that it is very valuable if you can get enough of it.

This subject is a very broad one and cannot be discussed properly in a few minutes. I have a great many reasons why I think wave power can be put to practical and profitable use, but I don't wish to occupy too much of your valuable time, as, perhaps, there are not many of you particularly interested in the subject; but to any that are, I earnestly request that you will allow me to show you my model and offer some other ideas for your consideration, for although I am not an engineer, and do not consider my invention perfect in all details, I think that my ideas on the subject are such that they could be taken up by some of you, worked out in detail, and carried out to a successful issue.

I am aware of the fact that it is the generally accepted idea of some of the most eminent engineers and scientists "that enough power cannot be obtained from wave motion to pay for the construction of the very costly apparatus necessary to obtain it," and that this conclusion has been reached by no less a person than Sir Wm. Thompson. Still, the very fact that so great a man as Sir Wm. Thompson has considered the subject, is conclusive evidence that there is much to be gained if wave power can be profitably utilized; because it is not likely that he would waste his time considering anything from which there were no prospects of obtaining valuable results; and I have no doubt that even he would be pleased to find out that he had made a mistake, if the proof of it were a valuable addition to our present resources.

Prof. D. S. Jacobus.—The practical portion of this paper by Mr. Stahl is instructing and interesting. I think, however, that it would greatly improve the paper to put in an explanation showing how the theoretical part of it could, perhaps, be made to have some reasonable and definite connection with the action of the models given at the end of the paper. We have the generally accepted theory of the motion of waves given in the beginning of the article, to which is added a calculation of the energy contained in a wave, but no attempt is made to arrive at the per cent. of this energy which any one of the machines described might be able to utilize. No machine can absorb all the power of a wave; and the important question is, how much of the energy could be utilized by such machines? There is neither an experimental figure here to show this, nor any approximation to it by theory. I think this would be the best part of such an article, because the greater portion of the theory is, as Mr. Stahl states, the ordinary theory of wave motion, which is available in other places.

All the equations and conclusions, up to equation 23 inclusive, are given by Rankine in the *Philosophical Transactions of the Royal Society* of London for the year 1863, vol. cliii., part i., where the interested reader will find references to other and more extended treatment.

Mr. Geo. W. Dickie.—The paper read by Mr. Stahl is exceedingly interesting, outside of the wave-motor portion of it. He introduces and illustrates a good many points which are of great service in other directions besides that of wave-motors. I think his table on page 455 is a very instructive table indeed. I don't recollect ever seeing such a table compiled before. But seeing that we have all this power in the waves, and we have the waves —and our members not only will see them, but will see besides a good many wrecks of wave-motors along the beach—seeing we have all this power, to use an expression which is well-known in the East, what are you going to do about it? Granting that the power can be utilized by such an instrument as Mr. Stahl describes, to what economic use can such an instrument be placed? Mr. Stahl has not gone as far as that point. His paper evidently is the beginning of the end; I trust it won't be the end of the beginning of this subject. If we take the unit of 1,000 H.P., it may be computed that the capitalization for that amount of power at the present price of coal in San Francisco would be about $400,000; that is, any apparatus, in all its connections and in all its applications, which would cost more than $400,000 would not be admissible for a production of 1,000 H.P., and it would be exceedingly interesting if the application of this power could have been figured out in such a way as that the capital which would be required to be invested for the production of 1,000 H.P. could have been ascertained, because thereon hinges the usefulness of such an application of power. Mr. Stahl indicated the storage of power by a reservoir. In the application of this power near by the bar of San Francisco harbor, where the waves could be most depended upon, an elevation of not more than 200 feet could be counted on.

Mr. Stahl.—300.

Mr. Dickie.—Mr. Stahl says 300. I have not measured that. I am just thinking about this thing as I talk. I am assuming that 200 feet elevation could be obtained, and that a storage would be required to maintain a constant power during periods of calm of 15,000,000 cubic feet, in order to produce that power without any surplus for lack of efficiency, to which 25% at least would have to be added for any motor that could be applied. Now, whether all that and the apparatus itself, works to protect it, pipes to convey the water, and motors to distribute the power of 1,000 horses could be established for $400,000 would be the question which, I think, would settle the economic possibilities of a wave-motor power. Perhaps there is no place in the world where a wave-motor power could be so successfully utilized as in the vicinity of this city. We are near the ocean; we have the waves, more at times than we would like to have; the price of fuel is high, and if wave-motors are ever to be used economically and compete with other sources of power, this would be the place for their installation. The subject, I think, is exceedingly interesting, but the whole thing, to my mind, hinges on what would be the capital required to be invested for a given power.

Mr. E. T. Molera.—I happen to have gone through what I believe nearly all engineers have at one time or another, who have either begun an attempt to utilize wave power, or at least have given it some consideration. Of course we all see in the water an immense power extended all over the ocean, and the manifestations of which are very grand. I used to be in the Light-house Department, being assistant light-house engineer from the year 1870 to 1875, and all the reasons for trying to apply wave power for some useful purpose are there very much emphasized. We had to transport coal, which is exceedingly high on this coast, to points nearly inaccessible, at double the price of the coal. Then we had to store water in cisterns and make a water-shed, as we used to call it, and therefore this became a very expensive way of supplying the power for the steam-whistles along the coast. Under the circumstances I began to think of applying wave power, and I considered floats and much apparatus similar to those which Mr. Stahl has described, and others which Mr. Stahl has not described. For instance, I thought of employing a water-tight vessel with appropriate valves, so that when the water rises the volume enclosed diminishes, the air is compressed, and when it gets to a certain density it is sent to a proper reservoir; and *vice versa*, when the water falls it should rarefy the air, allowing more to enter. I have devised also an apparatus like a hydraulic ram, where the water should pass through a trumpet-like pipe, entering at the large end and passing out at the other, and when it had acquired a certain velocity, a valve closed suddenly, as in a hydraulic ram. I think there will not be so much trouble in devising an apparatus which will convert the wave motion into some other motion as to find a way in which such motion can be applied. I think that the apparatus which Mr. Stahl has described is exceedingly ingenious; in fact, I know of no other as ingenious as that; but, as Mr. Dickie says, when you have got it, what about it? The wave motion is exceedingly unmanageable, and I have found it exceedingly unprofitable. You can depend upon it only at certain times. Mr. Stahl says that ten days will be required sometimes to have power stored and in that time you cannot depend on any of the wave power. I don't know how Mr. Stahl reached that figure; perhaps he will tell us. I know that I made an investigation from the records taken by Prof. Davidson at North Point, and Major Mendell, now Colonel Mendell at Fort Point, and while the apparatus was designed only for the prediction of the tide, yet it gave also the wave motion; and from the record of the waves the curve of the tide is traced. It gives so much the wave motion, that Prof. Davidson was able at one time to predict the occurrence of an earthquake which happened in Japan twenty days before we knew anything of it here from other sources. The disturbance in the tide here was of such a nature that he could announce it. Therefore this was a very good record of the wave motion; and I have had the advantage of going through the record for several years, and from that record I find that not only sometimes you will have to depend on not having the wave assistance for ten days, but even for twenty days. Of course whatever the minimum is of the wave power, you have to provide for that, because it would not do at all for you to supply power for three or four months regularly and then on a certain day have no power at all. But taking the ten days' limit, the minimum limit, and taking Mr. Stahl's figures, any engineer who has the time to find out what it would cost to build a structure of brick, or masonry, to act as a reservoir 30 feet high and 200 feet long, something about that, you will find that the reservoir alone will cost from $15,000 to $20,000. Then the apparatus. For instance, there is no city that I know of close to the ocean which is so well adapted as San Francisco for storing the water in high places; but even in San Francisco, favorable as it is, we have just exactly two points near the sea-shore where we can get an elevation of 300 feet. One is just one mile northeast of the Cliff House, and the other is just exactly south of Fort Point, and both are about a mile distant from the city. Therefore you have to provide from the motor a pipe one mile long to the reservoir. When you have the reservoir there you have to convert the power of the water into another power down at the bottom, 300 feet below, according to the slope of the hill where the reservoir is, and this will be the length of the pipe. Then you have again to reconvert this power into another power, to do which, I agree with Mr. Stahl, electricity will be the most available. Therefore you would have transformation of the weight of water into hydraulic power, then into an electrical current, then again conveyed through a conductor six or seven miles long to the centre of the city, and here again conveyed into small electric motors for the purpose of utilizing the power, and so on. Of course, having gone through this problem also in detail, I found that it was impracticable. At first I was very sanguine, perhaps more sanguine than Mr. Stahl. I sent a preliminary report to the Light-house Board and got a very flattering letter, which of course I keep as a relic, from Prof. Henry, who was a man of great eminence, and he encouraged me to go on; but when I went into the details of the cost of the thing I recommended that every fog-signal station on this coast should be furnished with a proper boiler and supply of coal, and obtain the power in that manner. If Mr. Stahl can do better than that of course we would all be very glad.

*Mr. E. T. Steen.**—I would like to inquire of Mr. Stahl how he proposes to protect his vane when the waves come in with great force ; how he proposes to protect it from being destroyed by the force of the waves, if the motor swings from the bottom?

Mr. Stahl.—I do not want to protect it, except by limiting its height, as explained in the paper.

Mr. Steen.—Do you propose to allow your vane to go entirely down, horizontally, in a heavy sea, and then come back again—or how do you prevent it being broken off? There must be great danger in very heavy seas, of destroying the vane. If it still maintains an incline in that way, it certainly will have a tremendous force against it. Of course you are aware that the force of the waves has been determined to be sometimes over three tons to the square foot. That has been tested. But that is very much above the average. Now, you have to build the machine strong enough to stand the greatest force that may come against it or you will not have a machine long; and if your machine cannot take care of itself, I don't know how it is going to be taken care of. I might state that I have made something of a business of observing the waves for several years, and have experimented a little in that line, and I find that there is no twenty-four hours in which you cannot pump more or less. My experiments were somewhat of an expense ; they cost me about $14,000, mostly on account of an explosion of a ship-load of dynamite in the neighborhood. I used the vane that Mr. Stahl observes is not the best, but it seems to me, from my experience, the most practical. To begin with, it is cheap and takes care of itself; it swings up out of the way of heavy waves, after doing its work each stroke; you don't have any connection with the bottom. There is no chance for the sand to get in and cut the journals, because they are above the water; and if we don't get as much force, we get sufficient, I think, for all practical purposes.

As for the height of the reservoir, there are convenient places on this peninsula where you can get 580 feet elevation. There is about 27 acres in one place and about 40 acres in another which, with a sufficient number of motors, as many as I would advise to do the work, would give 60,000 to 70,000 H.P. from those reservoirs. About one cubic foot a minute on impulse wheels would give a horse-power.

* Of San Francisco; by invitation.

In what depth of water would Mr. Stahl propose to build his motor?

Mr. Stahl.—In 30 to 40 feet; in the least depth which would ensure my apparatus being outside the breakers.

Mr. Steen.—Forty feet deep would be outside of the breakers, where there is very little if any force on and off shore to operate a motor, except when the wind comes directly on shore, and then there is nothing on the surface to carry the motor back for the next stroke. The cost in such deep water would be so great as to make it impracticable; inside of the breakers there is great force, and the cost of building small, as I find—about seven feet at low water is the best depth. A motor hung at the top should not be immersed more than one-sixth of its length, say 6 to 7 feet at high water.

*Mr. A. W. Stahl.**—In attempting the practical solution of the problem of utilizing the power of ocean waves, we must first ascertain which of all the apparatus and methods proposed is most logical and likely to be efficient. This having been established, the actual efficiency of the particular apparatus must be determined; the main element of such efficiency, and the one concerning which we are most ignorant, being the percentage of the total energy of the waves which will be absorbed by the motor, and transformed by the latter into useful work. Finally, and most important, comes the determination of the cost of the apparatus and all its accessories, together with the capitalization required per horse-power delivered to the consumer; and this last item is of course the one on which the commercial success of the apparatus depends. In this paper I have attempted to deal with the first of these questions only. I have not entered into a discussion of the question whether *any* wave-motor will pay commercially; but have confined myself to demonstrating which motor seems most efficient and seems likely to give the best results.

Possibly none of the methods described in this paper may ever prove commercially successful; indeed the problem may not be susceptible of a financially successful solution. My own investigations, however, so far as I have yet been able to carry them, incline me to the belief that wave power can and will be utilized on a paying basis; yet in any case it is evident that the

* Author's closure.

first and principal thing to establish is which of all the apparatus proposed holds out the greatest promise of success.

I have therefore discussed at length the theory of wave motion, so far as it applies to the subject on hand ; for it is manifestly one of the main conditions of efficiency in any wave-motor that its motion shall accord, as nearly as possible, with the natural motion of the particles of water, so as to interfere with the normal structure of the wave to the least extent.

The main features of this now generally accepted theory of wave motion were first advanced by Prof. Rankine, whose original paper on this subject may be found in the *Philosophical Transactions of the Royal Society* for 1863. The extension of his results to the formulæ of horse-power of waves, and the computation of the horse-power table given in the paper, are original with myself.

I regret my inability to answer Prof. Jacobus's question as to the actual efficiency of the motor proper; that is, as to how much of the energy of the waves will be utilized by the proposed apparatus. The theoretical solution of this question would be very difficult, and of little practical value when obtained. Direct experiment seems the only way to determine this very important point; and I hope that before very long I may be able to give some such experimental data.

Referring to Mr. Stodder's remarks, my criticism of his apparatus (shown in Fig. 121), and of all others of the same general type, has been given in the body of the paper, and need not be repeated here. His main objection to my proposed apparatus rests on his opinion that the amount of horizontal motion in deep water is comparatively slight. As the particles of water move in circular orbits in deep water, their horizontal motion at the surface must of course be equal to the height of the wave, gradually decreasing toward the bottom, as already explained.

There can be no question but that the total energy of the wave in deep water is greater than the energy of the same wave after it has come into shallow water and become a breaker, as the foam and froth in the latter is a visible manifestation of the dissipation of energy there taking place There is this further difference, that in the breakers the transmission of the energy of the wave to a wave-motor must be more or less in the nature of a shock or blow; while in deeper water the energy would be transmitted by a smooth and continuous variation of pressure, a condition much more favorable to efficiency than that existing in the breakers.

Now, about this question of cost or capitalization, the $400,000 that Mr. Dickie speaks of. There can be no question but that is the keynote of the whole business as a commercial enterprise. But I had not intended in this paper to take up the commercial question, which, though vitally important, can only be logically solved after we have satisfactorily answered the question discussed in my paper, viz., to which of the various wave-motors proposed must we look for success, if any at all is to succeed? I recognize fully the force of Mr. Dickie's remarks, and I have made careful estimates of cost as far as my plans have taken definite shape, but these estimates are as yet too incomplete to enable me to do more than state that, in my opinion, the capitalization required will be far within the limits set by Mr. Dickie.

I am both surprised and pleased at Mr. Molera's statements as to the cost of the reservoir. I have been figuring on the cost of the reservoir as a much larger item, and am gratified to know it can be built for the amount that Mr. Molera states.

In reply to Mr. Steen's question about protecting the wave-motor from the waves during a storm, I desire to repeat that I propose to limit the height of the vane above smooth water, and thus keep its upper edge below dangerous breakers. The fixing of this height of top of vane above smooth water is a purely commercial one, depending on the cost of work, and on the frequency of waves of various heights. The higher the wave with which the vane is to cope, the greater the energy of the wave and the greater the power which can be utilized; but on the other hand, the stronger must the vane be, and consequently the more will it cost. From the frequency of the waves of various heights and the corresponding cost of the vane for each of such waves, we must determine how high it will pay to construct the vane. Now, suppose we find that it would be profitable to construct a vane strong enough to take care of a wave 10 feet high. We make this vane so that its top will project 5 feet above the smooth water. As waves begin to come in, the vane is caused to move, its upper end projecting up through the crest of the wave so long as the latter is less than 10 feet high. The energy imparted to the vane gradually increases until the wave is 10 feet high; but any increase in height of wave above 10 feet will have no effect on the vane, which will go on just as if actuated by a 10-foot wave. The vane will thus never be called on to handle more than the energy due to a wave 10 feet high, and can thus work safely in any storm. The vane being built strong enough to resist the pressures due to a wave 10 feet high, will not be called on to resist any additional pressure during a storm, as the upper end of the vane is entirely below the reach of the crests of larger waves.

Mr. Steen's objection that the journals in the apparatus of Fig. 131 will be injured by the action of the sand, would be perfectly valid if the journals were unprotected and made in the ordinary way. This could, however, be obviated by making the connection at the bottom consist of a few links of chain, instead of an ordinary journal. But I do not consider this particular apparatus as practically very valuable. I merely introduced it into this paper as a step in the logical development of the other two motors, Figs. 132 and 133, the latter of which I regard as the practical apparatus which is likely to be thoroughly successful. It seems to me that the motor shown in Fig. 133 is a little better and nearer a scientific and logical apparatus than any other with which I am acquainted; and the statements of several gentlemen at this meeting have tended to confirm me in this belief.

In the practical construction of this apparatus, I should probably make the supporting structures of a conical shape, the lower portion being hollow and filled with concrete, and the upper portion being open lattice-work, so as to interfere as little as possible with the motion of the wave. I should build the whole apparatus ashore, and with the aid of lighters float it to its proper location. I would then sink it to rest on the bottom by pumping the concrete into the hollow portion of the supporting structures, after which I would further secure it by anchors imbedded in the bottom at some distance from the apparatus. The structures being 50 to 60 feet apart, and the centre of gravity being kept very low, there would be a very large stability developed, sufficient to prevent any damage in a storm.

Mr. Steen.—It seems to me that it is fundamental that you should have something that would protect itself.

Mr. Stahl.—The energy of a wave, per foot of breadth, varies as the square of the height and as the length. Now, suppose that I have decided, after computing, that it would pay me to put up a motor to handle a wave 10 feet high, and that a wave that is higher than that occurs so seldom that it would not pay to construct for that strain. You can build the motor strong enough for any size wave, but practically, commercially, you have to stop at a certain point. Now, I make the top of my vane so that it is 5 feet above water, on the basis of a 10-foot wave. The wave crest, when the wave is 10 feet high, will just touch the top of that vane. Suppose the storm wave is 20 feet high, the upper five feet of crest will pass over the top of the vane, without exerting any pressure on the latter; the rest of the water all the way down is doing its work just as it was when the wave was 10 feet high. The motor is thus never called upon to resist more than the pressure due to a 10-foot wave, and having been constructed for that pressure, needs no further protection. If it is considered desirable to build for a greater wave, it can of course be done.

Mr. Steen.—May I ask what is the maximum force of a wave here on this coast at the depth you propose to work it?

Mr. Stahl.—If you will consult the table given in the paper you will get the horse-power per foot of breadth of all the waves likely to occur. The horse-power of any wave whatever can be readily computed by means of formula (28).

CCCCXCII.*

SUMMARY OF RESULTS OF PRINCIPAL EXPERIMENTAL MEASUREMENTS OF PERFORMANCE OF REFRIGERATING MACHINES.

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(Members of the Society.)

In order to accurately measure the cold-producing effect of refrigerating machines, it is necessary that the refrigerating fluid, when at its lowest temperature, be circulated in a closed coil submerged in an insulated bath of brine, which, by circulation in contact with some source of heat, is made to pass through a fixed range of temperature between the outlet and inlet of the brine reservoir. The quantity of brine circulated per unit of time, its range of temperature, and its specific heat, being then determinable by measurement, the refrigerating effect per unit of coal consumed to operate the apparatus may be stated as equivalent to the cold required to freeze water at 32° Fahr. into ice at the same temperature. On this basis it is common to state that the performance of a refrigerating machine is equivalent to so many pounds of ice per pound of coal as a measure of fuel economy, and to so many tons of ice per 24 hours as a measure of its capacity. Any such statement of performance does not therefore represent that the refrigerating machine would make these amounts of actual ice, because: first, in making ice the water frozen is generally of about 70° temperature when submitted to the refrigerating effect of a machine; second, the ice is chilled from 12 to 20° below its freezing point; third, there is a miscellaneous dissipation of cold, not exactly definable from the exposure of the brine tank and the manipulation of the ice cans—therefore, the weight of actual ice made with ammonia machines represents only about three-fourths of the cold produced in the brine by the refrigerating fluid† per indicated

* Presented at the San Francisco meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

† It is shown in the appendix that the cold produced in the brine may be only about three-fourths of that due to the latent heat of the weight of ammonia corre-

horse-power of the steam-engine driving the compressing pumps. Again, there is considerable fuel consumed in ice making, to operate the brine circulating pump, the condensing water and feed pumps, and to reboil, or purify, the condensed steam from which the ice is frozen. This fuel, together with that wasted in leakage and drip water, amounts to about one-half that required to drive the main steam-engine. Hence, the pounds of actual ice manufactured from distilled water is just about half the equivalent of the refrigerating effect produced in the brine per indicated horse-power of the steam-cylinders. When ice is made directly from natural water by means of the "Plate System," some of the fuel, used with distilled water, is saved by avoiding the reboiling operation.

It is therefore agreed that the term "ice-melting effect" means the cold produced in an insulated bath of brine, on the assumption that each 142.2 B.T.U. represents one pound of ice, this being the latent heat of fusion of ice, or the heat required to melt a pound of ice at 32° to water at the same temperature.

Of the refrigerating fluids which have been, or which still are, used in practice, there are no records regarding air and ether which enable us to state the "ice-melting effect." Ether machines, used in India, are said to have produced about 6 lbs. of actual ice per pound of fuel consumed in driving the steam-engine used as a motor, which would roughly prove that they realize as great a proportion of their theoretical efficiency as will be shown below to be obtainable from ammonia and sulphur-dioxide. The ether machine is, however, obsolete, because the density of the vapor of ether at the necessary working pressures requires that the compressing cylinder shall be about 6 times more cumbersome than for sulphur-dioxide, and 17 times larger than for ammonia. Air machines require about 1.2 times greater capacity of compressing cylinder, and are, as a whole, more cumbersome than ether machines, but they remain in use on shipboard, because the use of air incurs no such risk of destruction of provisions preserved by cold, as exists by danger from leakage when such chemicals as ammonia or sulphur-dioxide are used. There are also no records of the measurement

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sponding to the displacement of the compressors and the suction pressure density of the saturated vapor, as the superheating of the ammonia gas by the walls of the cylinder makes the weight of ammonia circulated about 25% less than this theoretical amount, assuming no loss by clearance, leakage or valve slip.

of the performance of air machines by the use of a bath of brine, as described above, which enable the ice-melting effect to be stated, the air, at its lowest temperature, being either discharged free into the chambers to be cooled, as in the Bell-Coleman system for example, or circulated through closed pipes lodged within such chambers, as in the Allen Dense Air machine. All measurements of performance of air machines have therefore been based upon the measurement of the range of temperature of the air between its entrance to the compressing cylinder and to the chamber to be cooled, combined with the weight of air circulated, the latter being computed from the displacement of the compression, or the expansion cylinder, by the aid of indicator cards.

TESTS OF AIR MACHINES.

A fair sample of the above method of determination of performance of air machines is published by Prof. Schröter of Munich, for a Bell-Coleman machine, compressing air to about four atmospheres, and the result shows a refrigerating effect equivalent to 3.4 lbs. of "ice-melting effect" per pound of coal required to drive the steam-engine, assuming 3 lbs. of coal per hour per horse-power as the economy of the engine. This result is 43.1% of the theoretical result, computed upon thermodynamic principles, including an allowance for friction of mechanism of about 22% of the power of the steam-cylinder, which was the actual friction as deduced from the indicator cards given by Prof. Schröter, samples of which are shown in Fig. 135. A similar measurement of performance of a closed cycle air machine, compressing the air from 39 lbs. to 160 lbs. above the atmosphere, gives 3.0 lbs. of ice-melting effect, or 37% of the theoretical results. Such a machine, by handling the air at higher density than that corresponding to atmospheric pressure, has a proportionately smaller compressing cylinder, and is therefore more compact. The details of both of these tests are given in lines 22 and 23 of Table I. The principal cause of the large difference between the theoretical and actual efficiency of air machines is the fact that the temperature of the air leaving the expansion cylinder is very much higher than theory indicates for adiabatic expansion, even taking into account all possible influence of moisture. Thus, in the Bell-Coleman machine the theoretical temperature, due to the pressure to which

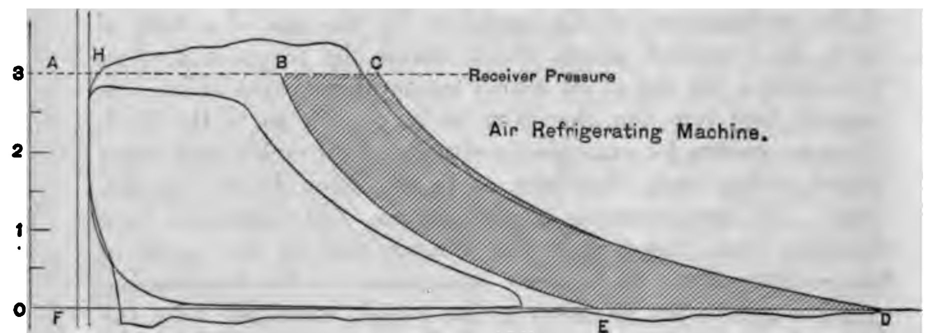


Fig. 135.

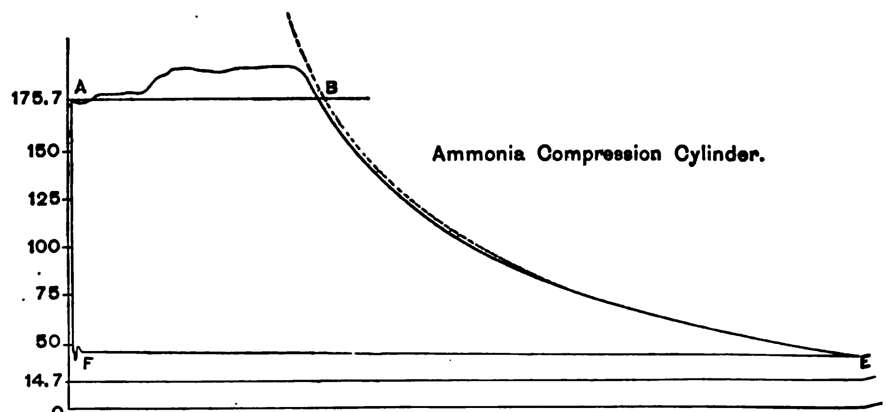


Fig. 136.

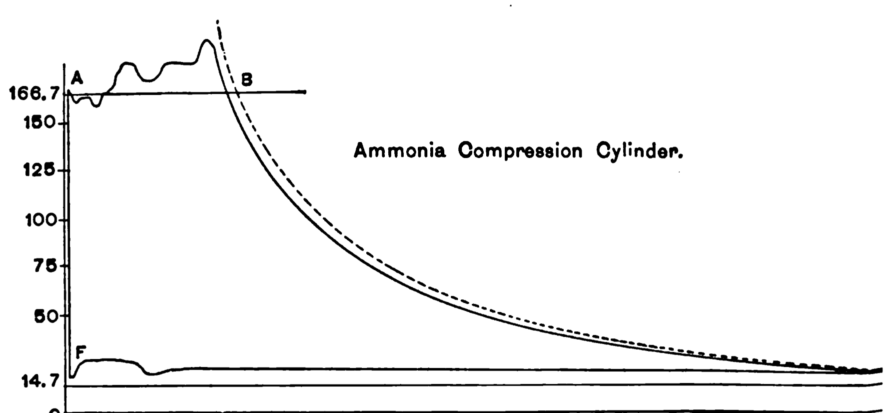


Fig. 137.

Class of Machine.	Authority.	Number of Test.	Dimensions of Compression Cylinder, in inches. Bore.	Stroke.	Absolute Pressure, in lbs. per square inch. Condenser.	Suction.	Temperature corresponding to Pressure in degrees Fahr. Condenser.	Suction.	Temperature of Brine, in degrees Fahr. Inlet.	Outlet.	Revolutions per minute.	Horse-power of Steam-cylinder.	Per cent. of Indicated Power of Steam-cylinder lost in friction.	Ice-melting effect, in tons per 24 hours.	Ice-melting effect in lbs. per lb. of coal at 3 lbs. per hour per H. P. of steam-cylinder of compression machine and an evaporation of 11.1 lbs. of water per lb. of combustible from and at 212° F. in the absorption machine. Theoretical, no superheating during aspiration. No frict'n.	With frict'n.	Actual.	Difference between theoretical ice-melting effect, no superheating or friction, and actual ice-melting effect. Per cent. of theoretical with no friction.	Loss due to heating during aspiration of gas in the compression cylinder and to radiation and superheating at brine tank. Per cent. of theoretical amount with friction.	Mean Effective Pressure, in lbs. per square inch. Actual, including resistance due to inlet and exit valves.	Theoretical, not including resistance due to inlet and exit valves.	Difference between theoretical and actual + resistance due to valves.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
Ammonia cold compression..	Schröter	1	9.9	16.5	135.4	55.3	72.3	26.6	42.8	37.2	44.9	17.9	14.4	26.2	58.67	50.23	40.63	30.8	19.1	54.8	52.9	1.9
	"	2	9.9	16.5	131.4	41.9	70.5	11.3	28.4	23.0	45.1	18.0	16.7	19.5	45.14	37.59	30.01	33.5	20.2	53.4	52.7	0.7
	"	3	9.9	16.5	123.0	30.3	69.2	0.5	14.0	8.8	45.1	16.8	16.0	13.3	35.04	29.44	22.03	37.1	25.2	50.3	48.7	1.6
	"	4	9.9	16.5	126.4	22.2	63.5	—11.8	—0.3	—5.5	44.8	15.5	19.5	9.0	28.29	22.76	16.14	42.9	29.1	44.7	43.5	1.2
	"	5	9.9	16.5	109.5	41.9	95.5	14.4	28.3	23.0	45.0	24.1	10.5	16.5	29.79	26.68	19.07	36.0	28.5	77.0	72.9	4.1
	"	6	9.9	16.5	135.8	60.0	72.4	30.2	43.7	37.2	45.2	17.9	10.7	29.8	64.74	57.85	46.29	28.5	19.9	56.8	52.1	4.7
	"	7	9.9	16.5	131.4	45.1	70.6	17.8	28.3	23.0	45.1	18.0	12.1	21.6	48.40	42.56	33.23	31.3	21.9	56.4	53.1	3.3
	"	8	9.9	16.5	125.6	23.7	68.2	—9.4	—0.4	—5.8	44.7	15.6	18.0	9.9	29.83	24.46	17.55	41.1	28.3	46.1	43.2	2.9
	"	9	9.9	16.5	116.9	41.0	64.2	13.1	28.4	23.0	45.0	16.4	13.5	20.0	50.53	43.70	33.77	33.1	22.9	50.6	46.5	4.1
	"	10	9.9	16.5	130.0	60.3	70.0	30.7	42.8	37.3	31.7	12.0	14.8	19.5	69.40	59.09	45.01	35.2	23.8	52.0	49.3	2.7
Pictet fluid dry compression..	"	11	11.3	24.4	56.7	20.9	77.3	23.5	43.0	37.5	57.0	21.5	22.9	25.6	55.01	42.48	33.07	39.9	22.2	24.1	21.7	2.4
	"	12	11.3	24.4	55.6	14.9	76.2	14.4	28.5	23.0	56.8	20.6	22.9	17.9	41.05	31.71	24.11	41.3	24.0	23.1	20.3	2.8
	"	13	11.3	24.4	54.6	9.7	75.2	—2.5	14.1	8.8	57.1	18.5	24.0	11.6	30.22	23.36	17.47	42.2	25.2	20.4	18.4	2.0
	"	14	11.3	24.4	60.4	6.7	80.6	—15.9	—0.3	—5.7	57.6	15.7	25.7	5.7	22.28	16.50	10.14	54.5	38.5	16.8	15.9	0.9
	"	15	11.3	24.4	90.9	14.9	104.4	14.4	28.3	23.0	59.3	27.2	16.9	15.7	25.16	20.88	16.05	36.2	23.1	31.5	28.4	3.1
	"	16	11.3	24.4	61.0	22.3	81.2	31.5	43.5	37.5	57.3	21.6	14.0	28.1	54.32	46.67	36.19	33.4	22.5	26.8	23.6	3.2
	"	17	11.3	24.4	59.3	15.6	79.6	16.2	28.4	23.0	57.5	20.5	12.8	19.3	40.13	35.01	26.24	34.6	25.0	25.6	21.7	3.9
	"	18	11.3	24.4	58.7	6.7	79.1	—15.9	—0.4	—5.6	57.8	15.9	21.1	6.8	22.72	17.90	11.93	47.5	33.4	18.0	16.6	1.4
	"	19	11.3	24.4	54.3	22.2	74.9	31.3	42.8	37.5	35.3	12.4	22.3	17.0	62.83	49.17	38.04	39.5	22.6	22.6	20.9	1.7
	"	20	11.3	24.4	88.7	15.6	102.9	16.2	28.3	23.1	42.9	19.9	14.7	11.9	26.78	22.86	16.68	37.7	27.0	32.7	28.5	4.2
	"	21	11.3	24.4	62.1	6.5	82.2	—16.9	—0.1	—5.3	34.8	9.9	24.3	3.5	21.51	16.30	9.86	54.2	39.5	17.7	16.8	0.9
Bell Coleman air.	"	22	28.0	23.8	58.8	14.7	64.8*	-52.6*	......	......	63.2	83.2	21.9	10.3	12.09	7.94	3.42	71.7	56.9	26.6	24.5†	2.3
Closed cycle air...	Renwick. Jacobus.	23	10.0	18.0	175.	53.7	81.3*	-40.2*	......	......	93.4	38.1	32.1	4.9	14.8	8.1	3.0	80.	63.	89.2	75.3†	13.9
Ammonia dry compression..	Denton.	24	12.0	30.0	166.	42.7	84.2	15.0	36.8	28.9	58.1	85.0	22.7	73.9	35.91	27.36	24.16	32.7	11.7	65.9	67.8	—1.9
	"	25	12.0	30.0	167.	22.9	84.6	—10.8	6.3	2.0	57.7	72.6	18.6	37.9	23.18	18.78	14.52	37.4	22.7	57.6	57.9	—0.3
	"	26	12.0	30.0	162.	27.7	82.7	—3.2	14.3	2.3	57.9	73.6	19.3	46.5	26.94	21.56	17.55	34.9	18.6	59.9	59.7	0.1
	"	27	12.0	30.0	176.	42.2	87.7	14.5	36.4	28.5	58.9	88.6	19.7	74.4	33.54	26.94	23.31	30.5	13.5	70.5	71.6	—1.1
Ammonia absorp.	"	28	...	....	152.3	40.4	79.1	12.6	20.7	15.7	....	....	....	42.2	38.5	.....	20.1	47.8	...	....	....	....

* Temperature of air at entrance and exit of expansion cylinder.

† Mean effective pressure for adiabatic compression.

the air was compressed, is $-109°$ Fahr., while the actual temperature was only $-53°$. This result is not due to any deficiency in the adiabatic law of temperature, but probably to a rapid absorption of heat in the air by its contact with the metallic parts of the machine. Another cause of loss of theoretical effect is in the fall of pressure through wire-drawing of the air in passing from the compressing to the expanding cylinder. The amount of this is shown by the superposition of the compression and expansion indicator cards in Fig. 135. It will be noticed that the compression line practically coincides with the adiabatic DC, notwithstanding that by water injection to the compression cylinder the temperature of the air at its exit from this cylinder was only about 82° Fahr.

AMMONIA MACHINES—ABSORPTION TYPE.

Several tests of this type of machine have been published by Prof. Schröter, of Munich, as a preliminary report of the Munich Polytechnic Commission, but the data do not permit the "ice-melting effect" to be satisfactorily determined, as the machines were devoted to ice making. Allowing for the difference in the "ice-melting" and ice-producing effects, the results obtained are, however, nearly the same as that given in Table I. for a Pontifex absorption machine, in which the data is approximately complete for scientific purposes. This test was reported in Trans. American Soc. Mech. Engrs., Vol. X., p. 792. The ammonia was worked between 138 and 23 lbs. pressure above the atmosphere. The ice-melting effect per pound of coal, was 20.1 lbs. on a basis of boiler economy equivalent to 3 lbs. of steam per indicated horse-power in a good non-condensing steam-engine. This result realizes 52.2% of the theoretical effect due to pure anhydrous ammonia, with no losses by imperfect action in any part of the apparatus. Detailed results are given in line 28, Table I., and in the appendix. The principal losses are,

First. The steam to drive the ammonia circulating pump, which exhausted into the atmosphere, and used 16.4% of the total steam at the rate of about 150 lbs. per hour per horse-power of the pump.

Second. The heat carried off from the weak liquor by the water from the absorber, which was equivalent to 19.7% of the steam consumption.

Third. Five per cent. of the weight of ammonia circulated is, on theoretical grounds,* believed to have been entrained with the ammonia throughout the cycle of the machine, which would cause 16.7% loss of theoretically perfect effect.

It is probable that more liberal proportions of the interchanger, or improvements in eliminating entrained water, and the saving of circulating pump steam, possible by arranging this pump to exhaust into the generator, would enable the realization of about 65% of the theoretical effect, a figure which, as shown below, is attained by the compression type of machine under average conditions. Under certain conditions of ammonia pressures, theory indicates that the absorption machine is superior in economy to the compression machine, even when the latter has the advantage of being driven by a compound steam-engine. Table I. *a* shows these conditions, but experiments are needed to test the truth of the conclusions. Such experiments are about to be made by the writers on a large Pontifex plant.

AMMONIA MACHINES—COMPRESSION TYPE.

Very complete determinations of "ice-melting effect" of this type of machine are now available. Ten experiments have been published by Prof. Schröter, as secretary of the Munich Commission, using the plant erected for them by the Linde Co. of Germany. Lines 1 to 10 of Table I. give the results, which we have reduced to English units. The "ice-melting effect" (column 17) varies from 46.29 to 16.14, according as the suction pressure varies from about 45 to 8 lbs. above the atmosphere, this pressure being the condition which mainly controls the economy of compression machines. These results are equivalent to realizing from 72 to 57% (col. 19) of the theoretically perfect performance (col. 16). The higher per cents. appear to occur with the higher suction pressures, indicating a greater loss from cylinder superheating (col. 20) as the range of temperature of the gas in the compressing cylinder is greater. See appendix, page 14. These experiments were made with double-action compressors. Lines 24 to 27 show the results of experiments with single-acting compressors on one of the leading makes of American compression machines, reported in the Trans. Amer. Soc. Mech. Engrs., Vol. XII., p. 326. The percentage of theoretical effect realized,

* For argument on which this assumption is based, see Stevens' *Indicator*, January, 1892.

TABLE Ia.

RELATIVE PERFORMANCE OF AMMONIA COMPRESSION AND ABSORPTION MACHINES, ASSUMING NO WATER TO BE ENTRAINED WITH THE AMMONIA GAS IN THE CONDENSER.

It is assumed in the calculation for both the absorption and compression machines, that 1 lb. of coal imparts 10,000 B. T. U. to the boiler. This is equivalent to an evaporation of about 8⅓ lbs. of water per pound of coal from a feed-water temperature of 70° Fahr., and a pressure of 90 lbs. per square inch above the atmosphere. The condensed steam from the generator of the absorption machine is assumed to be returned to the boiler at the temperature of the steam entering the generator. The engine of the compression machine is assumed to exhaust through a feed-water heater that heats the feed-water to 212° Fahr. The engine is assumed to consume 26½ lbs. of water per hour per horse-power. The figures for the compression machine include the effect of friction, which is taken at 15% of the net work of compression.

Number of comparison.	Condenser. Temp. in degrees Fahr.	Condenser. Absolute pressure in lbs. per square inch.	Refrigerating coils. Temp. in degrees Fahr.	Refrigerating coils. Absolute pressure in lbs. per square inch.	Temp. of absorber in degrees Fahr.	Pounds of Ice-melting Effect per Pound of Coal. Compressing machine. Using 3 lbs. of coal per hour per horse-power.	Compressing machine. Using 1.6 lbs. of coal per hour per horse-power.	Absorption * machine. Absorption machine in which the ammonia circulating pump exhausts into the generator.	Absorption * machine. Absorption machine in which the ammonia circulating pump exhausts into the atmosphere through a heater, yielding 212° temp. to the feed-water.	British Thermal Units per Pound of Ammonia circulated through the System. Heat furnished to generator of absorption machine.	Heat furnished to boiler. Compression machine using 3 lbs. coal per hour per horse-power.	Heat furnished to boiler. Absorption machine in which the ammonia circulating pump exhausts into the generator.	Heat furnished to boiler. Absorption machine in which the ammonia circulating pump exhausts into the atmosphere through a heater.	Refrigerating effect. (Same for both systems.)	Heat abstracted from condenser. (Same for both systems.)	Equivalent of work of compression. (Same for both systems.)	Abstracted from absorber of absorption machine.	Heat wasted in exhaust of compression mach. and by friction. Using 3 lbs. of coal per hour per H.P.	Heat wasted in exhaust of compression mach. and by friction. Using 1.6 lbs. of coal per hour per H.P.	Quantities of heat, in p. c. of amount furnished to the boiler, lost in exhaust of compression machine and by friction. Using 3 lbs. of coal per hour per H.P.	Quantities of heat, in p. c. of amount furnished to the boiler, lost in exhaust of compression machine and by friction. Using 1.6 lbs. of coal per hour per H.P.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
1	61.2	110.6	5	33.7	61.2	38.1	71.4	38.1	33.5	969	969	969	1,100	524	596	72	897	897	445	92.6	86.1
2	59.0	106.0	5	33.7	59.0	39.8	74.6	38.3	33.9	967	929	967	1,090	526	595	69	898	860	426	92.6	86.1
3	59.0	106.0	5	33.7	130.0	39.8	74.6	39.8	35.1	931	929	931	1,054	526	595	69	862	860	426	92.6	86.1
4	59.0	106.0	—22	16.9	59.0	23.4	43.9	36.3	31.5	1,000	1,550	1,000	1,151	516	631	115	885	1,435	711	92.6	86.1
5	86.0	170.8	5	33.7	86.0	25.0	46.9	35.4	28.6	988	1,399	988	1,221	497	601	104	884	1,295	642	92.6	86.1
6	86.0	170.8	5	33.7	130.0	25.0	46.9	36.2	29.2	966	1,399	966	1,199	497	601	104	862	1,295	642	92.6	86.1
7	86.0	170.8	—22	16.9	86.0	16.5	30.8	33.3	26.5	1,025	2,077	1,025	1,287	486	640	154	871	1,923	953	92.6	86.1
8	86.0	170.8	—22	16.9	130.0	16.5	30.8	34.1	27.0	1,002	2,077	1,002	1,264	486	640	154	848	1,923	953	92.6	86.1
9	104.0	227.7	5	33.7	104.0	19.6	36.8	33.4	25.1	1,002	1,707	1,002	1,332	476	603	127	875	1,580	786	92.6	86.1
10	104.0	227.7	—22	16.9	104.0	13.5	25.3	31.4	23.4	1,041	2,421	1,041	1,399	466	646	180	861	2,241	1,110	92.6	86.1

* 5% of water entrained in the ammonia will lower the economy of the absorption machine about 15% to 20% below the figures given in the table.

ranges from 69.5 to 62.6%. The friction losses (col. 14) are higher for the American machine. The latter's higher efficiency may be attributed, therefore, to more perfect displacement. No clearance figures are published by Professor Schröter, but a comparison of the clearance action in the two types of machines, by means of the indicator cards, proves that the more perfect displacement is not caused by excessive clearance spaces (see col. 6, Table III.), but by the greater loss by cylinder superheating (col. 20, Table I.). This result is possibly ascribable to the smaller dimensions of the compression cylinders of the German machine, which afford the greater surface in proportion to the volume of gas compressed per stroke. The largest "ice-melting effect" in the American machine is 24.16 lbs. This corresponds to the highest suction pressures used in American practice, for such refrigeration as is required in beer storage cellars. The conditions most nearly corresponding to American brewery practice in the German tests, are those in line 5, which give an "ice-melting effect" of 19.07 lbs. Line 2, with a condensing pressure more favorable to economy of coal, but less favorable for economy of water, gives 30.01 lbs., but the per cent of perfect action realized is about 1% less than for the American machine, notwithstanding that the latter's loss by friction is 6% greater.

For the manufacture of artificial ice, the conditions of practice are those of lines 3 and 4, and lines 25 and 26. In the former the condensing pressure used requires more expense for cooling water than is common in American practice. The "ice-melting effect" is therefore greater in the German machine, being 22.03 and 16.14 lbs. against 17.55 and 14.52 for the American apparatus. The percentage of theoretical effect realized again averages greatest in the single-acting machine, however, notwithstanding that the friction is less in the German apparatus.

The superheating of the ammonia gas by the warm cylinder walls during its entrance into the compressor, thereby rarefying it, so that, to compress a pound of ammonia, a greater number of revolutions must be made by the compressing pumps, than corresponds to the density of the ammonia gas as it issues from the brine tank, causes the principal loss of the theoretical effect given in col. 16, Table I.

In this theoretical estimate of the refrigerating effect, it is assumed that this density measures the displacement of compressor necessary to circulate a pound of the gas through the refrigerating coils, but experimental measurement of the weight of ammonia circulated, having been made in connection with the tests of lines 24 to 27, Table I., by metering the liquid ammonia (see Table IV., appendix, page 20), it was found that from 15 to 30% less ammonia is circulated than is accounted for by multiplying the compressor displacement by the density corresponding to the pressure and temperature of the ammonia gas at its entrance to the compressor.

Hence, as the pressure does not vary, the density must decrease by the rarefaction of the gas. Estimate shows that the heat, to rarefy the gas, so as to account for the above discrepancy, could be lodged in the cylinder walls, and cause the same action that is represented by cylinder condensation in steam-engines. The mean effective pressure is the same (see Figs. 136 and 137, and columns 21 to 23, Table I.) for any range of pressure in the compression cylinder, whatever the density. Hence, the increase of power to operate the machine is directly proportional to the increase of volume due to the heating produced by the cylinder walls as the ammonia enters. In the case of "cold compression" machines or those in which the ammonia is injected in the cylinder in order to prevent superheating during compression, there is an increase of volume, by cylinder heating due to the evaporation of a *portion* of the ammonia injected. In the case of dry compression, the increase of volume is produced by superheating the gas.

Besides that due to heating at entrance to the cylinder, the discrepancies between the theoretical and experimental results in ammonia compression machines are as follows:

1st. The liquid ammonia may be cooled by the condensing water below the boiling point corresponding to condensing pressure. This increases the useful effect, because there will be a less per cent. of each pound of ammonia vaporized in the brine tank, to chill the substance from the temperature at which it arrives at the expansion cock, to the boiling point corresponding to such pressures. Such gain averages about 4% in the case of tests 24 to 27, Table I. See appendix, page 519.

2d. The ammonia gas may be superheated before leaving the brine tank, above the boiling point due to suction pressure. This makes a temporary gain in the refrigerating effect obtained from a pound of the substance, amounting to about 2% in tests

24 to 27; but this gain is offset by the loss of effect due to the extra compression power, required by the increase of volume caused by the superheating.

3d. The ammonia gas is superheated by absorbing heat from the atmosphere, in passing from the brine tank to the compressing cylinder. This in tests 24 to 27 causes a loss by increase of compression power of about 2%.

4th. There are also small losses by radiation from the brine tank, which are probably not more than 1%.

On the whole, therefore, the loss of effect in ammonia compression machines is practically entirely due to cylinder superheating, which reduces the ice-melting effect from 14 to 23% of the theoretical amount with friction considered, being roughly in proportion to the range of temperature in the compression cylinder.

Tests in which the balance of heat is within 3%, afford values for the latent heat, agreeing with the best physical values, to within from 1 to 4%. See Tables VI. and VII., appendix.

SULPHUR-DIOXIDE OR PICTET MACHINES.

No records are available for determination of the "ice-melting effect" of machines using pure sulphur-dioxide. This fluid is in use in American machines, but in Europe it has given way to the "Pictet fluid," a mixture of about 97% of sulphur dioxide and 3% of carbonic acid. The presence of the carbonic acid affords a temperature about 14 Fahr. degrees lower than is obtained with pure sulphur dioxide at atmospheric pressure. The latent heat of this mixture has never been determined, but is assumed to be equal to that of pure sulphur dioxide. The Munich Commission have published tests of a "Pictet fluid" machine, erected, for testing, in their laboratory.

These tests are given in English units, in lines 11 to 21, Table I.

For brewery refrigerating conditions, line 17, we have, 26.24 lbs. "ice-melting effect," and for ice-making conditions, line 13, in which the "ice-melting effect" is 17.47 lbs. These figures are practically as economical as those for ammonia, the per cent. of theoretical effect realized ranging from 65.4 to 57.8%. At extremely low temperatures,—15° Fahr., lines 14 and 18, the per cent. realized is as low as 42.5%. This is less than any ammonia efficiency, but so low a temperature was not tried with this substance. The Pictet machine was at a disadvantage in the

German tests. In these tests the range of temperature of the circulating water was made the same in the two machines, and in tests of the sulphur dioxide machine less circulating water was used than in parallel tests of the ammonia machine, so that the condenser pressure was higher in the sulphur dioxide than it would have been had the machines been furnished with the same amount of condensing water.

On the whole the tests do not indicate any serious inferiority of efficiency in the Pictet system, and, theoretically, there should be no practical difference between the fluids.

It will be observed that sulphur dioxide is susceptible to the same loss of cylinder superheating as is ammonia. A similar loss occurs in compressing air, which for pressures of several hundred pounds appear to be considerable, as in the case in volatile vapors; but for pressures of 100 lbs., the experiments of Messrs. Gause and Post at Hoboken show that the cylinder superheating, with jacket cooling, reduces the displacement only about 8%.

CARBONIC ACID MACHINES.

No definite data exist regarding the performance of these machines. There seem to be no reasonable grounds for their use, as the pressures involved reach 800 lbs. per square inch, and their efficiency is considerably less than ammonia machines. The compression cylinder is 4 times less in volume than is necessary for ammonia, but the high pressures required more than offset this advantage.

Tables II., III., V., VI., and VII. of the appendix enable the calculations for all figures deduced, to be followed in detail.

APPENDIX I.

RESULTS OF CALCULATIONS.

The conclusions arrived at for compression ice machines are:

1st. If the effect of heating of the gas during the time that it is drawn into the compression cylinder is included in the analysis, the theoretical agrees with the actual performance.

2d. The mean effective pressure of the gas in the compression cylinder, as deduced by thermodynamic laws, agrees with that obtained in practice.

3d. The effect of heating of the gas during the time that it is drawn into the compression cylinder, can be determined only by means of direct tests on the machines. This effect is similar to that of cylinder condensation in the steam-engine, and can be expressed by no exact theoretical law.

4th. Special determinations of the latent heat of ammonia indicate that the figures given by Ledoux are correct, within the possible error in determining the experimental latent heat, by the method herein described, which is about 5%.

In order to determine the various losses in the compression machine, we must know the temperatures at many more points than are required in a commercial test. We are able, however, if the range of pressure is given, to compare the commercial efficiency with the efficiency corresponding to perfect action and no superheating of the gas during admission to the cylinder. Table I. gives such a comparison.

In Table I. the friction of the machines employed in obtaining the theoretical ice-melting effect with friction included is taken at that experimentally determined for the test in question.

The only tests of compression machines given in Table I. that give sufficient data to estimate separately each of the losses, are Nos. 24 to 27. These are for dry compression, so that the increase of volume during aspiration to the compression cylinder is produced by superheating the ammonia vapor. Selecting Nos. 25, 26, and 27, as representing the extreme and intermediate conditions, we obtain the following results:

	Test No. 25.	Test No. 26.	Test No. 27.
Difference of temperatures corresponding to pressures of ammonia in condenser and refrigerating coils, in degrees Fahr.	95.8	85.9	78.2
Weight of ammonia circulated per minute by meter. Lbs.	14.89	17.04	28.41
Theoretical "ice-melting effect;" no superheating. Lbs. ice per lb. of coal	18.78	21.56	26.94
Actual "ice-melting effect." Lbs. of ice per lb. of coal	14.52	17.55	23.31
Losses in per cent. of theoretical amount	22.7	18.6	13.5

The above losses may be separated as follows:

Losses due to increase of power in per cent. of theoretical ice-melting effect.	Superheating at brine tank	4.2	0.9	2.6
	Superheating by radiation between the brine tank and compressor	3.0	1.5	1.0
	Superheating by cylinder walls	22.8	20.8	14.0
	Total loss	30.0	23.2	17.6

The gain in refrigerating effect due to superheating and to cooling the anhydrous ammonia below the temperature corresponding to the pressure in the refrigerating coils must be subtracted from the total loss due to increase of work, in order to obtain the net loss. We have, for the amount gained:

		Test No. 25.	Test No. 26.	Test No. 27.
Gain in refrigerating effect in per cent. of theoretical ice-melting capacity	Cooling liquid below temperature corresponding to the pressure in the refrigerating coils	3.7	4.2	2.4
	Superheating in brine tank	2.6	0 4	1.7
	Sum of amounts gained	6.3	4.6	4.1
	Difference between losses and amounts gained, or net loss	22.7	18.6	13.5

The above losses of effect are due to the increase of volume of a pound of saturated vapor by superheating. The increase of volume above that corresponding to saturation, in per cent. of the saturated volume, is as follows:

	Test No. 25.	Test No. 26.	Test No. 27.
Due to superheating at brine tank	5.4	1.1	3.0
Due to radiation between brine tank and compressor	2.5	1.8	1.1
Due to superheating of cylinder walls	29.6	25.6	16.2
Total increase of volume	37.5	28.5	20.3

The superheating does not alter the range of pressures, hence the power to produce compression increases as the volume, and the refrigerating effect per cubic foot of compressive displacement or per horse-power is thus largely reduced.

A similar summation of the losses in Test No. 27 of the absorption machine, is approximately as follows: *

COMPARISON OF THE ECONOMY ACTUALLY OBTAINED, WITH THE RESULTS CALCULATED BY MEANS OF LEDOUX'S EQUATIONS.

Ice-melting capacity per pound of coal, obtained by test, assuming that each pound of coal imparts 10,000 B. T. U. to the boiler.....	20.1 lbs.
Ice-melting capacity per pound of coal, by Ledoux's equations, assuming a perfect heater action, no drip liquor, no water carried over by the ammonia, and no losses by radiation †....................	38.5 lbs.
The calculated losses in per cent. of the amount of heat imparted to the boiler, or 1,465,500 × 1,930 + 1,614 = 1,752,400 B. T. U. per hour, are: ‡	
Heater..	19.7%
Drip liquor..	3.9%
5% water entrained with the ammonia..........................	16.7%
Ammonia circulating pump..................................	16.4%
Radiation..	0.9%
	57.6%
There is a gain in refrigerating capacity in the machine tested, due to superheating the ammonia in the cooler, which is equivalent to.....	9.9%
Net loss..	47.7%
Equivalent of losses in ice-melting capacity............38.5 × .477	18.4 lbs.
Theoretical useful ice-melting capacity ....38.5 − 18.4	20.1 lbs.

In this test the amount of water entrained in the ammonia is assumed at 5%. There are no other tests of absorption machines which give enough data to allow an exact estimate of all the losses to be made. One of the most important factors is also missing from the data of test No. 27; this is, the weight of drip liquid returned from the condenser to the generator. The heat of dissociation of ammonia from water, has also not been determined for pressures other than that of the atmosphere, so that an

* From Stevens' *Indicator*, Jan., 1892.

† The conditions substituted in Ledoux's equations are: Temperature of condenser, 80° Fahr.; temperature of cooler, 13° Fahr.; temperature of absorber, 180° Fahr.

‡ The per cents. of loss, with the exception of that due to the 5% of water entrained with the ammonia, are found by dividing the heat expended, as shown in the text, by the total heat supplied to the boiler. The loss due to 5% of entrained water is found by comparing the efficiency of the machine when working with 5% of entrained water, with that obtained when no water is carried over with the ammonia. The latter calculations are given in the revised work of Ledoux on Ice-Making Machines, now passing through the press.

assumption is introduced in considering it constant at all pressures. The probable error of this assumption is, however, very small.

A theoretical discussion of the method of obtaining the approximate losses is given in the *Stevens' Indicator*, Jan., 1892.

In tests No. 25, 26, and 27, the amount of ammonia was measured by the means of a meter, so that we are able to compare the theoretical weight of ammonia with that actually circulated. Employing Ledoux's table of the properties of ammonia, we obtain the weight of ammonia circulated per minute calculated from the heat abstracted from the brine tank, which is compared below with the actual amount.

	Test No. 25.	Test No. 26.	Test No. 27.	Average.
Actual by meter	14.89	17.04	28.61	20.18
Calculated	14.34	17 64	28.35	20.11

We may employ the data in a reverse manner, and obtain the values of the latent heats. This gives the following:

	Test No. 25.	Test No. 26.	Test No. 27.	Average.
Temperature corresponding to pressure of refrigerating coils	−10.7	−8.2	14.5	
Latent heat by direct measurement	569.6	603.5	570.4	581.2
Latent heat by Ledoux's table	589.1	585.1	575.1	583.1

These results are nearly identical, and thus indicate that Ledoux's table may be relied on in practical use.

The latent heats at the temperature corresponding to the pressure in the condenser, obtained in a similar way, are

	Test No. 25.	Test No. 26.	Test No. 27.	Average.
Temperature corresponding to pressure of condenser	84.6	82.7	87.7	85.0
Latent heat by direct measurement	524.8	525.7	512.4	521.0
Latent heat by Ledoux's table	528.4	529.8	526.1	528.1

The greatest difference between the experimental results and those given by Ledoux's table is seen to be about 4%.

METHOD OF CALCULATING THE THEORETICAL RESULTS. ALL QUANTITIES IN BRITISH THERMAL UNITS AND FAHRENHEIT DEGREES.

In calculating Table I., Ledoux's formulas* have been employed.

* "Ice-Making Machines," by M. Ledoux, *Annales des Mines*, 1878; also Van Nostrand's Science Series. A revised edition of the Science Series translation is now in the hands of the printer.

For ammonia and sulphur-dioxide compression machines, the theoretical heat equivalent of the work of compression is calculated by taking the difference between the amount of heat abstracted at the condenser and that imparted to the refrigerating coils. Calling Q_1 the heat abstracted at the condenser, and Q that imparted to the refrigerating coils, we have, per pound of ammonia circulated,

$$Q_1 = c_p (t_1 - t_1') + r_1'$$

and

$$Q = \lambda_2 - q_1'.$$

In which

c_p = specific heat of the gas at constant pressure.
t_1 = temperature of superheated gas at end of compression.
t_1' = temperature corresponding to pressure of condenser.
r_1' = latent heat of ammonia at temperature corresponding to pressure of condenser.
q_1' = sensible heat of liquid ammonia at the temperature corresponding to the pressure at the condenser.
λ_2 = total heat of liquid ammonia at the temperature corresponding to the pressure in the refrigerating coils.

In order to calculate the temperature at the end of compression we make one of the formulas

$$T_1 = T_2 \left(\frac{P_1}{P_2}\right)^{\frac{AB}{c_p}}.$$

In which T_1 and T_2 are the absolute temperatures corresponding to t_1, and t_2 and P_1 and P_2 the absolute pressures of the vapor in the condenser and refrigerating coils. The value of the exponent $\frac{AB}{c_p}$ is .24 for ammonia and .21 for sulphur-dioxide.

Having calculated Q_1 and Q, we have for equivalent in thermal units of the work to compress one pound of vapor,

$$W_r = Q_1 - Q.$$

If v_2 is the volume in cubic feet of 1 lb. of vapor at the temperature of the refrigerating coils, we have for the mean effective pressure in pounds per square inch,

$$M.E.P. = \frac{772 (Q_1 - Q)}{144 v_2}.$$

the air was compressed, is $-109°$ Fahr., while the actual temperature was only $-53°$. This result is not due to any deficiency in the adiabatic law of temperature, but probably to a rapid absorption of heat in the air by its contact with the metallic parts of the machine. Another cause of loss of theoretical effect is in the fall of pressure through wire-drawing of the air in passing from the compressing to the expanding cylinder. The amount of this is shown by the superposition of the compression and expansion indicator cards in Fig. 135. It will be noticed that the compression line practically coincides with the adiabatic DC, notwithstanding that by water injection to the compression cylinder the temperature of the air at its exit from this cylinder was only about 82° Fahr.

AMMONIA MACHINES—ABSORPTION TYPE.

Several tests of this type of machine have been published by Prof. Schröter, of Munich, as a preliminary report of the Munich Polytechnic Commission, but the data do not permit the "ice-melting effect" to be satisfactorily determined, as the machines were devoted to ice making. Allowing for the difference in the "ice-melting" and ice-producing effects, the results obtained are, however, nearly the same as that given in Table I. for a Pontifex absorption machine, in which the data is approximately complete for scientific purposes. This test was reported in Trans. American Soc. Mech. Engrs., Vol. X., p. 792. The ammonia was worked between 138 and 23 lbs. pressure above the atmosphere. The ice-melting effect per pound of coal, was 20.1 lbs. on a basis of boiler economy equivalent to 3 lbs. of steam per indicated horse-power in a good non-condensing steam-engine. This result realizes 52.2% of the theoretical effect due to pure anhydrous ammonia, with no losses by imperfect action in any part of the apparatus. Detailed results are given in line 28, Table I., and in the appendix. The principal losses are,

First. The steam to drive the ammonia circulating pump, which exhausted into the atmosphere, and used 16.4% of the total steam at the rate of about 150 lbs. per hour per horse-power of the pump.

Second. The heat carried off from the weak liquor by the water from the absorber, which was equivalent to 19.7% of the steam consumption.

Third. Five per cent. of the weight of ammonia circulated is, The remaining steps in the calculations are indicated in Tables II. and III., which are introduced to give the detailed steps gone through with in the calculations for the vapor compression machines. Ledoux's table for the properties of ammonia, and Zeuner's table for sulphur-dioxide, are employed in performing the calculations.

TABLE III.

CALCULATION OF THEORETICAL MEAN EFFECTIVE PRESSURES GIVEN IN TABLE I.

No. of Test.	Temperature corresponding to pressure of ammonia in refrigerating coils, in deg. Fahr.	Density corresponding to this pressure, in pounds per cubic foot.	Thermal equivalent of work of compression per pound of ammonia. Col. Q – col. R, Table II.	Mean effective pressure in pounds per square inch, for full displacement of compression cylinder. Col. 4 × col. 3 × 5.4	Net displacement of compression cylinder in terms of the total volume corresponding to full displacement.	Theoretical mean effective pressure for actual displacement, in pounds per square inch.	*Cheval vapeur* from Schröter's tests.	Revolutions per minute.	Observed mean effective pressure, in pounds per square inch. $\frac{\text{Col. 8}}{\text{Col. 9}} \times 158.5$.*	Difference between actual and theoretical mean effective pressures, including the increase due to valve resistance. Col. 10 – col. 7.
1	2	3	4	5	6	7	8	9	10	11
1	26.6	.1906	53.3	54.86	.964	52.9	15.53	44.91	54.8	1.9
2	14.3	.1480	68.9	55.06	.958	52.7	15.20	45.10	53.4	0.7
3	0.5	.1095	88.1	52.11	.935	48.7	14.31	45.05	50.3	1.6
4	–11.8	.0824	108.2	48.15	.904	43.5	12.63	44.76	44.7	1.2
5	14.4	.1483	98.8	79.12	.922	72.9	21.86	44.97	77.0	4.1
6	30.2	.2042	48.4	53.37	.977	52.1	16.20	45.18	56.8	4.7
7	17.8	.1597	64.4	55.54	.956	53.1	16.06	45.12	56.4	3.3
8	–9.4	.0872	102.9	48.45	.891	43.2	13.00	44.74	46.1	2.9
9	13.1	.1444	62.3	48.58	.958	46.5	14.38	45.03	50.6	4.1
10	30.7	.2061	45.4	50.53	.976	49.3	10.39	31.68	52.0	2.7

* Volume of compression cylinder = .718 cubic foot.

$$M.E.P. \text{ in lbs. per square inch} = \frac{\textit{Cheval vapeur} \times .9863 \times 33000}{.718 \times 2 \times \text{Rev.} \times 144}.$$

In calculating the distribution of the losses, tests Nos. 25, 26, and 27 are employed. These tests are selected from those reported to this Society at the Richmond meeting, November, 1890, which are the only tests that give the weight of anhydrous ammonia circulated through the system. A revised balance of the heat received and rejected, and the weight of the ammonia circulated per minute, is given in Table IV.

It may be seen in Table IV. that the heat removed by the

TABLE IV.

BALANCE OF HEAT RECEIVED AND REJECTED IN A 75-TON REFRIGERATING MACHINE OF THE AMMONIA COMPRESSION TYPE.*

	Brine.									Condenser taken over a period during which the conditions are uniform.			Balance of heat received and rejected.								
No. of test.	Weight of brine circulated per minute.	Range of temperature of the brine.	Specific heat of the brine.	Heat given up per minute by brine to the refrigerating coils, in B. T. U. Col. 2 × col. 3 × col. 4.	Amount that the temperature of the brine in tank is raised or lowered during the test. If raised, – ; If lowered, +.	Corresponding corrections in B.T.U. per minute.	Cooling effect per minute, in B.T.U. Col. 5 + col. 7.	Horse-power of steam-cylinder.	Ice-melting effect per pound of coal. Col. 8 × 60 / 142.2 × 3 × col. 9	Range of temperature.	Water circulated per minute.	Corresponding heat per minute, in B.T.U.	Heat given to ammonia by compressor, in B. T. U.	Heat given up by the ammonia to the jackets, in B. T. U.	Heat given to ammonia by compressor and by radiation, in B.T.U. (calculated).	Heat given up by jackets and radiation, in B. T. U.	Total heat given to the ammonia, in B.T.U. Col. 8 + col. 16.	Total heat given up by the ammonia, in B.T.U. Col. 13 + col. 17.	Amount not accounted for, in B.T.U. Col. 18 – col. 19.	Amount not accounted for in per cent. of total heat given to the ammonia. Col. 20 × 100 ÷ col. 18.	Weight of ammonia circulated per minute, in pounds.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
23	2,296.	7.90	0.818	14,837.	−2.2	−238.	14,599.	85.0	24.16	39.01	443.2	17,289.	2,808.	600.	2,880.	847.	17,479.	18,136.	−657.	3.7	......
24	2,175.	4.24	0.801	7,387.	+1.0	+108.	7,495.	72.6	14.52	28.75	314.6	9,045.	2,526.	713.	2,613.	1,160.	10,108.	10,205.	− 97.	1.0	14.89
25	943.	12.00	0.799	9,041.	+1.0	+144.	9,185.	73.6	17.55	40.56	255.9	10,379.	2,538.	656.	2,610.	999.	11,795.	11,378.	+ 417.	3.5	17.04
26	2,374.	7.91	0.786	14,760.	−0.5	− 72.	14,688.	88.63	23.31	28.86	560.6	16,179.	3,044.	407.	3,117.	454.	17,805.	†16,633.	+ 333.	1.9	23.61

* See *Transactions American Society of Mechanical Engineers*, Vol. XII., p. 396.

† In this test, 839 B.T.U. were eliminated per minute by means of a special jacket placed on the pipe leading the gas from the compressor to the brine tank, this quantity is not included in the above figure, and therefore must be added to it in obtaining the balance of heat.

creases until the wave is 10 feet high; but any increase in height of wave above 10 feet will have no effect on the vane, which will go on just as if actuated by a 10-foot wave. The vane will thus never be called on to handle more than the energy due to a wave 10 feet high, and can thus work safely in any storm. The vane being built strong enough to resist the pressures due to a wave 10 feet high, will not be called on to resist any additional pressure during a storm, as the upper end of the vane is entirely below the reach of the crests of larger waves.

Mr. Steen's objection that the journals in the apparatus of Fig. 131 will be injured by the action of the sand, would be perfectly valid if the journals were unprotected and made in the ordinary way. This could, however, be obviated by making the connection at the bottom consist of a few links of chain, instead of an ordinary journal. But I do not consider this particular apparatus as practically very valuable. I merely introduced it into this paper as a step in the logical development of the other two motors, Figs. 132 and 133, the latter of which I regard as the practical apparatus which is likely to be thoroughly successful. It seems to me that the motor shown in Fig. 133 is a little better and nearer a scientific and logical apparatus than any other with which I am acquainted; and the statements of several gentlemen at this meeting have tended to confirm me in this belief.

In the practical construction of this apparatus, I should probably make the supporting structures of a conical shape, the lower portion being hollow and filled with concrete, and the upper portion being open lattice-work, so as to interfere as little as possible with the motion of the wave. I should build the whole apparatus ashore, and with the aid of lighters float it to its proper location. I would then sink it to rest on the bottom by pumping the concrete into the hollow portion of the supporting structures, after which I would further secure it by anchors imbedded in the bottom at some distance from the apparatus. The structures being 50 to 60 feet apart, and the centre of gravity being kept very low, there would be a very large stability developed, sufficient to prevent any damage in a storm.

Mr. Steen.—It seems to me that it is fundamental that you should have something that would protect itself.

Mr. Stahl.—The energy of a wave, per foot of breadth, varies as the square of the height and as the length. Now, suppose that I have decided, after computing, that it would pay me to put up a motor to handle a wave 10 feet high, and that a wave that is higher than that occurs so seldom that it would not pay to construct for that strain. You can build the motor strong enough for any size wave, but practically, commercially, you have to stop at a certain point. Now, I make the top of my vane so that it is 5 feet above water, on the basis of a 10-foot wave. The wave crest, when the wave is 10 feet high, will just touch the top of that vane. Suppose the storm wave is 20 feet high, the upper five feet of crest will pass over the top of the vane, without exerting any pressure on the latter; the rest of the water all the way down is doing its work just as it was when the wave was 10 feet high. The motor is thus never called upon to resist more than the pressure due to a 10-foot wave, and having been constructed for that pressure, needs no further protection. If it is considered desirable to build for a greater wave, it can of course be done.

Mr. Steen.—May I ask what is the maximum force of a wave here on this coast at the depth you propose to work it?

Mr. Stahl.—If you will consult the table given in the paper you will get the horse-power per foot of breadth of all the waves likely to occur. The horse-power of any wave whatever can be readily computed by means of formula (28).

TABLE VI.

CALCULATION OF THE THEORETICAL WEIGHT OF AMMONIA CIRCULATED PER MINUTE AND OF THE LATENT HEATS AT THE TEMPERATURES CORRESPONDING TO SUCTION PRESSURES FOR TESTS NOS. 25, 26, AND 27, OF TABLE I.

No. of Test.	Cooling effect per pound of ammonia, in B. T. U. Col. 8, Table V.	Total heat imparted to the brine per minute, in B.T.U. Col. 8, Table IV.	Theoretical weight of ammonia circulated per minute, in pounds. Col. 3 + col. 2.	Actual weight of ammonia circulated per minute, in pounds. Col. 22, Table IV.	Heat imparted to the brine per pound of ammonia actually circulated, in B.T.U. Col. 3 + col. 5.	Sensible Heat of Ammonia above 32° Fahr., in B.T.U. At entrance to refrigerating coils.	Sensible Heat of Ammonia above 32° Fahr., in B.T.U. At temperature corresponding to pressure of refrigerating coils.	Heat required to produce superheating at brine tank per pound of ammonia, in B. T. U. .51. col. 7, Table V.	Experimental latent heat of evaporation at temperature corresponding to the pressure in the refrigerating coils, in B.T.U per lb. Col. 6+col. 7−col. 8−col. 9.	Latent heat by Ledoux's table, in B. T. U. per lb.
1	2	3	4	5	6	7	8	9	10	11
25	522.8	7495.	14.34	14.89	503.36	37.53	−41.08	12.39	569.6	589.1
26	520.6	9185.	17.64	17.04	539.03	32.93	−34.13	2.55	603.5	585.1
27	518.1	14688.	28.35	28.61	513.39	46.99	−17.29	7.29	570.4	575.1

TABLE VII.

CALCULATION OF LATENT HEATS OF ANHYDROUS AMMONIA AT TEMPERATURES CORRESPONDING TO THE PRESSURES IN CONDENSER, TESTS NOS. 25, 26, AND 27 OF TABLE I.

No. of Test.	Heat given to the condensing water per minute, in B. T. U. Col. 13., Table IV.	Heat radiated per minute from the condenser, in B. T. U.	Total heat given up by the ammonia at the condenser per minute, in B. T. U. Col. 2 + col. 3.	Heat given to the condenser per pound of ammonia, in B. T. U. Col. 4 ÷ col. 5, Table VI.	Temperature of superheated gas on entering the condenser, in degrees Fahr.	Temperature corresponding to pressure of ammonia in condenser, in degrees Fahr.	Heat given up by the ammonia in cooling to its temperature of saturation. .53 * (Col. 6 − col. 7).	Sensible heat of ammonia above 32° Fahr. at temperature corresponding to pressure in condenser, in B. T. U.	Temperature of liquid ammonia leaving the condenser, in degrees Fahr.	Sensible heat of ammonia above 32° Fahr. corresponding to temperature at leaving condenser in B.T.U.	Difference in sensible heats at condensing temperature and on leaving the condenser, in B. T. U. Col. 9 − col. 11.	Experimental latent heat of evaporation, in B. T. U. per pound. Col. 5 − (col. 8 + col. 12).	Latent heat of evaporation by Ledoux's table, in B. T. U. per pound.
1	2	3	4	5	6	7	8	9	10	11	12	13	14
25	9045	92	9137	613.63	218	84.6	70.70	55.7	68.0	37.53	18.17	524.8	528.4
23	10379	72	10451	613.32	209	82.7	66.94	53.6	63.7	32.93	20.67	525.7	529.8
27	16179	47	16226	567.14	168	87.7	42.56	59.2	76.7	46.99	12.21	512.4	526.1

* Specific heat at the condenser pressure determined by an experiment on a large scale by placing a water-jacket over the pipe leading from the compressor to the brine tank. (See *Transactions of American Society of Mechanical Engineers*, Vol. XII., page 326.)

CCCCXCIII.*

THE MEASUREMENT OF POWER.

BY THOMAS GRAY, TERRE HAUTE, IND.
(Member of the Society.)

One of the most important, and at the same time most troublesome, problems in mechanical engineering is the measurement of the power produced by a motor or given to a machine. In this paper a brief description is given of the principles of three forms of dynamometer which have been used for this purpose with fairly successful results.

I. Fig. 138 illustrates a form of dynamometer which may be used either to measure the power given to any machine, as, for instance, a dynamo, or to measure the total power given out by an engine which is driving a number of machines. Suppose that *B* is the crank shaft of an engine. A cross-head such as, for a different purpose, is here represented by *J*, is keyed to the shaft, and the pulley *P*, which in this case is the driving pulley, is connected to the shaft by means of four links, *E*, *M*, *F*, *N*, connecting the cross-head to a pair of double bell-crank levers, *KLO* and *CDG*, mounted on bearings fixed to the pulley. The ends of the arms *O* and *G* bear against one end of a rod *R* passing along the axis of the hollow shaft of the pulley. When the shaft *B* is turned, two of the arms of the cranks, as *F* and *M* or *E* and *N*, are pulled toward the cross-head, and thus tend to push the rod *R* outward. This is resisted by the bell-crank lever *S*, which is supported by a plate resting on a diaphragm which closes the top of the cylinder *T*. The cylinder *T* is either partly filled with mercury and partly with water or filled with water and piped to a mercury or other form of pressure gauge placed in any convenient position. The pressure on the end of *R* is thus resisted by a column of mercury the height of which will be directly as the pressure and inversely as the size of the plate

* Presented at the San Francisco Meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

resting on the diaphragm. Thus by finding the indication of the gauge when a known pull or push is applied to S at the point of contact of R, the turning moment exerted by the engine can be calculated for any value of the reading on the gauge. The effect of centrifugal force on the levers attached to the pulley P must be eliminated, and this is done by counterpoise masses applied as indicated by the dotted lines inside the pulley. When the speed of the engine is known, the indications of the pressure-gauge give the means of determining the horse-power being transmitted at any instant. When a record of the variation of the resistance which the engine experiences is required, the tube V is provided with a float carrying a record-

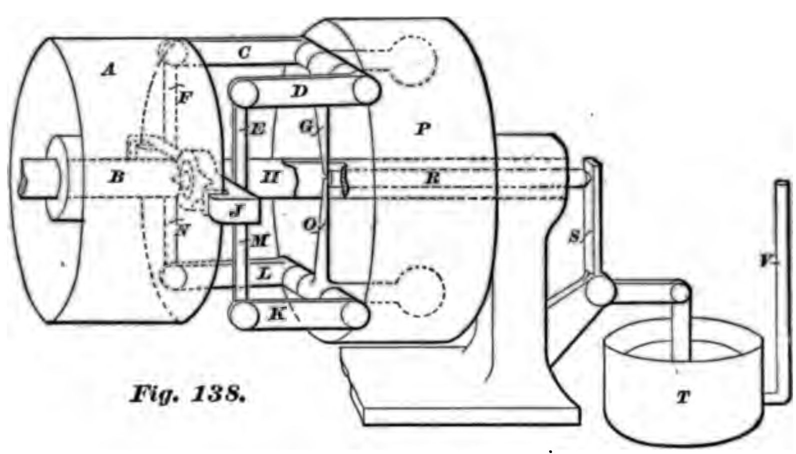


Fig. 138.

ing pen, and a record drum is mounted and driven either by clockwork or by the engine itself. In cases like those of electric street-car service very interesting and instructive records are thus obtained, showing the maximum power required and the fluctuations of power which occur in such service. The area of the record between the zero line and the line drawn by the recording pen is proportional to the whole work done in the time during which the record is made. This area can be very accurately and quickly obtained by cutting it out and comparing its weight with that of the same length of the whole breadth of the paper ribbon. A better method of mounting the dynamometer as an energy meter is to mount a light counter on the float and allow one wheel of the counter to be driven by contact with the face of a disk which is driven by the engine.

The friction wheel of the counter should be on the centre of the disk when no work is being transmitted. In the designs of the dynamometers built for use in the shops of the Rose Polytechnic Institute both the recording pen and the integrating apparatus have been included, but the integrator has not yet been attached to any of them. Arrangement can very easily be made in this particular form for checking the zero, as the thrust of R can be easily taken up by a collar, leaving S free and thus bringing the pen and friction wheel to zero.

When the dynamometer is used to measure the work given to any particular machine, P becomes the pulley driven by the belt, and A the pulley of the machine. The cross-bar J is clamped to the pulley, and the action is the same as that just described.

II. Another form of dynamometer which has been used with good results is illustrated diagrammatically in Fig. 139. In this case let A be the driving and B the driven pulley. Two pulleys, C and D, are mounted on a frame E, and properly supported, so that they can be pulled together on the belt. The frame E is supported by means of a plate F resting on a diaphragm as shown. The whole weight of the frame together with the weight of the deflected belt should, however, be independently supported, so that a change of length of the belt while running does not affect the indications. The water under the diaphragm transmits any change of pressure on the plate F to the pressure gauge, which, as in the case above described, may be of any ordinary form if a record is not required. When a record is required a similar method to that described in connection with Fig. 138 may be adopted, and a mercury column gauge is actually used on the instrument under consideration. The lower belt being supposed the driver, it becomes tighter and the upper slacker as the power increases; thus greater pressure is thrown on the plate F. This, however, does not sensibly change the positions of the pulleys or the deflections of the belt, which must be stretched sufficiently to prevent the upper half of the belt becoming slack when the maximum power is being transmitted. This apparatus may be standardized statically by placing a weight on the belt above the pulley D, but when possible it is better to standardize it by causing the machine to do a known or measured amount of work. In many cases this can be readily done by means of a brake. In the case of electrical achinery a definite increase of work may be obtained and easured electrically.

A part of a diagram of work taken from a test of the Terre Haute Electrical Street Railway is given in Fig. 140. This diagram is interesting as showing the extremely rapid variation of the power which the engine was called upon to deliver, and the promptitude with which the call was responded to by the belt. Variation of speed is not taken account of here, as no separate time marker was placed on the apparatus. The time scale marked on the diagram is therefore to be taken as simply the average speed of the paper. The diagram here shown is

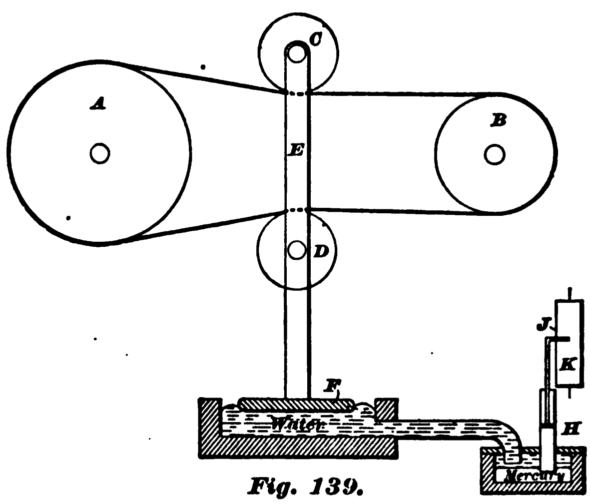


Fig. 139.

similar to the remaining portion, which contains the record of a complete day's work and is therefore of considerable length.

III. Perhaps the most interesting of the dynamometers here described is that the principle of the arrangement of which is illustrated in Fig. 141. In this figure B represents the belt which is supposed to be driving the pulley A. Bearing on the belt and just clear of the pulley two measuring wheels C and D are fixed. These measuring wheels communicate their motion to two counting wheels H and J, one of which, H, carries a dial, while the other carries a pointer K. Both the dial and the pointer are turned in the same direction, so that if both pulleys C and D make the same number of turns, the pointer K and the dial H turn at the same rate. When the belt is called on to transmit work, however, the tight side of the belt runs faster than the slack side on account of its smaller section. What is commonly known as the elastic creep of the belt is increased, and hence one of the pulleys makes a greater number of turns per minute than the other. Either the dial or the pointer goes ahead, and the rate of gain is proportional to the rate of working, while the total gain in any time is proportional to the total work done in that time. This dynamometer, as it has been made for laboratory use, has provision for adapting the distance between the pulleys C and D so as to fit any size of pulley A up to six feet diameter. The whole is mounted on a shaft, which, when the machine is in use, forms a continuation of the driven shaft, while FC and GD are carried by arms which allow the counting wheels to be placed at different angles apart on the driven pulley. The object of this is to enable the gradual increase of speed of the belt as it passes from the slack to the tight side to be illustrated. In this particular dynamometer there is a double worm and gear reduction between C and J, and between G and H, which causes 2,500 turns of the counting wheels to give one turn of the dial and pointer. One per cent. of creep thus causes the relative motion of the dial and index to be one turn for 250,000 turns of the wheels C and D. With wheels 1 foot in circumference, and a belt speed of 5,000 feet per minute, which is nearly the actual case for one of our dynamos, we get one turn of the index on the dial in 50 minutes. Full

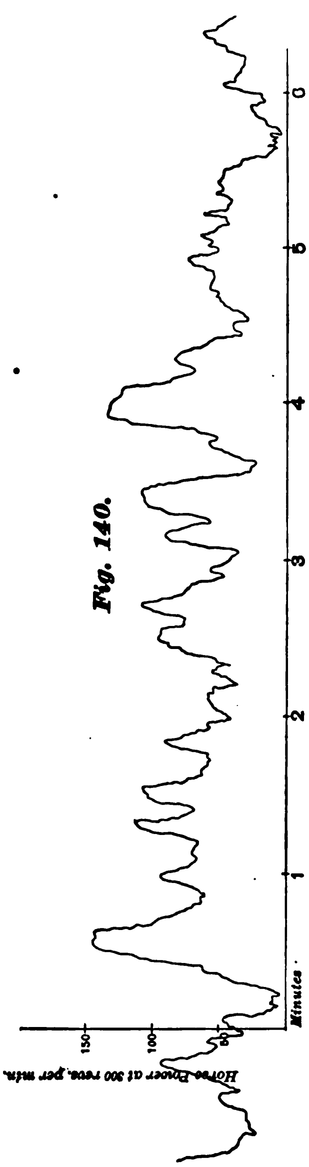


Fig. 140.

power gives considerably more than this, and therefore for continuous working the speed of indication is fast enough. For a power indicator with slow-speed belts the reduction is too great, as in this case it should be possible to measure the work done in one minute accurately. For such purposes it is intended to put on a steady deflection indicator worked by the counting wheels. Suppose, for example, an axis driven by CF and carrying a wheel turning with the shaft, but free to move sideways. Let this wheel run in contact with the face of a disk driven by DG.

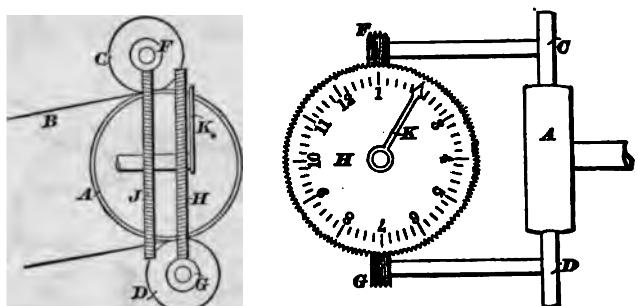


Fig. 141.

The wheel will move to the position on the disk where there is no slip, and it can therefore be made to turn a pointer and indicate the rate of working at any instant.

No attempt has been made to give details of construction. The object has been simply to point out the principles on which these dynamometers act; any one wishing to use similar devices will readily supply the details of construction suitable for his particular case. I have to express my obligation to my colleague, Prof. Ames, without whose assistance in preparing the diagrams I could not have found time to present this paper.

CCCCXCIV.*

MACHINE MOLDING.

BY HARRIS TABOR, ELIZABETH, N. J.
(Member of the Society.)

Of all the mechanical arts, that of molding has been the most difficult to formulate and to reduce to a system. Since the origin of metal-founding the molder has been pleased to shroud his methods in certain mysteries, which, to him at least, seem essential to perfect castings. It may be said of this trade, more than any other, that the traditions of generations cling to it. Like the good housewife of the olden time whose bread was often sweet and delicious and occasionally intolerable, the man of rammer and trowel will alternately score success and failure under apparently the same conditions. He can always tell why his casting is good, but can rarely give a reason when it is bad. There is much which can be accounted for in this; perhaps more that cannot be. In all other industrial branches the senses of touch and sight are always at the command of judgment. In the machine shop, contact between the workman and his work is always possible; an error may be detected as soon as made, and corrected at once; there are no final chances upon which the success of the machinist's job depends. With the molder, it is different. The conditions which insure bad work, and cannot be anticipated, are numerous. There may have been a bar in the "cope" under enough tension to induce a "drop" when the additional "strain" of clamping was put on; the core, with which he had nothing to do beyond setting, may have been made with no reference to free "venting," and a "blow" follows pouring. His troubles do not end here; the melter may have been in a careless mood to the extent of dull iron, and a casting with "cold-shuts" is his reward; if his foreman make a wrong estimate on the amount of iron necessary to "pour" his mold, and give him too little, another loss will be charged

* Presented at the San Francisco Meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

to his account. There is much, beyond the control of the molder, in the art of metal-founding, which tends to make bad castings. His strongest influence upon the quality of his work lies in skill which cannot be verified by caliper, gauge, or rule.

The molder's art is in making the mold of the proper density. Drawing a pattern from the sand after it has been rammed, and mending a broken mold, are mechanical operations easily taught. It is not so with ramming. If a touch of genius enter into molding, it is shown in making the mold of such density that it will stand pouring without "straining" and be soft enough to prevent "blowing" and "scabbing," with a certainty that the sand will remain in place until the iron has solidified. This is the molder's skill which cannot be formulated and passed down to succeeding generations, in books. Ramming a mold from the top to produce a required density on the under side may seem a simple operation to the layman, but a trial will convince him of one difficulty in molding. After he has learned to ram a flask with a given depth of sand, should he try one half as deep, he will discover his previous experience has not made him an all-around molder. This point is well illustrated in many large foundries doing duplicate work, where unskilled labor has been taught to mold a single pattern. In such cases, it is an invariable rule that it is not safe to change a man's work without putting him through a system of training on the new pattern. The writer has been told by foremen in such foundries that men who did excellent work on patterns which they had been taught to mold, were worthless on any other. In these cases the difficulty is in ramming and pouring, principally in ramming.

There can be no gauge to determine the force of the rammer's blow. It is a question of experience supplemented by good judgment. Some men have a capacity for acquiring this skill beyond others, and they are slow to impart it. On this account we see a greater relative difference in the skill of molders than in any other trade, and it is for this reason that there are so few molders coming to take the place of the old school which is disappearing.

As a rule, the foundry has less support from the office than any other department. Its conveniences and comfort are the last to be considered. Nearly all members present can call to mind plenty of establishments where the machine shop is a model of excellence in the way of modern tools, is warmed to a comfortable temperature, and shows signs generally of a kind consideration for the workmen; while the foundry is the same old kind with which we are familiar, filled with broken flasks, with no system of heating, and ventilated only through broken windows. There are no follow-boards or match-plates (the initial steps to machine molding) to cheapen processes. Even the core-boxes are misfits, and the work of filing cores to fit prints is equal to the cost of making the mold. There is a general air around the place which seems to indicate that the machine shop comes first, and the foundry last, in the affection of the manager. The foundry deserves a better consideration. It is here that the first step in machine construction is taken. It is here the first money is made; in fact, in a majority of cases, the foundry is the telling factor in the financial success of the manufacturer of iron products.

There is no doubt but the difficulty in formulating the molder's practice has had much to do in withholding improvements from this department. Skepticism prevails, which it is hard to overcome except by actual proof. In other departments a machine or fixture which has proven valuable in one shop may be sold to another on the strength of its record. In the foundry, success must be shown in the individual case before consideration is given. This doubt is gradually giving way under fierce competition. One year of low-priced castings will do more to set an iron founder thinking, than a dozen years' experience with prices of his own making.

The development of machine molding has been gradual, covering a long period. The follow-board which covers, or shuts off, from the sand that portion of the pattern above the joint line, was probably the first change from the original method of molding in boxes. The match-plate, which is a plate fitted with pins and pinholes for the flask, with a portion of the pattern fitted thereon like a medallion, came next. This was a greater improvement, for it compelled the flasks to be interchangeable. Silhouette, or stripping-plates, followed with decided advantage. The stripping-plate, often called drop-plate, is a plate cut out to receive the outline of the pattern at the joint line; enough is added to the pattern to project through the plate to the pattern base. Like the match-plate it is fitted with pins and holes to receive the flask; it is also a molding table, or board on which the flasks are rammed. Originally, this plate was turned with the flask, and the pattern drawn through the plate by hand. Modifications of the stripping-plate are numerous, nearly all embodying a frame or table, with lever attachment for drawing the pattern without turning the flask. A good illustration of this type is the machine for molding pulleys.

The evolution of the power machine from the hand machine was natural. The deep-rooted prejudice on the part of the workman has had its effect on the development of so great an innovation in foundry methods, and its progress has been slow. There are, however, a number of excellent power machines on the market; operated, respectively, by belts and cams, hydraulic, pneumatic, and steam pressure. It is not the purpose of this paper to discuss the merits of these various machines, but to touch upon some of the difficulties in the way of introducing machine molding in the foundry and the many advantages resulting therefrom, and incidentally a form of steam-operated machine with which the writer is identified.

All ramming machines may be said to have platens, of which there are two types: rigid and flexible. The rigid platen is simply a block of sufficient size to cover the surface of sand in the flask; the flexible platen is one which yields to irregular depths of sand, and exerts a like pressure on all parts of the mold. Of the flexible platen there are two: (1) the water bag, which is a rectangular box with a rubber diaphragm for the bottom, filled with water; (2) a group of rammers, equal in size to the flask to be rammed, hung on equalizing levers so that each rammer is independent of its fellows. At first thought, the flexible platen would seem to be perfect. If sand, under pressure, flowed like water, and its required density over the pattern was the same as needed on the joints of the mold, nothing could be better. But these conditions do not exist. The friction of sand upon itself and upon the walls of the flask, makes it comparatively unresponsive when rammed by equal pressures; if we add the fact that the mold, to pour well, should be softer over the iron than at the joints, we see that uniform pressure on a mold falls short of the requirements; it is better, however, than the rigid platen and requires much less hand work in the way of ramming and tucking. As an evidence of the difficulty in making equal pressures suit all conditions, a case can be cited where we had a pattern projecting vertically about 6 inches in the sand; at this point the side of the flask, which was 8 inches deep, came within 2 inches of the pattern. The average pressure, per square inch, over the mold was 40 lbs.; the pressure put over the narrow belt of sand between the flask and pattern was 70 lbs. per square inch, leaving only 28 lbs. per square inch over the higher portion of pattern; yet, notwithstanding the deep sand had 2½ times the pressure exerted over the lesser depths of the mold, it was necessary to precede the work of the machine rammers at this point by hand tucking. This was an unusual case. It is true, in all cases, that more pressure is needed along the walls of the

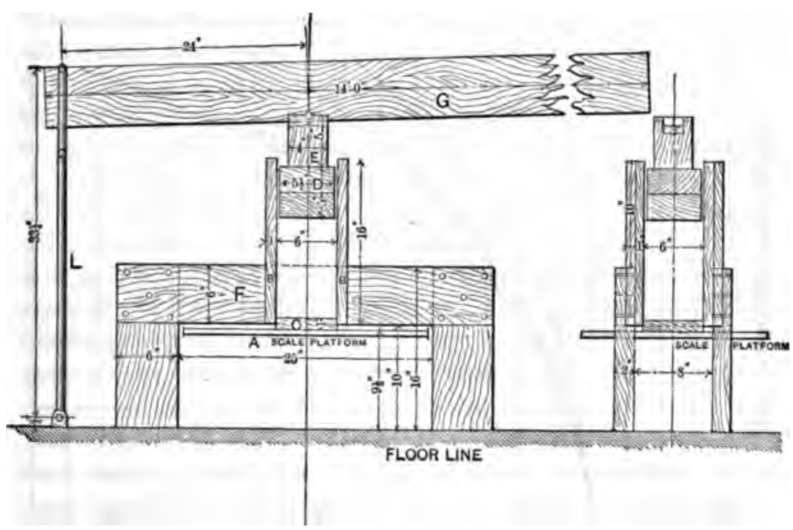


FIG. 142.

flask to overcome the friction of the sand; hence we have found it necessary to arrange our rammers to produce this result, giving the marginal rammers about 50% more pressure. This does not always give perfect ramming, and occasionally it is necessary to do some hand work, but as a rule this arrangement of rammers gives good results without hand manipulation.

In the spring of 1890, Mr. A. B. Moore, who was then a Stevens Institute senior, selected the rammer machine as the subject for his thesis. We discussed the lack of data bearing upon the friction of sand, and decided jointly to make experiments.

An ordinary platform-scale was used for weighing. A series of boxes, 4 × 4 inches, 5 × 5 inches, and 6 × 6 inches was decided on; these boxes were supported by frames spanning the scale and resting on the ground (Fig. 142); each box was fitted with a loose bottom which rested on the scale platform; the plunger used for ramming fitted its box loosely enough to avoid serious friction, and was connected to the weighted lever by a turned joint; the weight of the lever on the sand was found by weighing it in position. In all cases the scale was weighted to a pressure equal to 10 lbs. per square inch on the under face of the box. (This is about the density of the average mold surface.) We began with the

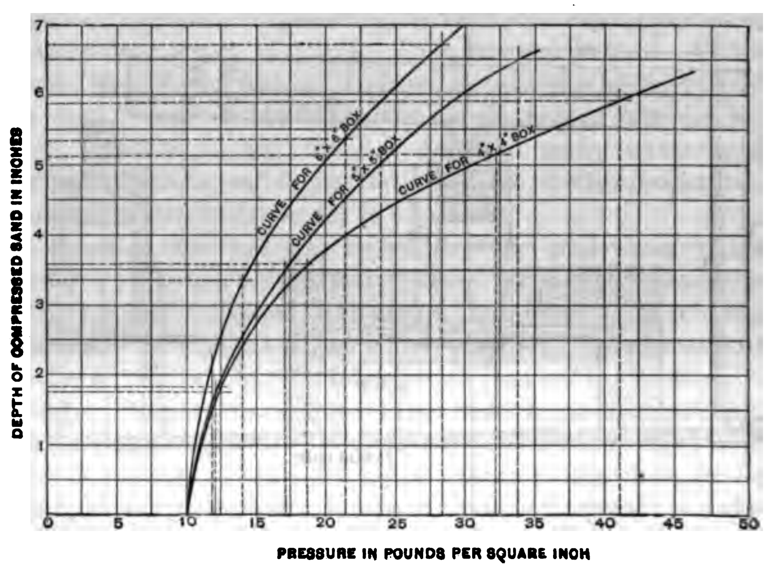


FIG. 143.

4 inches square box as follows: $2\frac{1}{2}$ inches of loose sand was put in and compressed until the scale-beam tipped; the intersection of the dotted line with 4 × 4-inch curve (Fig. 143) shows that the $2\frac{1}{2}$ inches of loose sand was compressed to $1\frac{11}{16}$ inches to give a density equal to 10 lbs. pressure on the under side, and it required a pressure of $12\frac{1}{2}$ lbs. on top of the sand to produce this result. With 5 inches of loose sand, $17\frac{1}{2}$ lbs. pressure was required on top to give 10 lbs. below; an addition of $2\frac{1}{2}$ inches in the depth of sand brought the ramming pressure up to 34 lbs., and the last $2\frac{1}{2}$ inches (making 10 inches) required a pressure of 42 lbs. to give 10 lbs. on the scales. With the 6-inch box only $11\frac{1}{2}$ lbs.

were needed to give 10 below, with 2½ inches of sand; with 10 inches, 26 lbs. raised the scale-beam, or 16 lbs. less than was required, under precisely the same conditions, with the 4-inch box. The walls of the boxes were of undressed plank, to represent the average condition of wood flasks. The friction on iron sides would have been less.

There are several methods of anticipating unequal depths of sand in machine molding. With the rammer system greater pressure may be given over portions of the mold which would otherwise be too soft. When flasks are of such size that bars are necessary the rammers are arranged to straddle them, thus doing away with all tendency of the bars to spring; this method also avoids the necessity of tucking under the bars. When the flat platen is used for ramming, sand may be scooped away from the higher portions of the pattern until the best result is obtained; where the flasks are not too large, sand will flow sufficiently to give excellent results when this plan is followed. With all our automatic machines we use the rigid platen for ramming; we make this of hard wood, which we cut out boldly over the pattern, and without much reference to the shape of the pattern; the amount cut from the ramming head is about 50% more than the displacement of pattern for ordinary cases. By this method no skill or judgment is required in putting sand in the flask, and the density of mold over the iron may be made to suit any condition. On small work, up to flasks 24 inches square, we get the best results from this system. We have a method of using flask-bars for ramming, which seems to suit certain conditions better than other methods; the bars are detached from the flask, and are made enough smaller so that they may be forced down without coming in contact with its walls; the flask and sand-box are filled with sand, and the bars forced down by a flat platen; the bars are deeper where the greatest ramming is required, and are made wedge-shaped at the bottom so that one bar will spread the sand until it meets the spreading influence of its neighbor. With this plan it is sometimes necessary to use a bottom board on the drag, to hold the bars when the drag is turned; the bars in the cope will hold in position against the pressure due to pouring.

An idea of the rammer connections may be obtained from the engraving showing details (Fig. 144); the rammers are hung to the ends of the cross-bars, which are grouped together by ball-

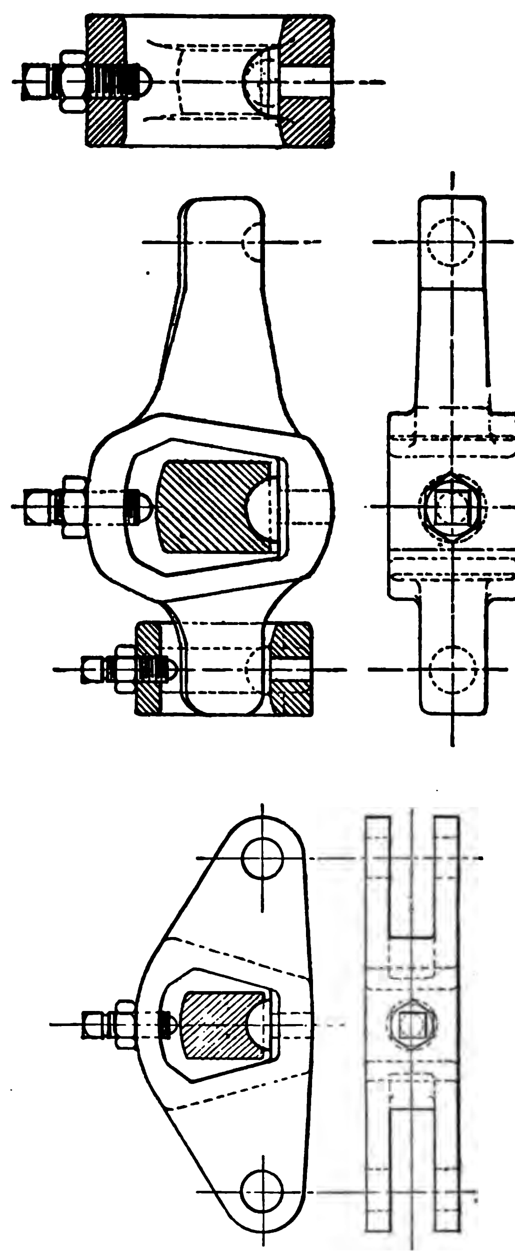


Fig. 144.

joints and operated centrally by a steam piston. The flasks are placed on trucks, which are topped with stripping plates and contain mechanism for drawing the pattern; the trucks are run under the machine for ramming, and withdrawn to take off the mold and replace the flask.

A description of such an automatic machine (taken mainly from

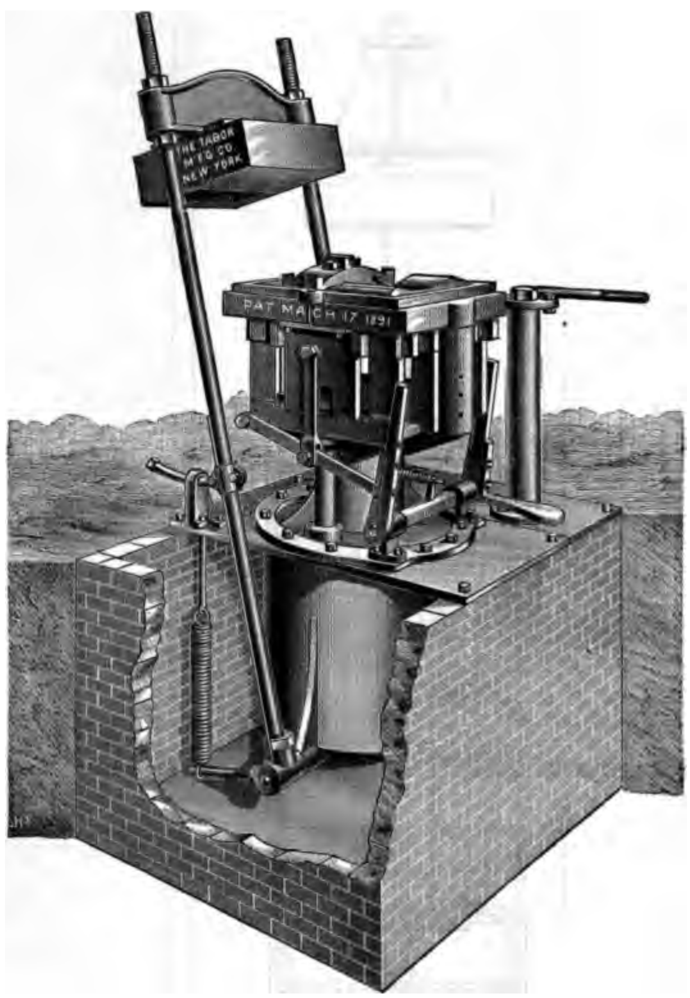


FIG. 145.

the columns of a technical journal *) may be conveniently introduced here to illustrate the operation of these principles:

Fig. 145 shows the floor broken to give a view of the machine below the floor line; and Figs. 146 and 147 show sections made

* *American Machinist*, of New York, October 22, 1891.

from working drawings. The piston takes steam on the under side only, its weight being sufficient to return it promptly after the mold is rammed.

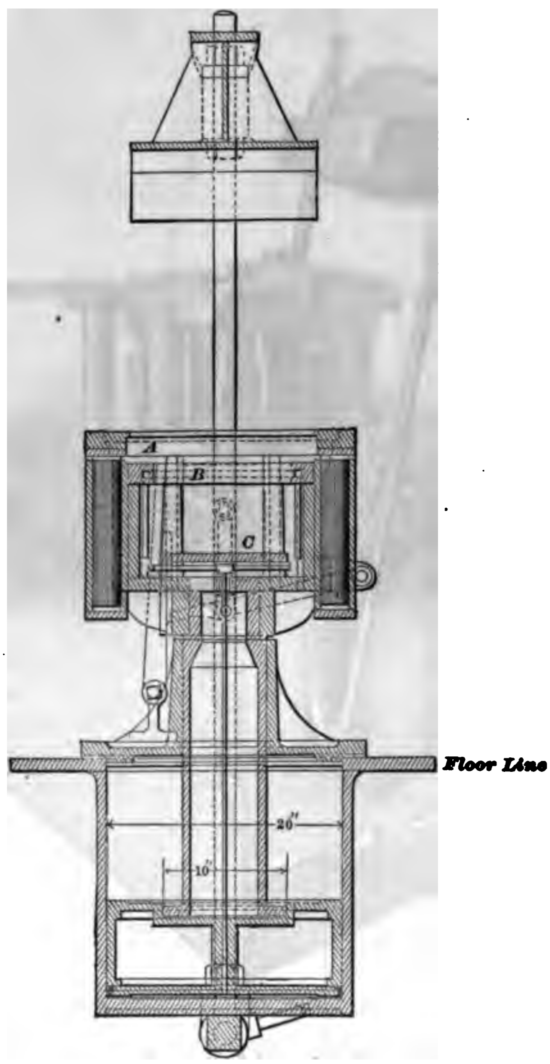


FIG. 146.

To the piston rod is attached the principal part of the mechanism, consisting of a table with lugs projecting upward, and supporting the pattern frame *B*, upon which rest the patterns; the stripping-plate frame *A* directly over the pattern frame, and resting on it, to which the stripping-plate is attached; the stool-plate *C* suspended to the stripping-plate frame, and

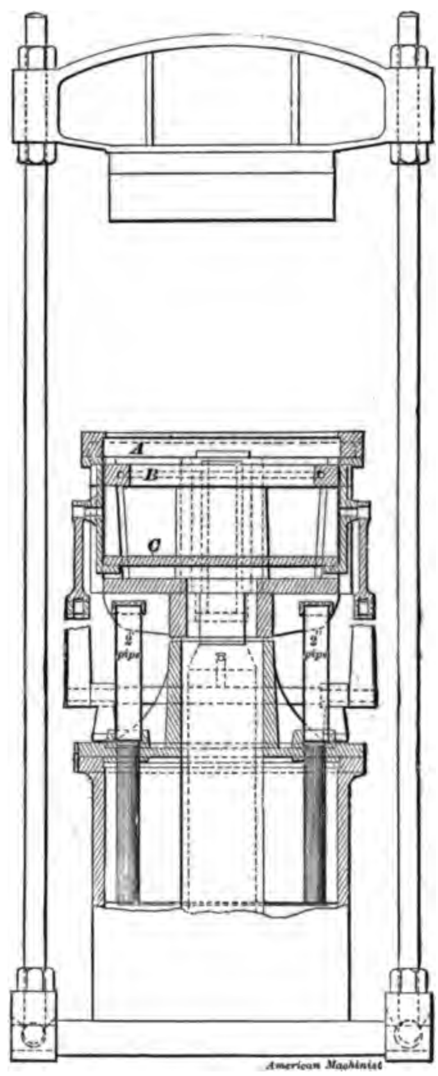


FIG. 147.

moving with it; side levers and tumbling shaft for tripping after the pattern is drawn. The pattern frame has an annular passage which is connected to the cylinder by the small pipe shown, the object of this being to admit some steam to the pattern plate at each movement of the piston, this steam serving to keep the patterns moderately warm, preventing "sweating" or accumulation of moisture from the atmosphere, and making them draw from the sand more freely and smoothly. The stripping-plate frame is guided by two bored sockets, one at the front, and the other at the back of the machine, there being air-holes below the pistons, by which any desired amount of cushion can be obtained for the drop of the stripping-plate frame. The stool-plate is really part of the stripping-plate frame placed below the pattern frame, and its object is to support stools or internal parts of the stripping-plate used in holding green sand cores, or heavy bodies of hanging sand, while the pattern is being drawn. The side levers are pivoted at one end to the table, and are connected at the middle, by links, to the stripping-plate frame, the outer end being free. The tumbling or tripping shaft is in front of the machine, near the floor, and has arms projecting upward along the line of travel followed by the free ends of the side levers; on these arms are stops which engage with the free ends of the levers on the downward motion, to draw the pattern.

The ramming head (Fig. 148) is carried by the wrought rods seen at either side of the machine, these being attached to a horizontal shaft at the bottom of the cylinder, which allows them to be swung forward and back as shown, a spiral spring being used to counterbalance the weight. The ramming head is usually of wood, roughly cut out over the pattern, to avoid too hard ramming on the high places. This block may, of course, be readily changed to suit any flask within the capacity of the machine. The stops on the stripping-plate can be also changed to suit any pattern within the range of the machine. The steam pipe enters the cylinder at the bottom, and from the throttle valve to the cylinder serves also as an exhaust pipe, the throttle valve being a two-way cock by which steam is either admitted or exhausted from the cylinder.

The operation of the machine is very simple. The half flask is put on the stripping-plate, with the sand-box to hold the sand which is to be compressed, and both are filled with sand. The ramming head is then swung forward over the flask against stops which define its position, and the throttle valve opened. The upward motion of the piston and attached parts carries the flask and sand up to the ramming head, where it is rammed instantly, and upon the throttle-valve lever being moved again, steam is cut off, and at the same time exhausted, allowing the flask to descend; the stops then engaging the free ends of side levers, and arresting the downward motion of the stripping-plate at a point about midway; the pattern, continuing to descend,

FIG. 148.

is drawn from the mold, and when the piston has returned to its lowest position the sand is struck off the flask, which is then taken from the machine. As the man removes it he presses the tripping treadle with his foot to release the stripping-plate frame, which then falls to its proper position with respect to the pattern, and the machine is then ready for another mold.

Water or compressed air may be used instead of steam, if it is desirable, though it is believed that steam is preferable in most cases; because it is usually easily obtained without the use of other special auxiliary machinery of any kind; it is cheaper, and the steam appliances coming under the charge of the fireman or engineer, the foundry is relieved of the care of them.

Good molding machines in a foundry where there is a fair amount of duplicate work may be made as profitable as the turret lathe has been in the machine shop. It has been said of the turret lathe that its limit is set by the ingenuity of the tool maker. If pattern maker be substituted for tool maker, and a fair amount of "nerve" in foreman be added, the same may be said of the molding machine. Like the turret lathe, its best results are obtained when operated by unskilled labor, or men trained only in the use of the machine. When the foreman has a reasonable amount of patience at the start, coupled with a disposition to make good castings cheaply, there is no doubt of his success in machine molding. If these qualities are lacking, it is better to defer the introduction of machines until the foreman has been brought to see their advantages, or has been supplanted by a more progressive man. We have a case where the manager of a large iron foundry came to see a machine making castings similar to his own. He was especially interested when he saw a duplicate of one of his patterns on the machine, and tons of machine-made castings in the yard for his inspection. There was no doubt, in his mind, of the value of molding machines in his business, and he wanted to introduce them. The question was left to his foreman, who decided against the purchase after seeing a machine in operation, and admitting the work was much better and cheaper than he could produce by hand. Here was a case where prejudice, or fear of a change in methods, overruled judgment. In another case, parties who are engaged in a competitive business, noted for low-priced castings, sent their foreman on a similar mission; on his return he insisted that one be furnished him, and their order for a machine followed. We have one more instance on record of a foreman demanding machines. Such cases, however, are not the rule.

If a builder of machine tools sell a machine under a guarantee that it will produce given results, and the purchaser fail to get them, the maker can send a man to make good his claim if the machine is as advertised. This cannot be said of molding machines. There are so many conditions and operations in the process of making castings, independent of the mold, which have their influence, that the molding machine vender cannot verify his work in any foundry but his own, unless he have the hearty co-öperation of the foreman. The most perfect mold may be ruined in pouring; a defective core may produce a blow for which an innocent machine would get credit; a vicious kick given to a flask might cause a drop of sand, and a handful of dirt dropped in the pouring hole will spoil what otherwise would have been a perfect casting. These are conditions beyond the reach of the molding-machine maker; and, if they exist, and he cannot have the support of the foreman, his case in that particular foundry is hopeless. But when he can count on the same encouragement that is given to the machine-tool maker by the machine shop, his showing will be quite as good.

Consideration has not always influenced the introduction of molding machines. Unlike all other merchandise, they should not be sold by the purely commercial man. The seller must have a fair knowledge of molding and general foundry work if he would avoid serious trouble; he must understand first, and all the time, that it is better to lose a sale than to place a machine where conditions are against its success. In too many cases cumbersome power machines that are slow in operation and expensive to handle, have been recommended to do work that should have gone on a different machine or been left to the molder's rammer. There are plenty of foundries in the country making a specialty of light bench-work that could not afford to use our rammer machines; but in these foundries our automatics would make a wonderful reduction in the cost of molding.

It is a mistaken notion that stripping-plates and metal patterns must be used on power machines. In 1887 we sold a twenty-four-inch rammer machine to parties operating a steel foundry. Each year since they have ordered more machines, and now they have eight, varying in size from 18 × 18 to 44 × 44 inches. Much of the time these machines are run night and day. Not more than two, and probably only one, have ever had a stripping-plate. There is nothing special in their business to warrant expensive patterns. Their work is general, and usually made from patterns sent them. If an order come for twenty-five castings from one pattern, they at once scheme to get it on a machine, and generally with success. They keep on hand, boards fitted with pins and pin-holes to match their flasks, on which they dowel the pattern, each half going on a board.

These boards are placed on the machine truck, with the flask over them, filled with sand and rammed under the machine. The pattern, of course, is drawn by hand. Each machine is operated entirely by laborers; one, brighter than the others, has charge of the machine and draws the patterns. When the 44 × 44-inch machine was in the course of construction, the superintendent of this foundry happened to be in the shop. He saw this machine on the floor and said that he was just finishing an order for fifty castings, weighing 1,500 lbs. each, and that he would have made them all on this machine if it had been in his foundry. I have never seen an instance where machine molding is so generally used as in this case.

The all-absorbing question: What is the economy in machine molding?—is very difficult to answer. The product of machines will vary in different foundries, as much as the product of the molder. What may be called a fair day's work is an unsettled question. From the standpoint of manufacturer and workman it is too small in some localities, and too great in others. A machine that will mold 175 flasks, 16 × 16 × 10 inches deep, with two men to operate it, in one foundry, would, under precisely the same conditions, mold 250 in another. One manager may surround his machine with little conveniences for handling work, and thus increase his product, while another would compel the machine men to work under disadvantages. The treasurer and practical shop man of a foundry were observing the operation of an automatic machine, with watch in hand; a complete half mold in 16-inch nowel, or, as they would say out West, drag, 5 inches deep, had just been made and turned on the floor for inspection in ten seconds after the sand was put in the flask, when the treasurer asked the question: "How many molds can be made in a day?" Before the writer could reply the shop man said: "That is not the question; the question is, How many molds can we take care of?" A better answer could not have been given. The first machine of our automatic type has been in use about a year. The conditions are not favorable; all the sand is handled by shovels, and the flasks and molds are carried to and from the machine by hand. The flasks used in this case are 14 × 17 × 10 inches deep, and weigh 70 lbs. The sand in flask, when rammed, weighs 156 lbs. Two men on this machine make 200 molds per day, and average during the working hours from 27 to 34 molds per hour. These men have made and carried away 158 nowels in one hour and thirty-five minutes, and have made 200 complete moulds, ready for clamping, in less than five hours. We must keep in mind that these two men must shovel into flasks over 31,000 lbs. of sand, and carry off the same amount, in making 200 molds; they must also handle twice 14,000 lbs. of iron in flasks. 200 molds, under these conditions, is too much for five hours' work. But this number is not too much for a day's work for two men. A greater product might be obtained from an additional man, or from a conveyor for elevating sand to a hopper over the machine. A system of handling the molds after they are made would also add to the machine's capacity.

DISCUSSION.

Mr. M. P. Higgins.—The importance of this subject is so great that I would be very glad to have some expression of ideas from the members of this Society. I think we will all admit that the importance of the foundry as a step in machine building is growing more and more apparent from year to year, and I doubt very much if it is keeping pace with other branches of machine production. I think the possibilities of the foundry have not received so much attention as other steps in building machinery; I doubt if it has had its due share of scientific investigation. There are many matters connected with the subject of this paper about which I would like to know.

For instance, we want to know, scientifically and accurately, what constitutes a good sand for foundry uses. We know that for a distance of 500 or 1,000 miles around Albany there is no sand quite equal to that which is obtained there. If it has not already been done, it is desirable that scientific men analyze the sand, chemically and physically, and let us know exactly what it is. If we can determine that, it seems to me that it will affect this question of machine molding, because it has been shown that extreme accuracy of ramming and condensation of the mold is necessary; but, of course, if we can use a kind of sand which is strong enough, and yet is porous enough, that extreme accuracy may not be required.

In regard to the possibilities in the direction of unusual and exact results in the foundry, I would like to mention what I have done and have seen. I have seen some remarkable pieces of foundry work made, where two pieces of cast-iron were to be fitted one to the other, when the fitting surfaces were to be irregular and complicated. The first casting was made and slightly finished by filing or scraping, and then by special skill and practice the second piece was poured upon the first piece, and a very excellent and satisfactory fitting made in that way. I have also in the same way seen pieces cast together for journals, the journal being turned in the lathe, with care and accuracy as to its shape, so as to account for contraction, and then after that surface is thus prepared, the external portion poured upon it, and a complete and almost perfect fit made for a journal which would serve admirably for a large pressure and slow motion. Some fifteen years ago I cast a nut for a screw for very hard service on to a steel screw—common United States standard thread, 2½ inches diameter—and it has stood very hard service for fifteen years, with a wearing quality to it that I doubt very much could be obtained by machine-shop fitting.

The nut was 6 inches long, through which the steel screw worked with almost a perfect fit. Owing to the glazed surface of the cast-iron, the friction is small and the wearing qualities great.

I was recently visiting at the foundry of the Deane Steam Pump Co., of Holyoke, Mass., and there I saw a method for venting cores which they claimed was new with them. It consisted in placing twine or strings throughout the mass of the core, the twine or string being covered thickly with wax or tallow, so that when the core was placed in the oven and dried, and the wax or tallow melted, the strings could be easily pulled out of the core, and the whole mass was left free to carry off the gas.

Mr. John Richards.—I am not a little astonished at the slow progress which this art seems to have made, considering that it is now more than twenty-five years since the Ransomes of Ipswich, England, began molding by machinery. Mr. Ransome came to Philadelphia about twenty-five years ago, and called on William Sellers & Co., of Philadelphia, to introduce his methods, which they refused to adopt. He returned to England unsuccessful, but Messrs. Sellers & Co. soon after made what they called their pulley machines for molding pulleys, which have been in successful use ever since. I need not describe these. They are well known. I am not a molder myself, but I have observed the process a good deal, and I think that these processes which we are trying to introduce in the way of molding machines are not new; neither have they ever failed. It is want of attention, as Mr. Moore says.

What I want to speak about more particularly is this: Some years ago, in this city, in making barrel cores for pipes, the core barrels were wetted up with clay and laid in a frame and started to revolve. The sand was dropped about twenty feet on the top side of the bars as they revolved. The impingement of the sand was regulated by its weight and the distance of falling, in exact imitation of what a core-maker would do. The bars were revolved by machinery, perhaps at a rate of 30 or 40 revolutions per minute, and when brought to size the core was shaved down by a knife at the side. I have seen cores from three to six inches diameter, eight or ten feet long, made in 30 seconds from the time the sand started to fall on them. The impingement of the sand in that case was perfectly regular and uniform from end to end, and the result seemed to be wholly satisfactory. In the Ransome works at Ipswich there was a saving of metal by the machine processes equal to 10%; also a premium in price paid for the castings so made.

Mr. D. G. Moore.—The gentleman speaks of molding-sand. I have had considerable experience with molding-sand. I have found that sand used by the molder, where the mold was put up by hand, would need to be vented, the vent wire being used very freely, as is the custom. The same sand used in a molding machine would make a perfect mold without vents of any kind. It goes to show that the molding machine does its work more perfectly or more practically than the molder. It is not possible for a molder to ram a mold equally all over: he will ram it hard in some places and soft in others; it just depends upon how the man feels. We all know there are certain days when molders do their work less well than others; it is generally pretty close to pay day, too. The molding machine never gets affected in that way. The argument only goes to prove that the sand is all right. It is surprising what a molding machine will do—the amount of work it will do, and the perfect work. I have seen three and four hundred castings made every day from one pattern, and put on a scale, and it would be almost impossible to detect the slightest difference in weight. (I don't mean by that that it would be weighed on a druggist's scales or a scale made to weigh gold dust, but on an ordinary scale, which would weigh very close, a scale that would weigh to a quarter of a pound accurately). Such a scale does not show the least particle of variation in the weight of the castings made by machine.

I did not come here to advertise any machine, but it may give my argument a little more weight when I say that the company of which I am vice-president does manufacture the particular machine of which Mr. Tabor is the patentee, and we have had a great deal of chance and opportunity to make fair and reliable tests. I suppose we have from one to a dozen visitors every day looking at those machines in practical operation. We run two ourselves, and I think they are the best investment we ever made. You can put any kind of work upon them that comes in, and it is a very mistaken idea, as the author of the paper says, that you must have specially prepared patterns. You can mold any pattern. Of course if the castings are to be duplicated many times, it is best to make stripping plates, but any pattern that can be drawn by hand it will pay to put on the molding machine. And your castings are perfect; they don't require any cleaning as compared to castings made by hand, and I don't think there is any one, if he would only try the molding machine, who would ever be without it. It seems strange that they are so hard to introduce. A man will buy all improved tools for his machine shop; he will buy special lathes and everything that is needed to cut down the cost of work, but about the molding machines he is skeptical. He seems to think that nothing can be made in a foundry except it is made by a molder in the old-fashioned way.

CCCCXCV.*

THE STEAM DISTRIBUTION IN A FORM OF SINGLE-ACTING COMPOUND ENGINE.

BY F. M. RITES, PITTSBURGH, PA.
(Member of the Society.)

THE mechanical performance of the single-acting engine has become well known, but the structural features are usually considered to the exclusion of the peculiar method of steam distribution which has been introduced in its compound type. Moreover, the treatment which this new method has received has been too superficial and disconnected to be acceptable as an analysis. This paper will therefore consider the engine as an economic study only, the steam distribution being entirely independent of single-action or other mechanical features, and will serve incidentally to satisfy a very natural curiosity regarding the remarkably uniform efficiency under extreme variations of conditions.

It is necessary to note briefly the character of previous work, to understand properly the nature of the departure made in this new type of compound. Broadly stated, all compound engines, with this single exception, may be considered under two heads: those having an intermediate receiver between the cylinders, each with a complete valve, or system of valves, interposed between it and the receiver; and those (known as the Woolf) having no such receiver, but so designed that a port shall periodically act as a passage between the cylinders.

All the energy of inventing engineers since the design of the first compound has been expended in the development of these two forms. The patent records are perhaps the best indication of the amount of thought given to the subject; and the almost unending claims, based on the arrangement of cylinders either in tandem or side by side, with cranks in line or at an angle, oscillating or otherwise, and with valves from one to eight in

* Presented at the San Francisco Meeting (1892) of the American Society of Mechanical Engineers, and forming part of Vol. XIII. of the *Transactions*.

number, are but variations upon the fundamental principles which distinguish one type from another. Very often a designer has felt justified in securing capital to promote an unimportant improvement or develop some ingenious though valueless mechanism. In other cases some established concern has found it necessary to build a line of compounds, without the ability to do them justice or in the least understanding the laws governing the efficiency of such steam distribution. That many of these forms can scarcely be considered improvements, and that others are positive failures, is but natural and to be expected; but the great majority give much better results than the same class of simple engines, assuming both under good conditions for the comparison.

Of the two types, the receiver is better known, although it is necessarily the more complicated. The general choice may be explained by the fact that each cylinder may be treated by the designer as an independent simple engine, with its own complete system of valves, and with the receiver acting as a secondary boiler.

While the Woolf type permits a simpler construction, the difficulties attending a proper steam distribution, especially under varying conditions, have prevented a general adoption. In fact, its occurrence is so rare as to be practically unknown.

To better appreciate the difference in the action of the steam in the two types, the following ideal comparative diagrams are given, with the distinguishing features in each noted (Figs. 149 and 150).

With the aid of these cards, let us trace the objections to each system as a preliminary to the reasoning which led to the correction of the faults in the new type.

In the Woolf engine the resultant enormous compression in the high-pressure cylinder with small clearance and early low-pressure cut-off, or the great loss through clearance with a moderate compression, and the inability of any construction to maintain its efficiency under even ordinary changes of conditions, has effectually prevented the development of this type.

It has been perfectly possible to design a special form of Woolf compound which should be highly economical; but the design invariably possessed no elasticity in its application, and rapidly lost in its efficiency with a change of load or pressure.

The receiver type, however, is much more flexible in this respect, and will allow more variation in construction and conditions of operation, without unduly increasing the rate of steam consumption, until the range of conditions becomes considerably greater. In all engines of this class, however, free

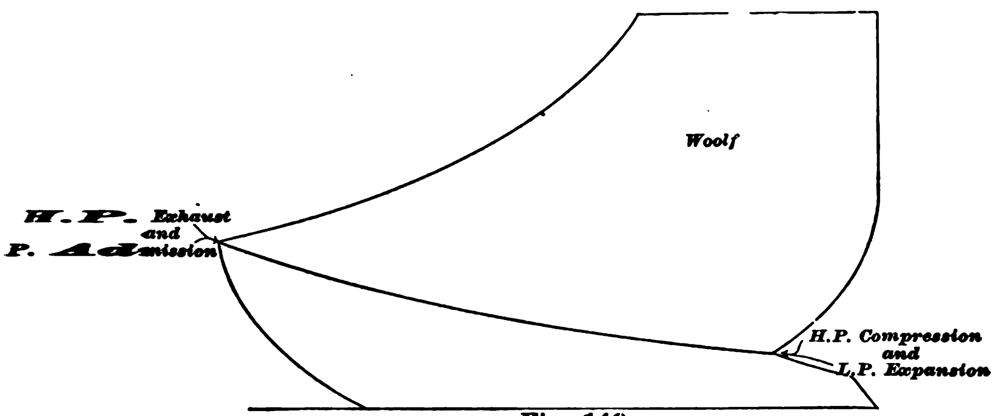


Fig. 149.

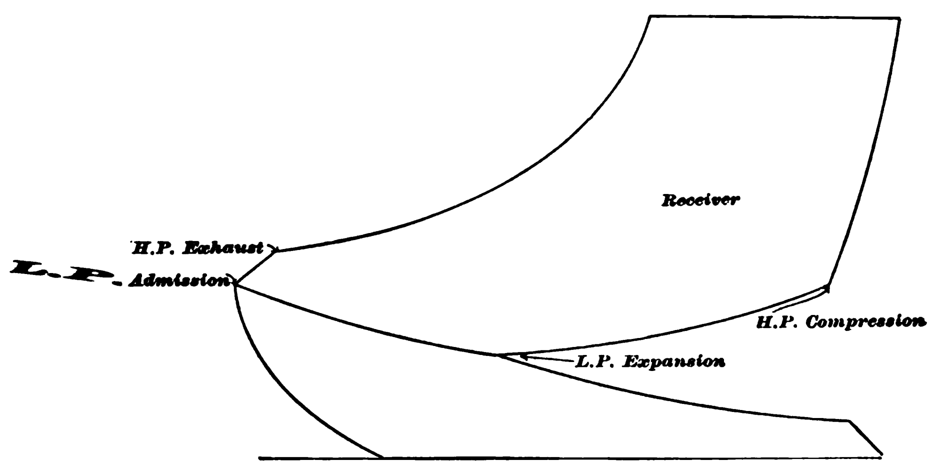


Fig. 150.

expansion into the receiver, as illustrated in the ideal card, means proportional loss in economy. The receiver is really a clearance space, causing a loss greater or less according to its capacity and to the rise in pressure during the exhaust of the high-pressure cylinder. The argument that the energy lost in this expansion is absorbed by the remaining steam in the receiver is in accordance with the conservation of energy, but it is wasted energy, as it is not afterward returned to the engine. Under constant conditions, the same amount of steam which leaves the high-pressure cylinder must enter the low-pressure cylinder; and if the initial pressure of the latter be less than the terminal of the former, there is a difference of potential energy in these two conditions that no subsequent manipulation can recover.

Another source of loss in the receiver is the change in pressure and temperature to which it is subjected.

There can be no doubt that the laws relating to condensation and retarded re-ëvaporation hold true with the receiver, as with the cylinders. An increase in the size of the receiver increases the area of condensation and radiation, so that, all other things being equal, the increasing consumption of steam must keep pace with the capacity of the receiver.

Recent practice seems to be in favor of the smaller receiver, so that the steam distribution more nearly approximates that of the Woolf type; but the margin of conditions under which the engine can be operated economically is reduced exactly in proportion as the high economy incident to the Woolf engine is approached in this manner.

For a considerable over-load or under-load, the engine must be redesigned with respect to its steam distribution, or operated at a much lower efficiency, as is the law in a greater degree with the Woolf engine.

Inasmuch as it is the exceptional case where an engine is exactly suited to its work, the average commercial performance is far from what can be obtained under test conditions.

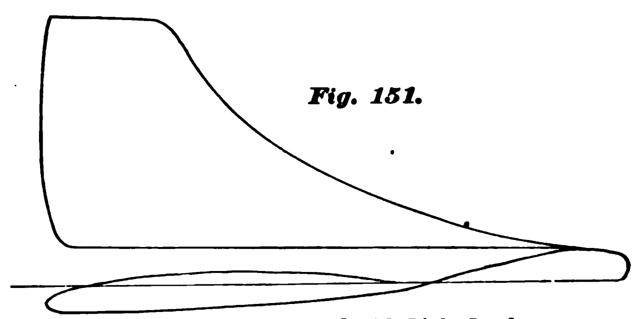
The evil is more marked with light than heavy loads; and in work demanding extreme variations of power, the high-pressure cylinder is frequently forced to supply all the power and in addition drag along with it the low-pressure piston, whose cylinder indicates negative work.

The indicator card (Fig. 151) will illustrate this action perfectly, and show that the proper division of temperature between the cylinders was possible only with the load for which the engine was intended.

As the whole range of temperature is covered in the high-pressure cylinder, the economy of the engine approaches that of a simple engine; but the net result is a considerable loss over what could be realized from the single cylinder alone, on account of the additional friction and enormous internal negative work of the low-pressure cylinder.

It has been but a short time since this question demanded consideration. Heretofore, compound engines have been built for particular kinds of work where load and steam pressure were maintained nearly uniform; while any considerable increase in the original plant was quickly followed by the installation of an engine of suitable power.

Since the rapid development of electrical industries, a demand has originated for an engine of high and comparatively constant economy under the widely different conditions incident to this service; and it was particularly true in the case of electric railways, where the average horse-power might correspond with



Receiver Compound with Light Load

the engine's rating, yet the actual working load would range far above or below this amount the greater part of the time. So great is this commercial loss with high-class compounds under such conditions, that it is still an open question with engineers whether simple engines in a sub-divided form and maintained at a uniform load, are not more economical than centralized power, in the form of a compound, subjected to such varying loads.

The objections which have been considered are general in their nature, and applicable to all compounds in degree according to the type they represent. They are wholly independent of the speed or any other distinguishing feature that characterizes a particular style of engine.

From the limitations thus described, it will appear that the problem to be solved was reduced to the following:

Given an extreme range of conditions as to load or steam pressure, either or both, to fluctuate together or apart, violently or with easy gradations, to construct an engine whose economical performance should be as good as though the engine were specially designed for a momentary condition—the adjustment to be complete and automatic.

Technically, this may be reduced to read: Design an engine compounded, whose individual cylinders shall be subjected to a constant range of temperature, under constant limiting pressures, with different loads, and whose governor shall automatically readjust the division of the total range of temperature with different pressures at either extreme of the cycle, and at the same time regulate the speed.

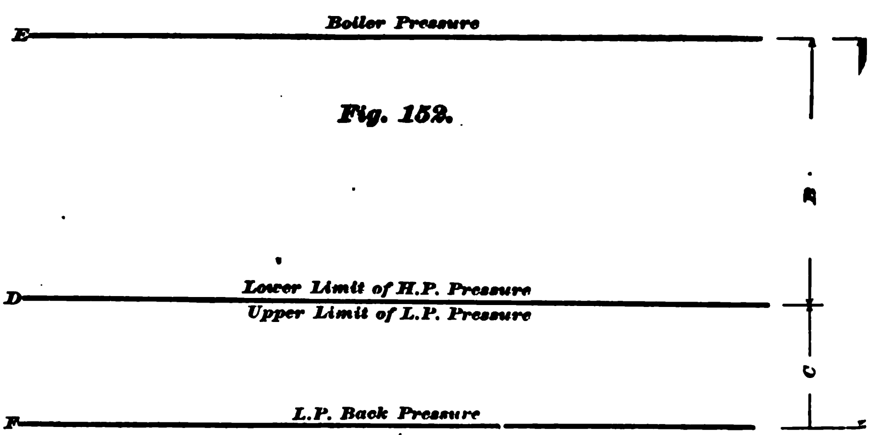


Fig. 152.

This sounds worse than the first presentation, but it may readily be explained graphically by the diagram above. (Fig. 152.)

A represents total range of temperature.

B represents range of temperature in high-pressure cylinder.

C represents range of temperature in low-pressure cylinder.

The total range of temperature due to fall of pressure and represented by the distance (A) must at all times, whatever the conditions, be automatically divided between the cylinders on the intermediate line (D) in the predetermined proportions represented by the distances (B) and (C), which are nearly inversely proportional to the surfaces subjected to the partial ranges of temperature.

The element of time, however, introduces a factor having no connection with the speed of the engine, but with a direct reference to the relative rapidity of the fall and recovery of temperature in each cylinder.

More properly, the line (D) should be located in accordance with the expression

$$\frac{AX}{\Sigma \tan x\, dx},$$

where (a) is practically constant, and includes in its composition the ratio of the exposed surfaces; while Σ is the sign of summation and $\frac{\Sigma \tan x\, dx}{x}$ represents the rapidity of recovery of the change of temperature.

With our modern high boiler-pressures, neglect of the proper correction for the difference in the rate of change of temperature becomes quite noticeable in its effect on the economy of the engine. Thus corrected, however, the line (D) falls very slightly in the early cut-offs and rises in the later ones, and except for this slight amount its position must be unchangeable with the load; but with a change of pressure and consequent temperature at either extreme, it must as quickly move to a new position, still maintaining its proportional distances (B) and (C) of the whole distance (A). For instance, in case of a failure of vacuum, the line will rise, or with a fall of boiler-pressure it will likewise fall, always, however, maintaining the same relative position between the lines (E) and (F) under constant load.

It may be readily believed that the fulfilment of this self-imposed contract was not an easy matter, so that the successive steps in their order of development may be interesting.

At first, expansion curves alone were considered and discussed, with, however, no idea of the development of a distinct type of engine. In fact, quite contrary to the usual plan, the form of the engine was not determined until after the steam distribution was developed. The objections to the older forms were recognized; but for a long time no means were discovered to correct them. Either seemed to require such a complicated system of valve gear to obtain a proper distribution of steam, that it became very doubtful if a commercially better article than already existed could be worked out. The Woolf engine seemed unsuitable, for reasons already explained, although long consideration was given it. In the receiver engine, the little line which marks the fall of pressure between the cylinders was a stumbling block that obstinately intruded itself.

After several months of investigation, the discovery was made of the peculiar value of a receiver of predetermined volume which should act as a clearance chamber for compression in the high-pressure cylinder. This can better be understood by reference to the fragmentary indicator card (Fig. 153).

It may be stated in general that an engine may be designed that will cut off at any convenient point (A) in the high-pressure cylinder, and through process of expansion reach the point (B), the cut-off in the low-pressure cylinder, from which point the

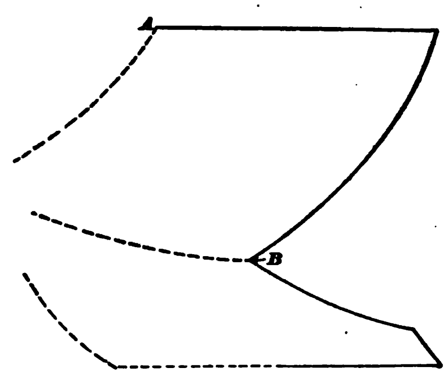


Fig. 153.

steam can be compressed to boiler pressure in the high-pressure cylinder, and expanded to a proper terminal in the low-pressure. We have so many unknown quantities, that we are at liberty to vary any one or several at will, without altering the results. For instance, if the clearance which allows a proper compression in the high-pressure cylinder be changed, we are at liberty to change the point of cut-off in the low-pressure cylinder so that the same degree of compression is maintained. If, however, it be stipulated that the cut-off in each cylinder shall be equal, then the high-pressure clearance can no longer be altered, but must be a certain definite quantity.

If, again, the distribution be effected by a single valve, driven by a shaft governor, the cut-off in each cylinder must be equally variable, and the point (B) will follow a path according to the law of its production, and dependent on the proportionate parts of the engine.

The particular discovery that proved so valuable was, that under these last conditions, the point (B) (Fig. 154) followed the path practically of a hyperbola, and that with a proportional clearance equal to the ratio of the cylinders, this path might be made to coincide with the hyperbola of the compression curve.

A variation in the cut-off of the high-pressure cylinder varies the point (B) to or from the atmospheric line: while a variation of the low-pressure cut-off changes its position in a direction parallel to this line. A simultaneous variation of both cut-offs makes the resultant movement of the point upon which so much depends, and which is determined by the clearance of the high-pressure cylinder.

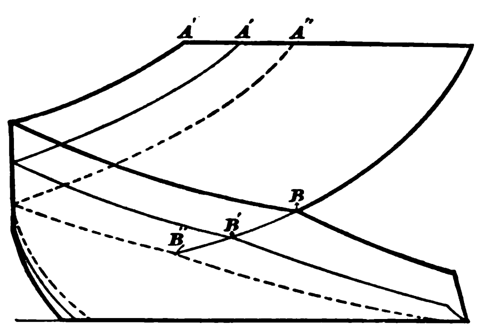


Fig. 154.

With an independent valve for each cylinder, it would be easy with any convenient clearance to so vary the cut-off in the low-pressure cylinder, that the variation of pressure at this point, due to varying cut-offs in the high-pressure cylinder, should cause the point (B) to fall on any high-pressure compression curve that should be determined on; but with a single eccentric valve-gear these points of cut-off must always be equal and similarly variable, so that the discovery was really essential to the simplicity that was attained.

It seems a lucky chance, although it is really part of the law which controls the movement of this point, that, as the pressure changes and the high-pressure compression varies, the point (B) leaves the path to which it has so closely adhered, and adopts a new one—still a hyperbola—which cuts the steam line at the same point of admission (Fig. 155).

So far, all considerations have been with the seemingly eccentric behavior of the point (B), and the reduction of its locus to a law which has been turned to important practical use, while the results have been obtained with a form of valve which has but one lip between the two cylinders.

In the course of the investigations, the proper amount of high-pressure clearance to suit the various points of cut-off was graphically determined, and a rather curious result obtained at the limiting full stroke and zero cut-offs. With steam carried full stroke in both cylinders, the high-pressure clearance must obviously be zero, to compress to boiler-pressure in the high-pressure cylinder; while with zero cut-offs in both cylinders,

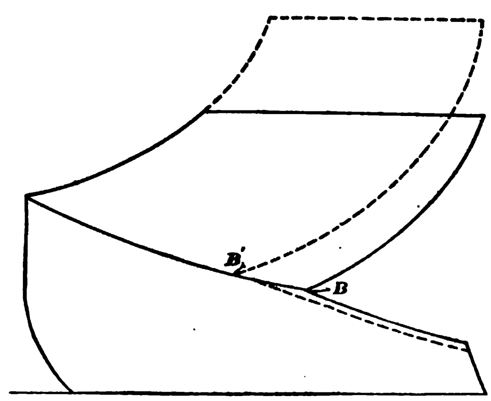


Fig. 155.

the clearance may be anything to accomplish this result. These unsatisfactory and indeterminate volumes were, however, caused by the error of considering conditions that are impossible in practice. Any intermediate position of the cut-off gives a positive value to the clearance chamber, varying but slightly (Fig. 156).

The average cut-off of one-half declares that the amount should be such a percentage of the high-pressure cylinder as the high-pressure cylinder is of the low-pressure cylinder; and at cut-offs on each side of this point to the limiting practical positions, the clearance volume departs so little from this amount that the error can be corrected by the very slight shifting of the valve through a coincident varying angularity of the eccentric rod. That is, under shorter cut-offs, a little greater clearance is required, and on longer cut-offs a trifle less is necessary; and to balance this error the angularity of the eccentric rod can be made to introduce a counteracting error, by which the high-pressure cut-off changes more rapidly than that of the low-pressure. In practice, the error measured in pounds of steam consumed is so slight as to be unnoticeable; and, in fact, for other reasons which shall appear, these varying dependent cut-offs are caused to depart from one another in exactly the opposite direction.

Although the compression in the high-pressure clearance space to initial pressure is theoretically correct and advisable, yet certain narrow departures from this degree may be allowed without materially altering the economy, if by means of this a

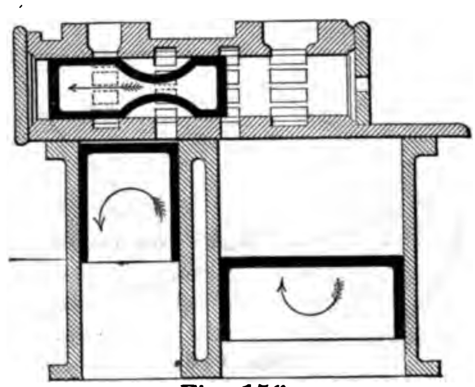


Fig. 156.

much greater gain can be made at some other point in the steam distribution.

If the distribution of steam were entirely dependent on the truth of the point (B) to the path in which we have constrained it to move, then the line (D) (Fig. 152) would in reality fall slightly with shorter cut-offs and rise again with the longer ones. Fortunately we have still the low-pressure compression with which to make a fit finish to the usually tiresome work of correction.

Under the influence of a single-valve variable cut-off, the low-pressure compression rises from a constant pressure, but with a period varying almost as rapidly as the cut-off.

The benefit in this case of such a peculiarity is in early exhaust closure with the lighter loads to prevent undue radiation to the condenser, and to raise the temperature of the cylinder so that there will be no surface condensation when the steam port reopens.

In the later cut-offs, the water of condensation is reëvaporated to a great extent before the point (*B*) is reached, and a considerable portion of the original steam is recovered for compression in the high-pressure cylinder; but, in the earlier cut-offs, reëvaporation does not begin until the high-pressure cylinder is shut off from the low-pressure, so that it is important there shall be little or no secondary initial condensation.

The amount of heat radiated during exhaust is dependent somewhat on the speed of the engine, and nearly proportional to the time the exhaust port is uncovered; so that an early compression, with a short cut-off, is valuable in reducing the total radiation to the condenser, thereby making the percentage

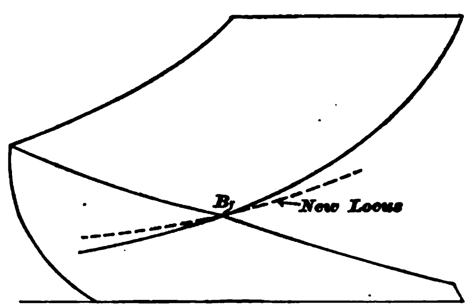


Fig. 157.

of loss more nearly constant, as well as reheating the cylinder to prevent initial condensation.

To the very natural question as to the reason for neglecting the radiation in the longer cut-offs, the answer is that the gain in power sinks the percentage of loss from this cause to such a small amount that the engine really effects a net gain by ignoring that factor.

Moreover, the variable low-pressure compression acts as a corrective influence over the point (*B*), which if left to follow the course subject to the equally variable cut-offs in the two cylinders alone, would too rapidly lower the line (*D*) (Fig. 152).

The variable low-pressure compression would however, overdo the work we have indicated, were the angularity of eccentric rod not interposed as an anti-corrective measure, and the progression, of the low-pressure compression retarded, while the point (*B*) thrown slightly from the path to which we have assigned it, varies the high-pressure compression a little on each side of the constant curve of its true path (Fig. 157).

The diagrams (Figs. 158, 159, 160), reproduced from actual cards, and representing light, medium, and full loads, respectively, illustrate the practically constant division of temperature between the cylinders, and are a graphic proof of the successful application of the new method.

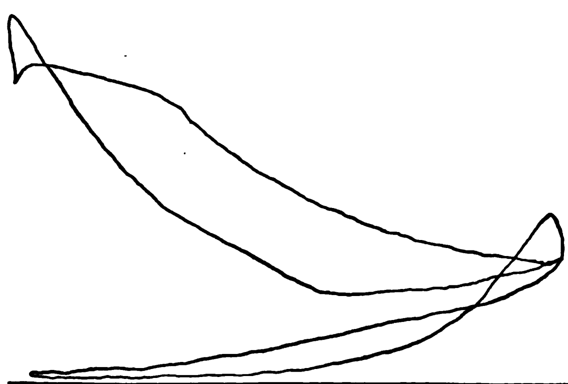


Fig. 158.

The practical results in fuel consumption are represented by the tabulated water rates under different conditions, both condensing and non-condensing.

WATER RATES, BY TEST, UNDER VARYING LOADS.

Horse-power.	210	170	140	115	100	80	50
Non-condensing.......... ...	22.6	21.9	22.2	22.2	22.4	24.6	28.8
Condensing	18.4	18.1	18.2	18.2	18.3	18.3	20.4

It would be insufficient to dismiss with so brief a mention the subject of radiation to the condenser, and the influence of the variable low-pressure compression as a partial correction for this loss and a factor in the development of the constant economy under varying conditions that has been accomplished.

A properly variable low-pressure compression may be regarded only secondary in importance to the practically constant high-pressure compression; and as this design has undoubtedly originated the principle of overcoming by a proper compression the otherwise great waste of a large high-pressure clearance chamber, so it is also believed to be the first to practically acknowledge and control the heat of radiation during low-pressure exhaust by a properly variable low-pressure compression.

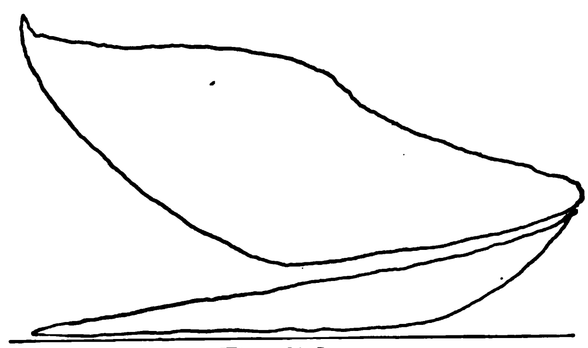


Fig. 159.

After the detailed explanations it will be interesting to note, as an evidence of the remarkable flexibility of the practical engine, that the valve adjustment may be set so far from true that the shape of the indicator cards is considerably altered without materially affecting the economy of the engine.

The explanation is that the line (*D*) does not depart from its

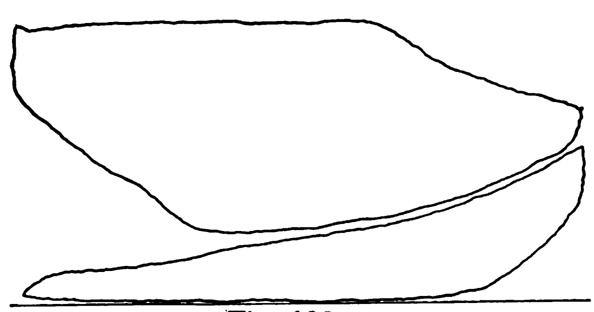


Fig. 160.

position unless the valve error be excessive, because the error introduced by such readjustment on one cylinder is in a measure corrected by a balancing error on the other cylinder. It must be understood that the whole economical result of the engine is directly dependent on the path of the point (*B*); and that, in the thrown slightly from the path to which we have assigned it, varies the high-pressure compression a little on each side of the constant curve of its true path (Fig. 157).

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CCCCXCVI.*

ON THE ELASTIC CURVE AND TREATMENT OF STRUCTURAL STEEL.

BY GUS. C. HENNING, NEW YORK CITY.
(Member of the Society.)

In the construction of the Henderson bridge, crossing the Ohio River at Henderson, Ky., U. S. A. (and of which the late F. W. Vaughan, C.E, was chief engineer), a considerable amount of medium and high structural steel was used for tension and compression members of the trusses. As the knowledge of such material was very limited at the time of the fabrication of this structure, in 1884–85, it was deemed advisable to investigate the steel in several ways, and to determine the state or condition in which the material was actually used in the bridge.

Theretofore all steel had been accepted after testing ¾-inch billets, rough from the rolls, which were supposed to give the properties of the material after rolling into a great variety of shapes. Previous experience had taught the writer that different rolled shapes give very varying properties to steel, and it was at his suggestion decided to test all the steel after rolling into shapes, as well as in the shape of ¾-inch billets at the steel works. The specifications had been written and the contract let before the writer became connected with the work, and hence he could not prescribe the manner of obtaining test pieces or method of making the tests, and it was assumed that the material filled the specification requirements if they were met, not by the rolled shapes in which the material was to be actually used in the bridge, but by the billet test.

The below results of tests will clearly show how erroneous this position is as regards uniformity of material as used in structures, which, after all, is the essential property which material used in one structure should possess.

* Presented at the San Francisco Meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

The specification requirements were as follows:

The steel shall be manufactured by the open-hearth process; Bessemer steel will not be accepted.

A small ingot shall be cast from every charge, and from this ingot a sample bar $\frac{3}{4}$ of an inch in diameter shall be rolled; if this bar fails to meet the requirements of the laboratory tests, the whole charge shall be rejected. The ingots of each cast and the blooms and finished pieces made therefrom shall each be marked with the number of the cast.

Steel used in *compression members*, bearing plates, pins, and rollers shall contain not less than $\frac{34}{100}$ nor more than $\frac{42}{100}$ of one per cent. of carbon, and less than $\frac{1}{10}$ of one per cent. of phosphorus. A sample test bar $\frac{3}{4}$ of an inch in diameter shall bend 180° around its own diameter without sign of crack or flaw. The same test bar in a lever machine shall show an elastic limit of not less than 50,000 lbs. and an ultimate strength of not less than 80,000 lbs. per square inch. It shall elongate at least 15% in a length of 8 inches before breaking, and shall have a reduced area of at least 30% at the point of fracture. It shall be incapable of tempering.

Steel for *rivets* and *eye-bars* shall contain not more than $\frac{26}{100}$ of one per cent. of carbon and less than $\frac{1}{10}$ of one per cent. of phosphorus. A sample bar $\frac{3}{4}$ of an inch in diameter shall bend 180° and be set back upon itself without showing crack or flaw; when tested in a lever machine, it shall have an elastic limit of not less than 40,000 lbs. and an ultimate strength of not less than 70,000 lbs. per square inch; it shall elongate at least 20% in a length of 8 inches, and shall show a reduction of at least 40% at the point of fracture.

It will be noticed that a maximum limit of 0.10% of P. was prescribed, and in all the analyses made and regularly reported to the inspector this limit was apparently never exceeded; but analyses of a number of these heats of steel showed that only one out of ten fell within this limit, while some of them exceeded it by 50%.

The only explanation of this discrepancy can be found in the excessive haste with which analyses are made in steel works laboratories, because there was never any attempt to furnish inferior steels; on the contrary, all possible safeguards were used to insure successful and satisfactory completion of the contract. After the steel had been accepted on billet tests at the steel works, the material was shipped to the rolling mills of another works and there rolled into the desired shapes.

Here, again, the material was examined by testing strips cut from finished shapes, and these in their natural state as they came from the rolls, and also after having been annealed.

The latter method was adopted because all of the tension members were to be annealed; and the compression members were to be treated similarly, if deemed advisable by the chief engineer.

As many of the plates and angles to be used in compression members were thin and wide, it was thought that annealing might become advisable to soften the material, which, during rolling, had perhaps become hardened to an undesirable degree. Experience obtained during fabrication, especially punching and shearing, demonstrated that annealing was really advisable, and therefore a number of finished chord sections—those which were built of thin shapes—were annealed, and without in the slightest degree injuring or distorting them or even loosening a single rivet.

It was noticed, however, that whenever there were open holes near the ends of angles, which were left for field connections and splice plates, the plates would become slightly longer than the angles; the difference in no case, however, amounted to $\frac{1}{32}$ inch, and by taking a very light cut over the ends the total lengths would again become true and correct.

In order to determine whether the shape became at all distorted, chalk centre lines and others near the edges were struck (using a fine sea-grass line), and after treatment and cooling these were invariably tested for accuracy without ever finding the slightest change or variation. Every rivet was also carefully tested at that time, and there never was a single loose one found in all the pieces so treated.

I desire to make this statement at the present time, because interested parties have repeatedly stated and do now assert, without ever having had an opportunity, or taken the pains, to examine the truthfulness of such statements, that annealing distorts members and loosens rivets. It is of such great interest and importance to all engineers and bridge builders to know that built sections—box shape with two or three webs—can be annealed without injury or distortion, that I cannot emphasize this fact with sufficient clearness or force in order to contradict statements based on opinions only, and not upon experience or observation.

It must be borne in mind, however, that a certain and absolute method of treatment was adopted, which prevented any possible or accidental injury or overheating, as described farther on.

At the same time I should like to call attention to the fact that all corrosion (oxidation) can be prevented absolutely by annealing in a sealed furnace charged with illuminating or other proper and convenient gas.

It is reserved for one of our progressive bridge builders to introduce a non-oxidizing, annealing furnace, and then use a high steel and anneal all members in case the material has been punched and not drilled.

Below are given the results of tests, tabulated in such a manner that a ready comparison of the properties of the material in its various shapes or conditions can be made at a glance. Before giving these, however, it is necessary to describe the shapes of test pieces, and the methods used in making the tests, as without such information the tables are incomplete and comparisons unsafe.

The steel was all made in open-hearth Pernot furnaces. The billets tested at the steel works were rolled in a billet train from the usual small test ingot, and it may be stated that in many instances the ¾ rods were covered with the peculiar red oxide usual when steel is rolled at a low temperature. The rods were more or less round—generally less—the ellipticity sometimes amounting to 0.03 inch in diameter.

These rods were tested in the condition in which they came from the rolls, in a Gill 100,000 lbs. machine, in a vertical position.

The elongation was invariably measured on 8 inches of length, and the distance between gripping wedges was generally about 12 inches. The modulus of elasticity was determined from the elongation due to a load of about 20,000 lbs. per square inch between 4,000 and 24,000 lbs. load per square inch, thus eliminating the initial error due to adjustment of instrument or straightening of rod in the wedges, which produces unavoidable error.

The measurements of elongation by micrometer were made by means of a Marshall double electric contact micrometer.

The yield point was observed by stretch measured by calipers and corroborated by drop of the beam.

Diameters before and after rupture were determined by a Richards micrometer.

Bending tests were always made, and quenched bending tests as well.

The loads were applied at a uniform rate, only stopping long enough at loads of 2,000 and 12,000 lbs. to take micrometer readings of elongation, and so as to produce about $2\frac{1}{2}\%$ stretch per minute.

The blooms for all bars and shapes were $7\frac{1}{2} \times 8\frac{1}{2}$ inches (except for 3 × 3 angles—for which they were smaller), and for plates they were slabs produced from the ingot by treatment under a steam hammer. As the plates were all to be rolled in a universal mill the slabs were generally but $1\frac{1}{2}$ inch narrower than the width of the finished plate.

All bars were groove rolled, different thicknesses being obtained by raising the rolls, all bars having the same uniform width of 7 inches.

From all rolled bars, plates and shapes, strips were cut of sufficient width to give, side by side, two specimens for test (three in the case of plates). All test pieces from bars were turned to 0.8 inch diameter for a length of 9 inches between shoulders, 4 inches long, with a fillet of about $\frac{1}{4}$ inch radius.

Tests were made the same as at steel works, except that a double electric contact micrometer* designed by the writer was used for all measurements within the elastic limit, measured for each increment of load of 1,000 lbs., and Brown & Sharpe calipers were used for measuring diameters.

An Olsen 50,000 lbs. vertical machine was used in making the tests, and the accuracy of the machine was determined by calibration, the results of which are given below, and which show that this machine was as accurate and sensitive as knife-edge machines usually are.

On the platform of the machine I-beams were placed so as to give ample bearing surface for the loads to be imposed.

Slabs of iron were weighed out on a standardized platform scale in 1,000 lb. lots, and these placed successively upon the machine. After weighing each such lot on the testing machine a

* Described in *Trans. A. S. M. E.* Vol. VI. pp. 479.

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Bending tests were always made, and quenched bending tests as well.

The loads were applied at a uniform rate, only stopping long enough at loads of 2,000 and 12,000 lbs. to take micrometer readings of elongation, and so as to produce about 2½% stretch per minute.

The blooms for all bars and shapes were 7½ × 8½ inches (except for 3 × 3 angles—for which they were smaller), and for plates they were slabs produced from the ingot by treatment under a steam hammer. As the plates were all to be rolled in a universal mill the slabs were generally but 1½ inch narrower than the width of the finished plate.

All bars were groove rolled, different thicknesses being obtained by raising the rolls, all bars having the same uniform width of 7 inches.

From all rolled bars, plates and shapes, strips were cut of sufficient width to give, side by side, two specimens for test (three in the case of plates). All test pieces from bars were turned to 0.8 inch diameter for a length of 9 inches between shoulders, 4 inches long, with a fillet of about ¼ inch radius.

Tests were made the same as at steel works, except that a double electric contact micrometer* designed by the writer was used for all measurements within the elastic limit, measured for each increment of load of 1,000 lbs., and Brown & Sharpe calipers were used for measuring diameters.

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small load was added and removed alternately until such increment or diminution of load produced $\frac{1}{8}$ inch motion of the beam, which was the accuracy usually observable or obtainable in careful operation.

Load.	Increment to Produce Motion.	Weight by Testing Machine.
Lbs.	Lbs.	Lbs.
4,878	1	4,878
6,080	....	6,082
8,005	....	8,010
10,005	5	10,017
12,005	....	12,019
14,022	....	14,032
16,042	10	16,040
18,072	....	18,100
20,004	...	20,000
22,002	....	22,010
24,000	15	24,120
26,004	....	26,090
28,004	....	28,100
30,010	20	30,130

The strips cut from plates and shapes were planed with straight parallel sides without shoulders, and their width was varied so that the testing machine was always used between about the same limits at which it had been standardized, so that the section was as near as possible $\frac{1}{2}$ square inch, the same as in test pieces cut from tension bars.

One of the test pieces was tested in its natural condition, while the other (two, in the case of plates) was tested after having been annealed. The results of all these tests of rolled shapes (pp. 586 to 617) have been plotted up to about the elastic limit in the accompanying plates (Figs. 161 to 165), on which full lines give curves of pieces tested in their natural condition; broken dotted lines, those of pieces annealed with full-sized bars or chord members in an annealing furnace free from flame or burning gases and properly sealed with clay; broken lines are those of curves of tests of pieces heated to a high heat (nearly yellow) in a coke-burning rivet furnace and allowed to cool off under ashes.

The method adopted for annealing full-size eye-bars and built members, and by which most test pieces from bars were treated, was the following: The furnace, about 55 feet long, was brought to a high heat by a soft coal fire at one end, the flames from which passed through channels and ports into the heating chamber, which had two compartments, separated by a brick wall, and each closed by a cast-iron door, which ports and doors could be sealed thoroughly to prevent entrance of gas or flame when desired, or cold air from the outside. Upon reaching a yellow heat the chamber door was opened after the ports had been closed, the pieces to be annealed were rolled in on cast-iron trucks, and the doors again closed and sealed with wet clay. After several hours' exposure to the radiated heat of the chamber walls the material reached a copper-color heat, and the trucks with the material were again withdrawn and allowed to cool. All fire being excluded from the chamber while the material was under treatment, there could be no overheating or burning, and all material was sure to be treated exactly alike. Before this method was adopted the most disastrous results had sometimes been obtained in more instances and places than one, and by several works and engineers annealing was declared impractical, unsatisfactory, and injurious to the material. It may here be recorded that one of our largest bridge works tried to anneal eye-bars by burying them in a great pile of wood and shavings, setting the whole afire, and keeping it burning for some time, after which the ashes were supposed to anneal the metal!

Another prominent works started a roaring soft-coal fire in a brick annealing furnace, and, after attaining a high heat, placing the material to be treated in it; then firing anew, so that the material was surrounded by flame, smoke, and unconsumed gases until it had reached an almost white heat, or at least a bright yellow, after which the fire was allowed to die out, and furnace material cooled off in three days' time until they could be handled. What would a metallurgist say about treatment of steel such as this?

To show any chemical change possibly produced by annealing steel, ten pieces, all treated at the same time, were annealed in a coke furnace, and were analyzed before and after treatment, with the following results; a partial analysis as given by the steel works is also given:

STEEL WORKS ANALYSIS. TENSION STEEL.

Heat No.	5,278	5,290	5,292	5,324	5,328	5,347	5,375	5,376	5,383
Carbon..........	0.20	0.23	0.25	0.25	0.26	0.23	0.23	0.22	0.23
Manganese......	0.63	0.57	0.74	0.67	....	0.57	0.83	0.79	0.74

Phosphorus—less than 0.10%.

CHEMICAL ANALYSIS OF UNANNEALED AND ANNEALED STEEL.

UNANNEALED.

Test No.	Heat No.	Si.	P.	Mn.	S.	C.	Cu.	Fe.
915	5,278	.018	.150	.696	.071	.190	trace	98.743
914	5,290	.015	.121	.621	.042	.190	trace	not det.
917	5,290	.011	.120	.712	.045	.200	none	98.887
918	5,292	.009	.126	.898	.046	.220	.004	98.999
916	5,324	trace	.116	.723	.056	.220	none	98.966
913	5,328	.006	.115	.800	.043	.210	.006	98.923
912	5,347	.006	.148	.672	.048	.220	.017	98.928
920	5,375	.011	.121	.857	.058	.190	trace	not det.
919	5,376	.015	.093	.775	.085	.190	.008	98.923
921	5,383	.032	.108	.849	.054	.210	.003	98.743

ANNEALED.

Test No.	Heat No.	Si.	P.	Mn.	S.	C.	Cu.	Fe.
....	5,278	.008	.149	.679	.064	.190	trace	not det.
....	5,290	.018	.124	.681	.047	.190	"	98.978
....	5,290	.019	.118	.674	041	.200	none	not det.
....	5,292	.014	.124	.889	.045	.220	trace	" "
....	5,324	trace	.118	.738	.050	.220	none	" "
....	5,328	.005	.114	.817	.048	.220	.004	98.854
....	5,347	.009	.150	.627	.036	.230	.014	not det.
....	5,375	.014	.122	.851	.057	.190	trace	" "
....	5,376	.009	.094	.787	.085	.190	"	" "
....	5,383	.034	.103	.780	.053	.210	"	" "

Signed, F. G. FRICKE, PH.D.

The samples obtained for analysis were taken with all possible precaution and out of the same hole in one end of each test piece. After annealing, all the rust in the hole previously drilled for sample was removed by following with a larger drill, and, this done, the hole was drilled deeper for another sample.

A careful comparison of these two sets of analyses among themselves and with the steel works analysis show nothing of importance except an amazing difference, which seems almost too much to be accounted for by supposing that such great differences exist in the several ingots of each heat.

However this may be, the analyses teach nothing, and do not explain the difference in the behavior of the material when tested before and after this treatment. The conclusion must necessarily be, that the reheating produces mechanical changes in structure of the steel, and this would account for the differences in the results of tests. Unfortunately, no determinations of specific gravity were made in any case; but it is altogether likely that these would have given some characteristic results, and it is probable that the specific gravity is less after reheating than before. This supposition would fully explain the observed differences.

The following tables of results of tests (pp. 586 to 590) are arranged to show the various properties of gradually increasing thicknesses of bars, each size grouped according to heat numbers. The billet test of each heat is given first; this is followed by test of specimen in condition as from the rolls, and the last the test of its companion piece after annealing.

Wherever it has been observed the elastic limit has been given as well as the yield point—the latter corresponding to drop of beam, the former determined by micrometer measurement verified by plotted curve.

The yield point as here given is what is commonly, though erroneously, called the elastic limit.

The last column in the tables is headed "Proportional Resilience," and is a factor obtained by multiplying tenacity by elongation per cent. in 8 inches. The factor given is proportional to the work done during rupture of a piece of metal, and is probably as good a factor for comparing materials as we can adopt. No other figure gives the relative life of metals as well as this does.

In this column two figures are sometimes given; the last of these (in brackets) is from a test of a $\frac{3}{4}$-inch billet rolled from a bloom, while the first is obtained from the test of the $\frac{3}{4}$-inch billet rolled from the test ingot.

The modulus of elasticity in the case of the billet tests was obtained by micrometer measurements as previously described. In all other cases it was determined directly from the curves obtained by plotting micrometer measurements of the elastic curve up to the true elastic limit; these curves are plotted so that the elongations measured are magnified 1,000 times, while loads are drawn 2,000 lbs. to the inch. On such scales the slightest errors and variations become apparent, and by multiplying the tangent of the angle made with the horizontal by 8 (inches of gauged length of test piece) we obtain the modulus of elasticity.

To do this some judgment must be exercised, as will be seen by an inspection of the curves; and in a few cases it is altogether impossible. Different observers might obtain slightly varying results, but those given in the tables are probably very nearly correct, and I must take all responsibility for them. The curves, with measurement and complete report of tests, are given in every case, and any one can verify the statements by his own investigation, if desirous.

This work has been so laborious and tedious that similar tests on angles are not given, but have been reserved for a future paper.

Of all of these tests of bars we find the treatment to produce the following results:

Elastic limit is raised in	33	cases	and	lowered in	8
Yield point is raised in	39	"	"	" "	5
Elongation per cent. is raised in	24	"	"	" "	15
Tenacity is raised in	17	"	"	" "	26
Modulus of elasticity is raised in	30	"	"	" "	8
Proportional resilience is raised in	27	"	"	" "	16

When from these tests we exclude some of the bars which had been treated in a coke furnace instead of the annealing furnace we find even a slightly better showing.

To look at these different changes numerically, we find that in the 33 cases of increased elastic limit the average increase was 4,570 lbs. per square inch, while the average decrease in the 8 cases was only 2,000 lbs. per square inch.

While the average increase of yield point of 39 tests was 3,100 lbs. per square inch, the average decrease in 5 was only 2,540 lbs. per square inch.

The average increase of elongation was in 24 tests 3.41%, while the average decrease in the 15 other tests was only 1.5%.

In the results of tenacity the reverse is true, as the average increase in 17 cases was but 1,972 lbs. per square inch, while the average decrease in 26 tests was 3,183 lbs. per square inch.

The average increase of modulus of elasticity in 30 tests was 991,200 lbs. per square inch, while the average decrease in 8 tests was 1,173,000 lbs. per square inch.

The proportional resilience was increased 198 in 27 cases, and was decreased 108 in 16 cases.

However this may be, the analyses teach nothing, and do not explain the difference in the behavior of the material when tested before and after this treatment. The conclusion must necessarily be, that the reheating produces mechanical changes in structure of the steel, and this would account for the differences in the results of tests. Unfortunately, no determinations of specific gravity were made in any case; but it is altogether likely that these would have given some characteristic results, and it is probable that the specific gravity is less after reheating than before. This supposition would fully explain the observed differences.

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The proportional resilience was increased 198 in 27 cases, and was decreased 108 in 16 cases.

All of these figures show the decidedly beneficial influence of proper reheating, and there is no doubt that with greater care and a better understanding much more uniform results could be obtained.

Another fact, which cannot be shown by the curves, but is nevertheless true, is this, " that permanent initial deflection, or permanent set of any structure, becomes less when the material has been reheated." There is but one indication of this in the curves, which is that those of treated material approach more nearly to a straight line than those of the untreated bars.

It is remarkable that there were not more cases in which the elongation was increased; in the latter, however, the increase as a rule is very much greater than the loss in the former.

It is, however, of the greatest importance for structural work that engineers should know that this proper treatment decidedly increases the stiffness within the elastic limit, and positively increases or improves the resilience or capacity for work of the material.

The curves and above tabulation show this, for a higher modulus of elasticity means greater rigidity or stiffness, and less deflection in structures ; and a greater proportional resilience means greater life and shock-resisting power. These are facts of the utmost practical importance, and nothing is of greater importance than more information on this point to still further corroborate it.

A careful examination of the securves seems to force several conclusions upon us, some of which will undoubtedly be very surprising to most readers. They are :

1. Proper annealing treatment is decidedly beneficial to this grade of steel.

2. Improper reheating or annealing in the presence of flame and gases is decidedly injurious to this grade of steel.

3. Proper treatment raises the true elastic limit materially.

4. Proper treatment raises the modulus of elasticity.

5. Proper treatment makes this steel more ductile.

6. Proper treatment increases per cent. of elongation of this steel.

7. Proper treatment decreases tenacity of this steel.

8. Proper treatment increases the proportional resilience, ductility, and capacity for work, and to resist shock.

9. Ordinary methods of testing are quite unsatisfactory to determine the most essential characteristics of materials.

10. Autographic registering devices, magnifying very considerably, would be of the utmost importance and value in the investigation of materials.

A further comparison of the billet tests with all of the others shows very plainly that they are by no means critical or definite in determining quality of material.

There does not seem to be any definite relation between these and the others made on specimens cut from the rolled bars, and all the tests here given, and innumerable others, still further prove that material should be invariably tested after having been rolled into finished shape. There seems to be but one reason why steel should be accepted on billet tests, and that is that it is in favor of the steel maker to do so, for it is well known that manipulation during heating and rolling to produce a $\frac{3}{4}$-inch rod does result, in the same steel, in greatly varying results. If, on the other hand, all the $\frac{3}{4}$-inch billets were reheated in a uniform manner before testing, then they would undoubtedly indicate more closely the true character of the metal, but still not that which it would have after proper rolling. Still such tests of reheated billets could be used as a correct guide to possible results obtainable by careful rolling. Repeated urging has, however, failed to induce any steel maker to introduce this method, although directly to his own interest.

Now let us examine the results of tests of plate specimens (pp. 610 to 617), bearing in mind that full lines on plate of curves (plate 165) represent tests of untreated metal; that broken lines show results of specimens treated in a coke-heated furnace to a yellow heat; and that broken dotted lines show those from specimens properly reheated in a sealed furnace, the temperature not rising above a copper color.

We find by comparing the tests of untreated specimens with those properly reheated (pp. 598) that the—

Elastic limit is lowered in every (7) case;

The yield point is lowered in every (8) case;

The tenacity is lowered in every (8) case;

The elongation is increased in 5 cases, and lowered in 3;

The modulus of elasticity is increased in 5 cases, and lowered in 3;

The proportional resilience is increased in 5 cases, and lowered in 3.

By comparing the tests of untreated specimens with those improperly heated, as above described, we find that the—

Elastic limit is lowered in every (17) case;

Yield point is lowered in every (17) case;

Tenacity is lowered in every (17) case;

Elongation is increased in 10 cases, and lowered in 6;

Modulus of elasticity is increased in 8 cases, and lowered in 9;

Proportional resilience is increased in 6 cases, and lowered in 11.

While the elastic limit in the first seven cases is lowered on an average but 3,514 lbs. per square inch, in the seventeen others it is lowered 9,188 lbs. per square inch. Likewise in the case of the yield point: while in the eight cases there is an average drop of 3,746 lbs. per square inch, there is as much as 8,707 lbs. per square inch in the case of the seventeen (17) others.

Again, the drop in tenacity in eight (8) cases is 5,270 lbs. per square inch, while in the seventeen (17) others it is 10,146 lbs. per square inch.

Then in elongation in the five (5) cases properly treated there is an average increase of 4.6%, and an average decrease in three of 1.5%; while in the case of the improperly tested specimens there is an average increase of only 3.1% in 10 cases, and a decrease of 1.27% in six.

Looking at the modulus of elasticity we find that in the first case the average increase in six tests was 643,000 lbs. per square inch, and the decrease in three tests was 850,000; and in the second case in eight (8) tests the average increase was 761,000 lbs. per square inch, and in nine (9) cases the average decrease was 572,000 lbs. per square inch.

While in the first case proper treatment increased the proportional resilience an average of 262 in five cases, it was decreased 159 in three others; in the second case, improper treatment, the increase was 302 in only six tests, while the decrease was 219 in eleven cases.

We may summarize all of this in the following general conclusions, bearing in mind that the number of tests given is but a limited one, and further investigation is necessary and essential.

1. Reheating to a high temperature is decidedly injurious to this class of steel in every way, as the elastic limit, yield point, tenacity, modulus of elasticity, and proportional resilience are decidedly lowered.

2. Reheating to a copper color by radiated heat in a closed

chamber in which there are neither gases, flame, nor smoke is probably very beneficial to the material as regards ductility and resilience.

3. Proper treatment undoubtedly lowers elastic limit and tenacity of plates of this steel of all sizes and thickness.

Although the latter is undoubtedly true, the more important qualities of this steel, those which indicate its shock-resisting power, ductility, and resilience, more than counterbalance this. If the unit strains allowable in any structure are based on these qualities, as they properly should be, then the material is undoubtedly improved.

Some, basing unit strains on elastic limit or yield point alone, may, however, urge that the material is weakened. To such we can only say that they must base their calculation on a better factor, which recent investigation has demonstrated to be a reliable one.

It must also be borne in mind that the material shows much more uniform qualities within the elastic limit, as shown by the accompanying plate of curves, and that, as the material is only strained considerably below this point, its exact location is of secondary importance, so long as it is well beyond the unit strains, and the ductility and resilience are satisfactory.

Looking at the results of tests on table p. 591, it will be seen that the elastic limit is found at about 43,850 lbs. per square inch after the material had been properly reheated, while the untreated specimen shows 47,360 lbs. per square inch. This reduction is of very little importance when considering that the average resilience is increased over 15% and that the unit strains never exceed 15,000 lbs. per square inch.

To recapitulate, then, all of the foregoing, we find that—

1. Improper reheating (annealing) these two kinds of structural steel injures them decidedly in several ways, particularly as regards elasticity and ductility.

2. Properly reheating (annealing) these two kinds of steel is decidedly beneficial in both cases, but much more so in the case of bars than in plates.

3. Further experiment in the directions above described is highly desirable and may lead to valuable results.

4. Better apparatus for recording results of tests, particularly autographic recording devices for large curves up to the yield point, are a most pressing necessity.

RESULTS OF TESTS OF BARS.

KIND OF TEST.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
7" x ⅞" BARS.											
Billet Test........	5,309	1,483 / 1,484	.25	.98	... .	48,990	74,920	22.0	35.3	30,858,000	1,648
Rolled Test........	......	819	....	....	35,700	40,000	72,420	25.0	54.8	28,883,000	1,810
Annealed Test ...	......	897	....	....	37,500	40,560	66,800	24.75	50.27	30,453,000	1,653
Billet Test........	5,328	1,500 / 1,501	.26	...	....	45,980	72,100	24.25	40.2	30,580,000	1,748
Rolled Test........	......	793	....	....	35,900	38,000	70,200	26.5	55.9	30,746,000	1,860
Annealed Test....	.. ...	913	....	....	....	34,900	72,470	25.0	54.9	31,513,000	1,812
Billet Test........	5,344	1,508 / 1,509	.24	.75	....	46,830	73,300	24.45	43.0	29,135,000	1,792
Rolled Test........	......	774	....	....	36,000	39,020	72,000	23.75	44.0	29,767,000	1,710
Annealed Test....	......	1,644	....	....	38,200	41,700	70,600	26.25	57.5	31,904,000	1,853
7" x 1⅛" BARS.											
Billet Test........	5,355	1,528 / 1,529	.24	.87	... ...	47,060	73,590	23.9	40.85	29,850,000	1,707
Rolled Tests....	.. ...	754	...	...	30,680	39,590	70,230	23.25	42.4	31,157,000	[illegible]
	.... .	1,104	...	...	34,000	37,607	67,880	26.0	48.4	29,894,000	[illegible]
Billet Test........	5,374	1,530 / 1,531	.22	.73	....	46,210	70,590	24.35	42.6	28,940,000	1,719
Annealed Test....	......	911	...	...	37,700	40,500	64,990	26.9	57.9	31,718,000	1,748
Rolled Test........	......	1,096	...	...	32,880	39,650	72,130	20.5	30.1	32,082,000	1,479
Billet Test........	5,376	1,536 / 1,537	.22	.79	....	43,070	67,570	24.1	48.3	30,510,000	1,698 / 1,761
Rolled Test........	......	778	...	...	38,400	42,275	75,640	21.25	46.6	31,814,000	1,607
Annealed Test....	......	919	...	...	32,420	39,780	70,870	25.75	56.2	31,416,000	1,895
Billet Test........	5,378	1,542 / 1,543	.26	.77	... .	46,825	71,630	20.8	34.9	28,690,000	1,490 / 1,785
Rolled Test........	... ..	786	...	....	32,200	37,550	69,630	25.0	55.9	29,286,000	1,741
7" x 1 3/16" BARS.											
Billet Test........	5,287	381 / 382	.20	.68	....	48,695	68,360	22.4	43.85	34,850,000	1,531
Rolled Test........	. ....	765	....	....	28,300	36,460	69,500	23.25	44.7	28,800,000	1,616
Billet Test........	5,289	383 / 384	.21	.61	....	45,575	72,805	21.55	35.95	28,500,000	1,569
Annealed Test....	... ..	901	....	....	33,220	40,290	70,850	24.75	51.2	31,058,000	1,734
Billet Test........	5,292	389 / 390	.25	.74	....	44,255	70,010	26.05	44.85	29,530,000	1,834
Rolled Test........	......	824	....	....	31,000	36,780	70,000	25.0	53.2	31,078,000	1,750
Annealed Test....	......	918	....	....	33,000	39,780	70,870	25.75	56.2	31,763,000	1,895
7" x 1¼" BARS.											
Billet Test........	5,289	383 / 384	.21	.61	....	45,575	72,800	21.53	35.95	28,500,000	1,569
Rolled Tests....	......	728	...	...	32,000	34,650	71,540	23.75	53.1	28,534,000	1,699
	......	707	...	...	35,800	38,910	74,270	24.5	43.0	30,422,000	1,830
	.....	1,095	...	...	35,200	38,210	70,410	21.1	31.3	30,29[illegible],000	1,446
Annealed Test ...	......	1,651	...	...	36,000	39,070	73,090	23.25	43.7	30,967,000	1,700
Billet Test	5,291	387 / 388	.20	.66	....	44,445	69,270	23.55	38.8	29,800,000	1,631
Rolled Test........	......	1,091	...		35,900	41,050	72,210	21.75	38.4	30,818,000	1,571
Annealed Test....	......	900			40,250	41,780	66,510	26.0	51.0	31,921,000	1,729

RESULTS OF TESTS OF BARS.—*Continued.*

7″ x 1$\frac{5}{16}$″ Bars.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test........	5,290	385, 386	.23	.57	....	44,315	69,265	23.80	41.8	29,700,000	1,648
Annealed Tests.	......	903	....	....	35,230	38.810	69,400	25.0	55.6	31,302,000	1,735
	......	917	....	....	35,000	36,400	66,460	26.5	56.6	31,530,000	1,761
Billet Test........	5,375	1,532, 1,533	.23	.83	....	46,085	69,470	24.25	40.4	29,840,000	1,684, 1,760
Rolled Test........	.. ..	789	...	....	31,000	37,470	67,430	26.25	55.8	31,580,000	1.770
Annealed Test....	......	1,643	....	....	33,720	41,700	66,235	27.0	56.4	32,117,000	1,789
Billet Test	5,394	1,560, 1,561	.22	.51	....	47,910	70,740	24.0	43.3	30,160,000	1,698
Rolled Test........	......	852	....	....	32,100	35,800	68,160	25.75	54.1	30.813,000	1,755
Annealed Test....	......	889	....	...	39,000	42,530	70,520	24.6	50.6	30,856,000	1,735

7″ x 1$\frac{3}{8}$″ Bars.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test........	5,278	381, 382	.20	.63	....	43,695	68,360	22.4	43.85	24,350,000	1,581
Rolled Test........	......	870	....	....	27,380	33,140	65.200	27.5	45.4	28,051,000	1,793
Annealed Test....	.. ..	915	....	...	30,680	37,880	67,590	25.75	53.4	29,387,000	1,740
Billet Test.... ...	5,324	1,493, 1,494	.25	.67	....	44,100	70,800	23.15	22.3	32,190,000	1,639
Rolled Test........	......	726	....	....	31,800	34,110	70,850	24.0	35.1	30,102,000	1.700
Annealed Test....	......	916	....	....	36,260	40,160	70,370	21.9	32.8	31,079,000	1,541
Billet Test.......	5,424	1,596, 1,597	.25	.72	....	45,740	72,880	23.65	38.1	29,455,000	1,710
Rolled Test........	.. ...	853	....	....	30,700	39,000	74.850	14.0	24.1	29,588,000	1,065
Annealed Test....	......	895	....	...	37,280	39,200	69,600	22.9	41.2	30,528,000	1,594
Billet Test.......	5,342	1,506, 1,507	.22	...	....	45,645	68,910	25.1	47.6	29,170,000	1,730
Annealed Tests.	......	908	...	....	33,000	39,800	70,000	24.2	54.8	31,531,000	1,694
	......	1,653	....	....	30,480	33,844	65,240	28.25	58.4	31,078,000	1,848
Billet Test........	5,344	1,508, 1,509	.24	.75	....	46,830	73,300	24.45	43.0	29,135,000	1,792
Rolled Tests....	......	723	....	....	31.500	36,970	71,030	21.6	52.2	28,800,000	1,533
	. ...	822	....	....	35,450	40,253	74,560	23.0	49.3	30,856,000	1,715
	......	1,096	....	....	34,400	36,330	68,170	25.0	51.9	30,528,000	1,704
Billet Test.... ..	5,345	1,510, 1,511	.18	.63	....	44,610	68,620	24.0	47.1	27,950,000	1,647, 1,609
Rolled Tests....	......	746	...	....	34,200	39,780	70,670	23.0	39.6	30,420,000	1,625
	......	777	...	....	32,850	37,260	69,090	22.5	35.0	31,416,000	1,554
Billet Test.... ...	5.375	1,534, 1,535	.23	.83	....	46,085	69,470	24.25	40.4	29,840,000	1,684, 1,760
Rolled Test........	......	707	....	....	34,300	35,580	67.170	25.0	53.0	29,791,000	1,679
Annealed Test...	......	920	....	....	33,800	40,000	67,590	25.0	55.5	29,895,000	1,689
Billet Test........	5,376	1,536, 1,537	.22	.79	....	43,070	67,570	24.1	48.8	30,510,000	1,628, 1,761
Rolled Test	......	757	....	....	33.120	37,490	68,000	22.0	40.5	29,792,000	1,496
	......	787	....	....	37,330	40,440	79,080	18.0	24.2	30,208,000	1,423
Annealed Test....	......	* 1,094	....	....	42,420	46,830	81,520	13.75	12.4	31,078,000	1,120
Billet Test........	5.381	1,552, 1,553	23	.90	... .	47,660	73,470	24.5	48.75	28,400,000	1,800
Rolled Test........	.....	829	...	....	31,840	38,620	71,300	27.0	58.20	33,314,000	1,925
Annealed Test....	......	887	....	....	40,800	41,950	72,950	25.9	51.9	30,637,000	1,890

* Test piece 1,094 was wrongly marked.

RESULTS OF TESTS OF BARS.—*Continued.*

7″ x 1⅜″ Bars.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test........	5,394	1,560 / 1,561	.22	.51	....	47,910	70,740	24.0	43.3	30,160,000	1,698
Rolled Test	......	868	....	....	34,250	37,760	74,300	22.75	40.7	30,856,000	1,690
Annealed Test....	......	892	...	....	33,200	37,930	68,650	22.5	43.2	31,647,000	1,544
Billet Test	5,407	1,580 / 1,581	.22	.55	....	45,925	69,825	24.6	45.1	30,580,000	1,617 / 1,785
Rolled Tests....	......	823	....	....	26,900	34,000	68,800	20.0	23.9	31,763,000	1,376
	......	850	....	....	31,700	38,430	71,910	16.25	21.2	31,531,000	1,169
Billet Test........	5,412	1,590 / 1,591	.18	.69	....	45,400	69,920	24.45	45.8	30,040,000	1,709 / 1,555
Rolled Test.......	......	867	....	....	36,500	40,160	74,900	20.0	27.9	31,531,000	1,598
Annealed Test....	......	907	....	....	34,500	39,520	73,240	25.5	50.9	31,999,000	1,868
Billet Test........	5,425	1,598 / 1,599	.23	.41	....	45,685	71,240	23.9	41.7	29,755,000	1,702
Rolled Tests....	......	711	....	....	31,250	32,670	70,700	23.0	50.2	30,208,000	1,626
	......	758	....	...	36,050	38,820	75,000	21.3	39.5	31,999,000	1,597
Billet Test........	5,426	1,600 / 1,601	.25	.39	....	43,705	70,440	26.65	44.2	30,240,000	1,877
Rolled Test.......	......	825	....	....	38,280	41,310	74,020	13.75	20.0	31,022,000	1,018
Annealed Test....	......	890	....	....	37,750	38,250	68,200	21.25	37.0	31,416,000	1,449

7″ x 1 7/16″ Bars.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test........	5,290	385 / 386	.23	.57	....	44,815	69,260	23.8	41.8	29,700,000	1,646
Rolled Test.......	... ..	826	..	....	30,690	33,720	64,350	23.75	45.0	30,102,000	1,528
Annealed Test....	......	914	..	....	28,850	35,940	66,260	28.0	55.3	32,976,000	1,855
Billet Test..	5,299	393 / 394	.25	74	....	50,070	74,680	20.7	37.4	32,770,000	1,546
Rolled Test..... .	......	701	...	..	29,950	33,260	66,400	25.5	51.6	28,514,000	1,693
Annealed Test....	......	* 1,102	...	..	33,800	40,850	81,610	18.0	29.0	31,189,000	1,469
Billet Test........	5,300	395 / 396	.24	71	....	47,330	70,350	21.65	41.9	28,920,000	1,523
Rolled Test.......	......	716	...	..	31,530	31,510	66,180	26.25	54.9	32,601,000	1,737
Annealed Test....	. ....	1,654	...	..	30,500	34,470	65,430	27.0	52.67	29,558,000	1,766
Billet Test........	5,307	401 / 402	.21	67	....	47,705	69,700	20.7	35.15	27,630,000	1,443
Rolled Test	......	851	...	..	28,700	34,370	66,000	25.0	50.6	29,893,000	1,650
Annealed Test....	......	886	...	..	37,620	39,600	68,780	26.25	56.0	30,747,000	1,805
Billet Test...... .	5,324	1,493 / 1,494	.25	.67	....	44,100	70,800	23.15	22.3	32,190,000	1,639
Rolled Test.......	......	865	...		29,070	33,000	67,130	26.25	48.7	30,856,000	1,762
Annealed Test....	......	898	...		33,000	37,840	65,750	25.4	45.9	31,303,000	1,670
Billet Test........	5,328	1,500 / 1,501	.26	....	....	45,980	72,100	24.25	40.2	30,580,000	1,748
Rolled Test.......	......	747	. .	....	29,050	37,850	66,390	24.5	58.8	30,313,000	1,627
Annealed Test ..	......	1,100	...	....	36,280	37,890	68,640	18.0	23.5	30,746,000	1,236
Billet Test	5,347	1,514 / 1,515	.23	.57	....	47,625	72,150	21.8	33.3	29,055,000	1,573
Rolled Test	......	720	....	....	27,000 ?	34,390	65,880	26.75	53.2	....	1,763
Annealed Test....	......	905	....	....	38,220	40,100	67,400	25.0	52.0	30,421,000	1,693
Billet Test...... ..	5,381	1,552 / 1,553	.23	.90	....	47,660	73,470	24.5	43.75	28,400,000	1,800
Rolled Test.......	.. ...	709	...	...	26,840 ?	32,975	66,390	20.25	30.5	....	1,344
Annealed Test....	......	899	...	...	32,100	35,520	63,920	26.6	54.0	30,855,000	1,700

* Test piece 1,102 was wrongly marked.

RESULTS OF TESTS OF BARS.—*Continued.*

7" x 1 7/16" Bars.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 Inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test	5,426	1,600 / 1,601	.25	.39	....	43,705	70,440	26.8	44.2	30,240,000	1,888
Rolled Test	......	828	....	....	27,130	35,280	66,930	25.0	51.0	....	1,673
Annealed Test	......	902	....	....	35,950	38,300	66,650	24.6	53.4	28,420,000	1,640
Billet Test	5,441	......	....	....	....	......	......	.....	.....	....	......
Rolled Test	......	763	....	....	34,100	37,090	68,290	22.0	40.0	29,998,000	1,502
	......	1,101	....	....	33,580	39,140	76,900	20.5	44.2	28,991,000	1,576
Annealed Test	......	906	....	....	36,930	39,080	67,510	25.5	50.9	28,512,000	1,722
Billet Test	5,454	1,610 / 1,611	.21	.55	....	44,700	71,000	23.25	38.25	29,055,000	1,651
Rolled Test	......	697	....	....	28,200	36,000	69,880	24.75	51.2	32,602,000	1,730
Annealed Test	......	1,648	....	....	32,530	36,580	66,810	26.25	55.25	30,857,000	1,754

7" x 1¼" Bars.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 Inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test	5,406	1,576 / 1,577	.23	.64	....	47,350	71,595	22.0	37.9	29,840,000	1,575
Rolled Test	......	792	....	....	33,730	38,000	70,090	26.25	48.8	30,528,000	1,840
Annealed Test	.. ..	1,646	....	....	34,620	38,500	69,840	27.5	57.5	30,528,000	1,921

7" x 1 1/16" Bars.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 Inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test	5,347	1,514 / 1,515	.23	.57	....	47,625	72,150	21.8	38.3	29,055,000	1,578
Rolled Test	......	827	....	...	30,900	35,340	68,270	26.25	52.4	30,312,000	1,792
	......	1,092		....	30,500	34,050	68,680	26.0	49.9	29,999,000	1,786
Annealed Test	.....	891	....	....	39,830	39,400	72,600	24.5	52.0	31,761,000	1,779
	......	912	....	....	35,480	37,260	72,000	26.0	51.5	31,032,000	1,872
Billet Test	5,857	....	....	....	....	......	......	.....	.....	....	......
Rolled Test	.....	869	....	....	34,500	37,665	69,540	24.4	43.5	30,635,000	1,697
	......	1,093	...	..	36,180	41,640	73,760	11.75	14.8	30,746,000	867
Annealed Test	......	893	....	....	38,550	40,640	71,855	24.0	48.0	31,880,000	1,724
Billet Test	5,383	1,554 / 1,555	.23	.74	....	45,855	70,020	23.55	39.75	28,600,000	1,649
Rolled Test	......	776	....	..	34,280	36,020	66,630	26.5	57.7	30,856,000	1,766
Annealed Test	....	921	...	....	35,000	38,400	68,000	29.0	56.4	31,763,000	1,972
Billet Test	5,411	1,586 / 1,587	.19	.79	....	45,250	69,530	24.2	46.4	29,675,000	1,688 / 1,759
Rolled Test	......	830	....	....	28,800 ?	36,050	70,770	25.0	53.5	30,209,000	1,769
Annealed Test	......	894	....	....	39,100	40,060	70,000	25.9	53.5	30,966,000	1,818
Billet Test	5,412	1,590 / 1,591	.18	.09	....	45,400	69,920	24.45	45.8	30,040,000	1,709 / 1,556
Rolled Test	......	790	....	....	28,900	35,540	70,000	25.0	47.0	29,999,000	1,750
Annealed Test	... ..	902 *a*	....	....	38,250	39,850	68,600	24.1	53.4	30,856,000	1,650
Billet Test	5,426	1,600 / 1,601	.25	.39	....	43,705	70,440	26.8	44.2	30,240,000	1,888
Rolled Test	......	840		. .	28,740	32,780	65,840	24.75	51.8	30,637,000	1,629
Annealed Test	......	909		....	34,830	37,340	64,400	26.0	53.5	31,190,000	1,674
Billet Test	5,446	....		...	....	......	....	.....	.....	....	.....
Rolled Test	.....	791		....	28,880	34,390	69,540	20.6	26.9	31,999,000	1,432
Annealed Test	......	910		..	37,200	38,120	67,350	26.9	46.8	29,791,000	1,815
Billet Test	5,454	1,610 / 1,611	.21	.55	......	44,700	71,000	23.25	38.25	29,055,000	1,651
Rolled Test	......	863		....	36,250	39,240	72,110	22.0	35.7	31,880,000	1,586
Annealed Test	......	904		...	37,000	40,450	73,000	26.0	53.9	32,358,000	1,898

RESULTS OF TESTS OF BARS.—*Continued.*

7″ x 1 11/16″ Bars.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test........	5,406	1,576 / 1,577	.23	.64	....	47,350	71,595	22.0	37.9	29,840,000	1,575 / 1,655
Rolled Test.......	......	866	....	....	34,970	36,730	68,820	23.75	44.9	31,078,000	1,634
Annealed Test...	......	888	....	....	36,970	40,000	69,070	21.6	32.4	31,078,000	1,499

7″ x 1¾″ Bars.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test........	5,447	1,606 / 1,607	.27	.42	....	44,960	68,770	25.0	40.0	28,285,000	1,717 / 1,673
Rolled Test.......	......	696	....	....	33,100	41,040	74,060	19.75	42.0	31,647,000	1,468
Annealed Test. ..	......	896	....	...	35,000	37,550	65,420	22.9	35.0	30,746,000	1,496

7″ x 1 13/16″ Bars.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test........	5,424	1,596 / 1,597	.25	.72	....	45,740	72,330	23.65	33.1	29,455,000	1,710
Rolled Test.......	......	719	....	....	29,080	31,510	66,180	26.25	54.9	32,479,000	1,787
Annealed Test....	......	1,650	....	....	32,100	36,640	73,670	23.75	50.4	31,308,000	1,750

7″ x 1⅞″ Bars.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test.......	5,878	1,542 / 1,543	.26	.77	....	46,330	71,680	20.8	34.9	28,585,000	1,499 / 1,795
Rolled Test.......	.....	713	....	....	29,000 ?	32,750	71,370	25.0	52.9	28,420,000	1,784
Annealed Test....	......	1,649	....	....	33,650	39,060	71,260	24.5	53.7	31,078,000	1,746

RESULTS OF TESTS OF PLATES.

24" x ½" Plate.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test........	5,483	1,206 / 1,207	.33	.59	....	48,830	79,520	21.6	38.6	30,440,000	1,718
Rolled Test.......	.....	966	....	...	46,260	53,470	86,430	20.0	37.8	29,893,000	1,728
Annealed Test....	......	1,083	....	....	33,730	41,420	69,840	24.0	53.4	29,999,000	1,676
		1,662	....	....	46,860	50,310	78,800	24.0	35.8	29,688,000	2,004
Billet Test........	5,444	1,208 / 1,209	.32	.75	....	50,770	82,920	18.65	31.4	30,550,000	1,546
Rolled Test.......	......	963	....	....	48,630	32,870	88,260	17.0	31.8	29,999,000	1,500
Annealed Test....	......	1,087	....	....	38,170	40,845	75,970	23.0	49.2	31,077,000	1,747
		1,663	....	....	43,830	46,800	78,270	24.25	51.2	29,188,000	1,898
Billet Test........	5,534	1,316 / 1,317	.34	.70	....	54,040	86,480	19.2	32.5	31,730,000	1,660
Rolled Test........	......	996	....	....	44,820	52,900	90,210	18.75	25.9	30,103,000	1,691
Annealed Test....	.. ...	942	....	....	38,550	40,920	79,610	17.5	41.2	29,999,000	1,393
		946	....	....	40,700	44,180	79,630	24.25	35.4	30,745,000	1,931
		1,070	....	....	40,420	43,380	70,744	20 25	51.8	27,871,000	1,432
Billet Test..	5,566	1,339 / 1,340	.26	.52	....	33,850	78,270	21.5	34.6	30,085,000	1,683
Rolled Test.......	......	1,003	....	....	48,830	51,300	81,350	22.12	47.8	30,103,000	1,881
Annealed Test....	......	1,081	....	....	36,430	41,870	71,640	22.0	53.2	29,687,000	1,576
Billet Test........	5,568	1,341 / 1,342	.32	.55	....	51,010	80,340	21.75	41.85	30,200,000	1,747
Rolled Test.......	......	994	....	....	46,700	50,060	86,610	19.5	41.0	30,208,000	1,693
Annealed Test....	......	1,084	....	....	37,020	43,560	74,890	23.5	54.2	30,855,000	1,700

18" x ⅝" Plate.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test.......	54.19	1,192 / 1,193	.32	.52	....	48,960	79,040	21.9	35.7	29,840,000	1,781
Rolled Test..	......	979	..	...	47,170	48,440	76,550	23.0	51.2	29,487,000	1,761
Annealed Test ...	......	1,107	..	...	38,550	40,410	68,600	22.5	51.0	29,089,000	1,543
		1,658	...	...	45,100	46,900	75,780	27.5	46.8	30,313,000	2,084
Billet Test	5,472	1,244 / 1,245	.40	.56	....	49,560	83,840	20.2	31.8	30,160,000	1,698
Rolled Test.......	......	980	...	...	47,300	49,310	91,090	10.35	11.7	30,421,000	944
Annealed Test....	......	1,108	...	...	39,700	42,740	82,290	21.25	44.3	30,854,000	1,748

22" x ⅝" Plate.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test........	5,566	1,339 / 1,340	.26	.52	....	33,850	78,270	21.5	34.6	30,085,000	1,683
Rolled Test.......	. ....	982	..	....	44,030	45.485	75,470	22.25	49.7	31,580.000	1,679
Annealed Test....	......	1,073	...	....	31,510	34.740	67,710	23.25	51.1	30,208,000	1,574
		1,659	....	....	40,470	42,855	73,270	21.25	55.4	30,637,000	1,557

24" x ⅝" Plate.

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
Billet Test........	5,484	1,264 / 1,265	.43	.60	.. ..	50,130	84,970	19.7	30.8	29,220,000	1,674
Rolled Test.......	......	998	...	. .	43,570	46,040	79,520	20.5	39.3	30,636,000	1,630
Annealed Test....	......	1,076	...	...	38,100	40,120	70,330	19.5	35.8	31,302,000	1,371
Billet Test.	5.494	1,280 / 1,281	.34	.72	....	52,030	82,030	19.75	32.45	29,555,000	1,630
Rolled Test.......	......	968	...	. .	45,800	48.650	80,130	19.75	47.1	31,647,000	1,582
Annealed Test....	......	1.079	...	...	39.900	40.090	69.990	23.25	55.7	30.421,000	1,626
		1,660	...	...	45.300	46,790	77,710	18.0	23.1	31,189,000	1,398
Billet Test........	5,863	1,446 / 1,447	.29	.86	....	51,890	84,530	20.65	37.9	29,282,000	1,475
Rolled Test.......	. ....	965	...	...	47,500	50.120	87,760	23.75	45.2	30,855,000	2,084
Annealed Test....	......	1,090	...	...	39,500	43,945	76,000	22.0	61.4	31,303,000	1,672

RESULTS OF TESTS OF PLATES.—*Continued.*

Kind of Test.	Heat No.	Test No.	C.	Mn.	Elastic Limit.	Yield Point.	Tenacity.	Elong. p. cent. in 8 inches.	Per cent. Reduction.	Modulus of Elasticity.	Proportional Resilience.
20″ x ¾″ Plate.											
Billet Test........	5,530	1,310 / 1,311	.27	.64	....	50,610	79,155	21.05	35.3	27,170,000	1,668
Rolled Test.......	......	997	....	....	48,800	48,920	79,120	23.75	52.1	31,078,000	1,879
Annealed Test....	......	1,077	....	....	36,630	38,220	69,330	25.0	54.9	30,744,000	1,733
		1,656	....	....	43,100	46,030	77,620	22.0	45.9	31,766,000	1,708
Billet Test........	5,540	1,331 / 1,332	.29	.60	....	55,740	86,010	21.95	37.0	30,840,000	1,888
Rolled Test.	......	993	....	....	44,330	54,020	81,720	20.75	35.5	30,964,000	1,696
Annealed Test....	......	1,080	....	....	36,220	38,890	76,330	21.25	40.7	31,417,000	1,623
Billet Test........	5,568	1,341 / 1,342	.32	.55	....	51,010	80,340	21.75	41.85	30,200,000	1,747
Rolled Test.......	......	986	....	....	46,100	47,580	81,280	20.75	38.6	30,855,000	1,686
Annealed Test....	......	1,109	....	....	37,000	38,900	71,760	22.0	38.5	30,636,000	1,578
		1,657	....	..	42,250	44,560	78,170	22.5	51.3	31,078,000	1,758
Billet Test.......	5,866	1,450 / 1,451	.26	.81	....	50,620	81,025	21.25	42.65	29,762,000	1,723
Rolled Test.......	......	964	...	....	41,230	42,820	75,700	19.25	46.9	30,312,000	1,457
Annealed Test....	......	1,106	..	....	35,900	38,680	72,020	25.0	55.6	31,999,000	1,800
22½″ x ¾″ Plate.											
Billet Test........	5,910	1,456 / 1,457	.34	.58	......	51,700	88,560	19.7	37.2	30,090,000	1,698
Rolled Test......	......	967	....	....	43,000	44,875	77,815	24.5	38.5	31,188,000	1,906
Annealed Test....	......	1,071	....	...	36,770	35,190	65,500	23.5	47.4	30,526,000	1,474
24″ x 1⅜″ Plate.											
Billet Test........	5,484	1,264 / 1,265	.43	.60	......	50,130	84,970	19.7	30.8	29,220,000	1,674
Rolled Test.......	......	971	...	....	45,580	48,280	88,230	19.0	31.2	29,188,000	1,676
Annealed Test....	......	1,075	....	...	41,730	43,590	77,010	25.75	50.8	30,528,000	1,983

7" x $\frac{7}{8}$" Bars.

Test No. 774.		Test No. 793.		Test No. 819.	
Area (.5127) Load per sq. in.	Heat No. (5,744 B) Average Microm. Reading.	Area (.5153 Load per sq. in.	Heat No. (5,328 B) Average Microm. Reading.	Area (.5178) Load per sq. in.	Heat No. (5,309 B) Average Microm. Reading.
3,900	.0113.5	3,880	.0144.25	3,860	.0154.5
5,850	118	5,820	148.25	5,790	159.75
7,802	123.25	7,760	153	7,720	165
9,950	128.5	9,700	158	9,650	170.25
11,700	133.75	11,640	162.25	11,580	175.5
13,650	138.75	13,580	166.75	13,520	181
15,600	144	15,520	171.0	15,450	186.5
17,550	149.5	17,460	176.25	17,380	191.5
19,500	154.5	19,400	181.5	19,310	196.75
21,450	159.75	21,350	186	21,240	202.25
23,400	165.25	23,290	191.5	23,170	207.75
25,380	170.5	25,230	196.5	25,100	212.75
27,300	175.75	27,170	202.75	27,040	218.25
29,260	181	29,110	208.25	28,970	223.5
31,200	186.5	31,050	214	30,900	229
32,180	190	32,990	220	32,830	234.25
32,670	191.5	33,960	224	33,800	236.5
33,160	193	34,450	226	34,760	239.75
33,640	194.25	34,930	227.5	35,700	242.5
34,130	195.75	35,420	229.5	36,690	245.75
34,620	197.25	35,900	230.75	37,170	247.125
35,110	199.5	36,380	233.50	37,660	249
35,600	200.75	36,870	237	38,140	250.75
36,080	202.5			38,620	253
36,570	205			39,110	257.5
77,060	208				

7" x $\frac{7}{8}$" Bars.

Annealed.

Test No. 897		Test No. 913.		Test No. 1,644.	
Area (.5153) Load per sq. in.	Heat No. (5,309 A) Average Microm. Reading.	Area (.4681) Load per sq. in.	Heat No. (5,328 A) Average Microm. Reading.	Area (.4185) Load per sq. in.	Heat No. (5,744 A) Average Microm. Reading.
3,880	.0099	4,270	.0103.75	4,780	.0061.75
5,820	107.75	6,419	109.5	7,168	.0069.25
7,760	108.5	8,540	114.5	9,560	.0075
9,700	112.5	10,680	119.75	11,940	.0081.25
11,640	118.5	12,820	125.5	14,330	.0087
13,580	123.5	14,950	130.75	16,720	.0093.25
15,520	128.5	17,090	136.25	19,110	.0099.25
17,460	137.5	19,220	141.5	21,500	.0105.5
19,400	139	21,360	147	23,890	111.25
21,350	144	23,500	152.5	26,280	117.75
23,290	149	25,630	160.5	28,670	123.25
25,230	154.25	27,770	168.5	31,060	129.25
27,170	159	29,910	172	33,450	134.75
29,110	164.5	30,980	174.25	35,840	140.25
31,050	169.5	31,510	175.75	38,230	147.5
32,990	174.5				
33,960	177.5				
34,930	180.5				
35,420	183				
36,870	185.5				
37,360	187.25				
37,840	188.5				

7" x 1⅛" BARS.

Test No. 754.		Test No. 773.		Test No. 786.	
Area (.5052) Load per sq. in.	Heat No. (5,355 A) Average Microm. Reading.	Area (.5804) Load per sq. in.	Heat No. (5,376 B) Average Microm. Reading.	Area (.5158) Load per sq. in.	Heat No. (5,378 B) Average Microm. Reading.
3,960	.0063.5	3,840	.0168.25	3,880	.0168.5
5,938	66.25	5,760	173	5,820	173.25
7,919	70	7,680	178	7,760	178
9,898	75.25	9,600	183	9,700	183.5
11,878	80.25	11,520	187	11,640	188
13,858	84.75	13,450	192.5	13,580	193
15,837	89.5	15,370	196.75	15,520	197.75
17,815	94.5	17,290	201.5	17,460	203
19,795	.0100	19,210	206	19,450	208
21,775	105.5	21,140	211	21,340	213
23,753	110.25	23,060	216	23,280	219
25,733	115.5	24,980	220.75	25,220	223.75
27,713	117.75	26,900	225.5	27,170	230
29,693	120.75	28,820	230.5	29,110	235
31,672	127.25	30,740	235.25	31,050	241.5
33,651	126	32,660	241	32,020	243.75
34,642	128.75	33,620	243	32,990	249
35,137	131.5	34,110	244.75	33,470	251.25
35,633	134.75	34,590	245.5	33,960	253.5
36,128	137	35,040	247.25	34,440	255.75
		35,550	248.5	34,930	258.75
		36,030	250	35,420	264
		36,510	251.25		
		36,990	252.75		
		37,470	254.25		
		37,950	256		
		38,430	257.5		
		38,910	260		
		39,390	266.5		

7" x 1⅛" BARS.

ANNEALED.				NOT ANNEALED.			
Test No. 911.		Test No. 919.		Test No. 1,098.		Test No. 1,104.	
Area (.5229) Load per sq. in.	Heat No. (5,374 A) Average Microm. Reading.	Area (.5242) Load per sq. in.	Heat No. (5,376 A) Average Microm. Reading.	Area (.5178) Load per sq. in. Elongations.	Heat No. (5,374 B) Average Microm. Reading. Stress.	Area (.5153) Load per sq. in.	Heat No. (5,335 B) Average Microm. Reading.
3,825	.0116.25	3,815	.0127.5	3,860	.0139.5	3,880	.0172.5
5,737	120.5	5,722	132	5,790	144	5,820	177.5
7,650	125.5	7,630	136.25	7,720	148.5	7,760	182.5
9,562	130.25	9,539	141.25	9,650	153.5	9,700	187.75
11,473	135.5	11,440	146	11,580	158	11,640	194
13,387	140	13,350	151	13,520	162.5	13,580	198
15,300	145	15,260	156	15,450	167.5	15,520	203
17,210	150	17,170	161	17,380	172.5	17,460	208.5
19,120	154.5	19,070	165.5	19,310	177.25	19,450	213.5
21,035	159.5	20,980	171	21,240	183	21,340	219.25
22,930	164.5	22,890	175.75	23,170	187.5	23,280	224.75
24,860	169.5	24,800	181.25	25,100	192.5	25,220	230.25
26,770	174	26,710	187.5	27,040	197.5	27,170	236
28,680	179	28,610	194	28,970	203	29,110	241
30,600	184	30,520	204.5	30,900	208	31,050	247
32,510	189.5	31,470	211	32,870	214	32,990	254.25
33,470	192	32,430	216.5	34,760	222.5	33,960	258.5
34,420	194.5	33,380	227			34,930	269.5
35,380	197.5						
36,330	199.5						
37,290	202						
37,770	203.5						
38,250	206.5						

7" x 1$\frac{3}{16}$" Bars.

Not Annealed.				Annealed.			
Test No. 765.		Test No. 824.		Test No. 901.		Test No. 918.	
Area (.5294) Load per sq. in.	Heat No. (5,287 A) Average Microm. Reading.	Area (.5166) Load per sq. in.	Heat No. (5,292 B) Average Microm. Reading.	Area (.5102) Load per sq. in.	Heat No. (5,289 A) Average Microm. Reading.	Area (.5153) Load per sq. in.	Heat No. (5,292 A) Average Microm. Reading.
3,778	.0096.5	3,870	.0119.5	3,920	.0111	3,880	.0122.75
5,666	101	5,807	123.75	5,880	116	5,820	127.25
7,566	106.25	7,740	128.75	7,840	120.5	7,760	132
9,445	110.75	9,680	133.75	9,800	125.5	9,700	136.75
13,330	116	11,610	138.5	11,760	130	11,640	142
13,220	121.25	13,550	143.5	13,720	135	13,580	146.5
15,110	126.5	15,480	148.5	15,680	140	15,520	151
17,000	131.25	17,420	153.5	17,640	145	17,460	156.25
18,890	136.25	19,360	158.5	19,600	149.5	19,400	161.25
20,780	141.5	21,290	163.5	21,560	155.5	21,350	166
22,670	147.5	23,230	168.75	23,520	161.25	23,290	171
24,550	152.75	25,160	174	25,480	165.5	25,230	176
26,440	159	27,100	179	27,440	171	27,170	181.75
28,330	165.5	29,030	184	29,400	176	29,110	185.75
30,220	173	30,970	190.25	31,360	181.5	31,050	191.25
31,170	177.5	32,910	200.75	33,320	186.5	32,020	194.5
32,110	183	34,840	209.5	34,300	189	32,990	198.5
32,390	186.5			35,280	192	33,470	201.5
				36,230	194.5	33,900	204.5
				37,240	198	34,450	207
				38,220	206.5	34,930	213.5

7" x 1$\frac{1}{4}$" Bars.

Not Annealed.						Annealed.	
Test No. 728.		Test No. 767.		Test No. 821.		Test No. 900.	
Area (.5166) Load per sq. in.	Heat No. (5,289 B) Average Microm. Reading.	Area (.5166) Load per sq. in.	Heat No. (5,289 A) Average Microm. Reading.	Area (.5229) Load per sq. in.	Heat No. (5,289 B) Average Microm. Reading.	Area (.5217) Load per sq. in.	Heat No. (5,291 A) Average Microm. Reading.
3,870	.0097.75	3,870	.0107	7,650	.0150	3,830	.0122.5
5,807	103.25	5,807	111.75	9,560	155	5,750	127.5
7,740	107.75	7,740	117.25	11,470	159.75	7,660	131.5
9,680	113.25	9,680	121.75	13,380	164.5	9,580	136
11,610	117.75	11,610	127.25	15,300	169.75	11,500	141
13,550	123	13,550	132.50	17,210	174.25	13,410	146
15,480	128.75	15,480	137.5	19,120	179.5	15,330	150.5
17,420	133.5	17,420	142.5	21,030	184.51	17,250	156.5
19,360	138.75	19,360	147.5	22,950	189.75	19,160	161
21,290	144.5	21,290	152.75	24,860	195	21,080	165.5
23,230	150	23,230	158	26,770	200	23,000	171
25,161	155.5	25,160	163.5	28,680	205.5	24,910	176
27,100	162	27,100	168.5	30,600	211	26,830	180.5
29,030	167.25	29,030	173.75	32,510	218.25	28,750	185
30,000	170.5	30,970	179.5	32,980	221.25	30,670	190
30,970	173.75	31,940	182.25	33,460	223.75	32,580	194.75
31,940	176.75	32,910	185.5	33,940	227.5	33,540	197.5
32,420	178.75	33,880	188.25	34,420	233	34,500	200
32,910	180.75	34,160	189.25			35,460	203
33,390	183	34,360	189.5			36,420	205$\frac{1}{4}$
33,880	185.5	34,840	191.75			37,370	208
34,360	189.5	35,330	193			38,370	210.5
34,840	200	35,810	195.5			39,290	213
		36,100	196.75			40,250	216
		36,300	198.25				

7" x 1¼" Bars.

Not Annealed.				Annealed.	
Test No. 1,091.		Test No. 1,095.		Test No. 1,651.	
Area (.5294) Load per sq. in.	Heat No. (5,291 B) Average Microm. Reading.	Area (.5397) Load per sq. in.	Heat No. (5,289 B) Average Microm. Reading.	Area (.5281) Load per sq. in.	Heat No. (5,289 A) Average Microm. Reading.
3,778	.0121	3,706	.0146	3,787	.0052.75
5,666	126	5,560	150.75	5,680	57.75
7,556	130.25	7,410	155.5	7,570	63
9,445	135.5	9,260	160.5	9,470	67.75
11,330	140	11,120	164.75	11,360	73
13,220	145.5	12,970	169.5	13,250	78.25
15,110	150	14,820	174.75	15,150	83
17,000	156	16,670	179.5	17,040	88
18,890	161	18,530	184	18,930	92.5
20,780	165.5	20,380	189.25	20,830	98
22,670	171	22,230	193.75	22,720	102.5
24,550	176	24,090	198	24,620	107.5
26,440	181	25,940	204	26,510	112.25
28,330	186	27,790	209.5	28,400	117
30,220	191.5	29,650	215	30,700	121.75
32,110	197	31,500	219.5	32,190	126.75
34,000	203	33,350	224.5	34,080	132.75
35,890	209.5	35,210	230.5	35,030	135.5
37,780	219			35,980	139

7" x $1\frac{5}{16}$" Bars.

Not Annealed.				Annealed.	
Test No. 789.		Test No. 852.		Test No. 889.	
Area (.5191) Load per sq. in.	Heat No. (5,375 B) Average Microm. Reading.	Area (.5217) Load per sq. in.	Heat No. (5,394 B) Average Microm. Reading.	Area (.5255) Load per sq. in.	Heat No. (5,394 A) Average Microm. Reading.
3,850	.0187	3,830	.0138.5	3,806	.0230
5,780	191	5,750	143.5	5,709	235
7,700	195.75	7,668	148	7,610	239
9,630	200.5	9,580	153	9,510	244.5
11,560	205.25	11,500	158	11,410	249
13,480	210.25	13,420	163.25	13,320	254
15,410	215	15,330	168	15,220	259
17,340	220	17,250	173.25	17,120	263.75
19,260	225	19,170	178.5	19,030	269
21,190	230	21,080	183.5	20,930	274
23,110	234.75	23,000	189.25	22,830	279
25,040	239.75	24,920	194.25	24,730	284
26,970	245.25	26,830	200.25	26,640	288.5
28,900	250.75	28,750	206.5	28,540	293.25
30,820	258	30,670	213.25	30,450	298.5
31,780	262.5	31,620	217.5	31,400	301.5
32,270	265	32,100	219.25	32,350	303.5
32,750	267.5	32,580	223.75	33,300	306.5
33,230	272			34,250	308.5
				35,200	310.75
				36,150	313.5
				37,100	316
				38,060	318.25
				39,010	321

7" x $1\frac{5}{16}$" Bars.

Annealed.

Test No. 908.		Test No. 917.		Test No. 1,645.	
Area (.5102) Load per sq. in.	Heat No. (5,290 A) Average Microm. Reading.	Area (.5140) Load per sq. in.	Heat No. (5,290 B) Average Microm. Reading.	Area (.5217) Load per sq. in.	Heat No. (5,375 A) Average Microm. Reading.
3,920	.0096.75	3,800	.0187.75	3,830	.0194.75
5,880	101.5	5,830	142.25	5,750	199
7,840	106	7,780	147	7,668	203.25
9,800	110.5	9,730	151.5	9,580	208
11,760	115.5	11,670	156.5	11,500	212.75
13,720	120.5	13,620	161.5	13,420	217.5
15,680	125.5	15,560	166.25	15,330	223.75
17,640	130.125	17,510	171	17,250	227.5
19,600	135.25	19,450	176	19,170	232.5
21,560	140.25	21,400	181	21,080	237
23,520	145.5	23,340	185.75	23,000	241.5
25,480	150.75	25,290	191.25	24,920	248
27,440	155.5	27,240	196.25	26,830	254.75
29,400	160.5	29,180	201.25	28,750	258.75
31,360	165.5	31,130	207.25	30,670	263.5
33,320	171	33,070	211.5	32,580	268.5
34,300	177.5	35,020	219	33,540	271
35,280	176.5			34,500	278
36,260	186				

7" x $1\frac{3}{4}$" Bars.

Test No. 707.		Test No. 711.		Test No. 723.		Test No. 726.	
Area (.5242) Load per sq. in.	Heat No. (5,375 B) Average Microm. Reading.	Area (.5127) Load per sq. in.	Heat No. (5,425 B) Average Microm. Reading.	Area (.5153) Load per sq. in.	Heat No. (5,344 B) Average Microm. Reading.	Area (.5204) Load per sq. in.	Heat No. (5,324 B) Average Microm. Reading.
3,805	.0233	3,900	.0101.75	3,880	.0206.75	3,840	.0076.25
5,723	237	5,850	106.25	5,820	210.75	5,765	79.75
7,630	242	7,800	111	7,760	213.5	7,686	84.75
9,558	244.75	9,750	116.5	9,700	218	9,608	89.5
11,440	249.75	11,700	121	11,640	220.5	11,520	94
13,350	255	13,650	125.5	13,580	227.75	13,450	98.5
15,260	259.5	15,600	131	15,520	233	15,370	103.5
17,160	261	17,550	136.25	17,460	237.75	17,290	109
19,070	270.75	19,500	141.75	19,450	243.25	19,210	113.75
20,980	275	21,450	147.25	21,340	248.5	21,140	119
22,890	281	23,400	154	23,280	254.75	23,060	124.25
24,800	285.5	25,360	162	25,220	259.5	24,980	129.25
26,710	290.5	27,310	170.25	27,170	266	26,900	134.625
28,610	296.5	29,260	180	29,110	272	28,820	139.75
30,520	302	31,210	198.25	31,500	281	30,740	146
32,430	308	32,180	245.5	32,020	286	31,710	149.5
34,330	317	32,670	257.5	32,500	289	32,660	153.75
35,290	331.5			32,990	291	33,150	156.5
35,570	368			33,420	297	33,630	159
				33,960	300	34,110	162.75
				34,440	308		
				34,930	332.5		

7' x 1⅜" Bars.

Test No. 746.		Test No. 757.		Test No. 758.		Test No. 777.	
Area (.5115) Load per sq. in.	Heat No. (5,345 A) Average Microm. Reading.	Area (.5281) Load per sq. in.	Heat No. (5,376 A) Average Microm. Reading.	Area (.5204) Load per sq. in.	Heat No. (5,425 A) Average Microm. Reading.	Area (.5166) Load per sq. in.	Heat No. (5,345 B) Average Microm. Reading.
3,910	.0104.5	3,787	.01135	3,840	.0048875	3,877	.0079.25
5,860	109.5	5,680	1185	5,760	53.75	5,807	83
7,820	114	7,570	123	7,686	59	7,743	87.75
9,770	119.5	9,468	128.75	9,600	63.75	9,680	92.75
11,730	124.5	11,360	133.5	11,520	68.5	11,610	97.75
13,680	129.5	13,250	138.5	13,450	73.375	13,550	103
15,640	134.25	15,150	143.5	15,370	78	15,480	107.5
17,590	139.5	17,040	148.25	17,290	83	17,420	113
19,550	144.25	18,930	153.5	19,210	87.75	19,350	117.5
21,510	149.5	20,830	158.75	21,140	92.125	21,290	121.75
23,460	154.5	22,720	163.75	23,060	97.25	23,230	128
25,420	160	24,610	169.25	24,980	102.125	25,160	133
27,370	165	26,510	177.75	26,900	106.75	27,100	139
29,330	170.5	28,400	179	28,820	111.5	29,030	144.25
31,280	175.75	30,300	184.25	30,740	116.5	30,000	147.25
32,260	179	32,190	190	32,660	121.75	30,490	148
33,240	182	33,140	193.5	33,150	122.75	30,970	149.5
34,210	185	33,520	195.75	33,630	124.25	31,450	151
35,190	189.5	33,900	198.5	34,110	125.5	31,940	152.75
35,680	193.5			34,590	127	32,400	154.5
36,170	196.75			35,070	128.25	32,900	156.5
36,460	200.25			35,550	129.75	33,390	160
36,660	202.5			36,030	131.75	33,870	161.5
36,950	204.75			36,230	133.26	34,360	165.5
37,150	207.5			36,510	134		
				36,990	136.5		

7' x 1⅜" Bars.

Test No. 787.		Test No. 822.		Test No. 828.		Test No. 825.	
Area (.5153) Load per sq. in.	Heat No. (5,376 B) Average Microm. Reading.	Area (.5217) Load per sq. in.	Heat No. (5,344 B) Average Microm. Reading.	Area (.5204) Load per sq. in.	Heat No. (5,407 A) Average Microm. Reading.	Area (.5229) Load per sq. in.	Heat No. (5,426 B) Average Microm. Reading.
3,880	.0039.25	3,830	.0144	3,840	.0157	3,820	.0018.75
5,820	44.25	5,750	148.75	5,760	161	5,736	23.25
7,760	49	7,668	153	7,680	166.75	7,650	28
9,700	54	9,580	158.5	9,600	172	9,560	32.75
11,640	59.5	11,500	163	11,520	175.5	11,470	37.25
13,580	64.25	13,420	168.25	13,450	181	13,380	42.25
15,520	29.25	15,330	173.5	15,370	185.5	15,300	47.25
17,460	74.25	17,250	178.5	17,290	190.75	17,210	51.75
19,400	79	19,170	183	19,210	195.25	19,170	56.25
21,340	84.25	21,080	188	21,140	201.25	21,030	61
23,280	89.5	23,000	193.25	23,060	206.75	22,950	66
25,220	93.5	24,920	198.5	24,980	212.5	24,860	71
27,170	99.5	26,830	203.25	26,900	219	26,770	75.75
29,110	105	28,750	208.25	28,820	229.25	28,680	80.5
31,050	110.75	30,670	213.5	29,780	239.5	30,600	85.75
32,020	113.5	32,580	218.5	30,740	245.5	31,550	88
32,990	116.5	33,540	221.5			32,510	90.75
33,470	118	34,500	224			32,980	92.25
33,960	119.5	35,460	226.5			33,460	93.50
34,440	120.5	36,420	232			33,940	94.5
34,930	122.25	36,900	234			34,420	95.75
35,420	124	37,370	238			34,900	97.25
35,900	125.25					35,380	98.50
36,380	127					35,850	99.5
36,870	128.5					36,330	101
37,350	130.5					36,810	102.5
						37,290	103.5
						37,770	105
						38,250	106.25
						38,720	108.5

7" x 1¾" Bars.

Test No. 829.		Test No. 850.		Test No. 853.		Test No. 867.	
Area (.5178) Load per sq. in.	Heat No. (5,381 B) Average Microm. Reading.	Area (.5204) Load per sq. in.	Heat No. (5,407 B) Average Microm. Reading.	Area (.5217) Load per sq. in.	Heat No. (5,424 B) Average Microm. Reading.	Area (.5204) Load per sq. in.	Heat No. (5,412 B) Average Microm. Reading.
3,860	.0022	3,840	.0054.25	3,830	.0162.25	3,840	.0124.25
5,790	27.25	5,760	59	5,750	167.25	5,760	128.5
7,720	32	7,680	63.75	7,668	172.75	7,680	133
9,650	37.5	9,600	69	9,580	178.75	9,600	138
11,580	42.5	11,520	74	11,500	183.5	11,520	143
13,520	47	13,450	78.5	13,420	188.25	13,450	147.5
15,450	52	15,370	83.25	15,330	193	15,370	152.5
17,380	57	17,290	88	17,250	197.75	17,290	157.5
19,310	62	19,210	93.25	19,170	204	19,210	162.5
21,240	67.75	21,140	98	21,080	209	21,140	167.25
23,170	72.5	23,060	103	23,000	214	23,060	172.5
25,100	77.5	24,980	108.25	24,920	219	24,980	177
27,040	83.25	26,900	112.75	26,830	224	26,900	182
28,970	88.5	28,820	118	28,750	230	28,820	186.75
30,900	93.75	30,740	123	30,670	236	29,780	189.5
31,860	96.5	31,710	126	31,630	241	30,260	190.75
		32,190	127.75	32,110	242.5	30,740	192
		32,660	129	32,580	245.5	31,220	193
		33,150	130	33,060	247.5	31,710	194.75
		33,630	131.5	33,540	250.5	32,190	196
		34,110	133.5	34,020	253.5	32,660	197.125
		34,590	134.75	34,500	257.5	33,150	198.75
		35,070	136	31,980	262	33,630	199.75
		35,550	138			34,110	201
		36,030	140			34,590	202.25
		36,510	143			35,070	203.75
		36,990	147.75			35,550	205
						36,030	206.375
						36,510	207.5

7" x 1⅜" Bars.

Not Annealed.				Annealed.			
Test No. 868.		Test No. 870.		Test No. 887.		Test No. 890.	
Area (.5191) Load per sq. in.	Heat No. (5,394 B) Average Microm. Reading.	Area (.5115) Load per sq. in.	Heat No. (5,278 B) Average Microm. Reading.	Area (.5268) Load per sq. in.	Heat No. (5,381 A) Average Microm. Reading.	Area (.5294) Load per sq. in.	Heat No. (5,426 A) Average Microm. Reading.
3,852	.0022.25	3,910	.0151.25	3,796	.0096.5	3,770	.00945
5,778	27.25	5,860	156	5,690	100.5	5,660	99
7,704	32.25	7,820	160.75	7,590	105.5	7,550	103.5
9,630	37	9,760	166	9,490	110	9,440	108
11,560	42	11,730	171	11,390	114.5	13,330	113
13,480	46.5	13,680	176.25	13,290	119.5	13,220	118
15,410	51.75	15,640	181.5	15,180	124.5	15,110	123
17,340	56.5	17,590	187.25	17,080	129	17,000	127.5
19,260	61.5	19,550	193	18,980	134	18,890	132
21,160	67	21,500	199	20,880	139.5	20,780	137
23,110	71.75	23,460	205.25	22,780	144	22,660	142
25,040	76.75	25,410	212	24,670	149	24,550	147
26,970	81.75	27,370	219.5	26,570	154.5	26,440	152
28,900	87.5	28,350	235	28,470	159	28,330	157.5
30,820	92	28,840	238	30,370	164.5	30,220	162
31,780	95	29,320	243.5	31,320	167	32,110	167
32,750	97.5			32,270	169.25	33,050	169.5
33,710	100.25			33,220	172	34,000	171.75
34,188	101.75			34,170	174.25	34,940	174.5
34,670	103.5			35,110	176.75	35,890	177
35,160	105			36,060	179.5	36,830	180.5
35,640	107			37,010	182	37,780	188
36,120	109.5			37,960	184.25		
36,600	114.75			38,910	187.5		
				39,390	189		
				39,860	190		
				40,340	191.5		
				40,810	192.5		
				41,280	194.5		

7" x 1⅜" Bars.
Annealed.

Test No. 892.		Test No. 895.		Test No. 907.		Test No. 908.	
Area (.5127) Load per sq. in.	Heat No. (5,394 A) Average Microm. Reading.	Area (.5102) Load per sq. in.	Heat No. (5,424 A) Average Microm. Reading.	Area (.5217) Load per sq. in.	Heat No. (5,412 A) Average Microm. Reading.	Area (.5106) Load per sq. in.	Heat No. (5,342 A) Average Microm. Reading.
3,900	.0096.5	3,920	.0110.25	3,830	.0110	3,877	.0107
5,850	100.5	5,880	115	5,750	114.5	5,807	110.75
7,800	105	7,840	120.25	7,668	119.5	7,743	115.5
9,750	110	9,800	124.5	9,580	124.5	9,680	119.5
11,700	115	11,760	135.5	11,500	129.5	11,610	124
13,650	119.5	13,720	130.5	13,420	133.5	13,550	128.5
15,600	124.5	15,680	140.5	15,330	138.5	15,480	133
17,550	129	17,640	145.5	17,250	143	17,420	138
19,500	134.5	19,600	151	19,170	148	19,350	143
21,450	139	21,560	155.5	21,080	153	21,290	147.5
23,400	144	23,520	161.5	23,000	158.5	23,230	153
25,350	149.5	25,480	166.5	24,920	163	25,160	157.5
27,300	154.5	27,440	172	26,830	167.5	27,100	162.5
29,260	159.5	29,400	177	28,750	172.5	29,030	168.5
31,200	164	30,389	181	30,670	177	30,970	172.5
33,160	170	30,870	182.5	32,580	183	32,900	177
34,130	175	31,360	183.5	33,540	185	33,880	180
35,110	179.5	31,850	185.5	34,500	188	34,840	182.5
36,080	186.5	32,340	186.75	35,460	192.5	35,810	185
		32,830	188			36,780	187.5
		33,320	189.75			37,750	191
		33,810	191			38,230	193.5
		34,300	192.5				
		34,750	194				
		35,280	195.5				
		35,770	196.5				
		36,260	199				
		36,750	200.5				
		37,240	202.5				

7" x 1⅜" Bars.
Annealed.

Test No. 915.		Test No. 916.		Test No. 920.	
Area (.5217) Load per sq. in.	Heat No. (5,278 A) Average Microm. Reading.	Area (.5242) Load per sq. in.	Heat No. (5,324 A) Average Microm. Reading.	Area (.5178) Load per sq. in.	Heat No. (5,375 A) Average Microm. Reading.
3,830	.0116.25	3,810	.0128.75	3,860	.0125.5
5,750	121.25	5,720	133.5	5,790	130
7,668	125.5	7,620	138.25	7,720	135
9,580	130.75	9,540	142.75	9,650	139.5
11,500	136	11,450	147.5	11,590	145
13,420	140.5	13,350	152.75	13,520	149.75
15,330	145.5	15,260	157	15,450	155.25
17,250	150.75	17,170	162.5	17,380	160
19,170	156	19,070	167.25	19,310	165.5
21,080	161	20,980	172.5	21,240	170.5
23,000	166.75	22,890	177.25	23,170	176
24,920	172	24,800	182	25,100	181.5
26,830	177	26,710	187.5	27,040	187.5
28,750	182.5	28,610	192.25	28,970	192.5
30,670	188.5	30,520	196.75	30,900	199.5
32,580	198.5	32,430	202.5	32,830	203.5
		34,340	207	33,800	208.5
		35,290	209.75	34,760	217
		36,240	212		
		36,720	214.75		
		37,200	216		

7" x $1\frac{3}{8}$" Bars.

Not Annealed.				Annealed.	
Test No. 1,094.		Test No. 1,096.		Test No. 1,653.	
Area (.5178) Load per sq. in.	Heat No. (5,376 B) Average Microm. Reading.	Area (.5230) Load per sq. in.	Heat No. (5,344 B) Average Microm. Reading.	Area (.5242) Load per sq. in.	Heat No. (5,842 B) Average Microm. Reading.
3.900	.0164	3,820	.0162	3,810	.01635
5,850	168.5	5,736	167	5,720	168
7,800	173	7,650	171.5	7,620	173.25
9,750	178.25	9,560	176.25	9,540	178
11,700	183	11,470	181.5	11,450	183
13.650	188.25	13,380	186.5	13.350	188
15.600	192.5	15,300	190.5	15,260	192.75
17,550	198	17,210	197	17,170	197.5
19,500	203	19,120	201.5	19.070	202.25
21,450	208	21.030	206	20,980	206.75
23,400	212.5	22,950	211	22.890	212.5
25.350	218	24,860	217	24,800	217.75
27,300	223	26,770	222	26,710	222.75
29,250	228	28,680	227	28,610	228
31,200	233.5	30,600	233	30,520	234.5
33,150	238	32,510	239	31,470	241
35,100	243.25	34,420	247.5		
36,080	246.5				
37,050	249				
38,030	252				
39,000	254.5				
39,980	257				
40,470	258.75				
40,950	260				
41,440	261.5				
41,930	263				
42,420	264.5				
42.900	266.5				
43,390	268				
43,880	269.75				
44,370	271.25				
44,850	273.25				

7" x $1\frac{7}{16}$" Bars.

Test No. 697.		Test No. 708.		Test No. 709.		Test No. 716.	
Area (.5319) Load per sq. in.	Heat No. (5,454 B) Average Microm. Reading.	Area (.5001) Load per sq. in.	Heat No. (5,377 B) Average Microm. Reading.	Area (.5307) Load per sq. in.	Heat No. (5,381 B) Average Microm. Reading.	Area (.5077) Load per sq. in.	Heat No. (5,300 B) Average Microm. Reading.
3,760	.0071.25	4,000	.0067	3,768	.0196	3,940	.0097
5,640	76.75	6,000	70.875	5,650	200.5	5,910	101.75
7,520	81	8,000	75.875	7,530	203.75	7,880	106.5
9,400	85	10,000	80.25	9,420	208	9,850	111
11,280	90	12,000	85.25	11,300	212.25	11,820	116.25
13,160	94.125	14,000	90.25	13,190	216.5	13,790	120.75
15,040	98.75	16,000	95.375	15,070	221.125	15,760	125.75
16,920	103.5	18,000	98.25	16,960	225.75	17,730	131
18,800	108.25	20,000	106	18,840	230.375	19,700	135.5
20,680	112.75	22,000	111.75	20,720	236	21,670	141.25
22,560	117.25	24,000	117	22,610	241.5	23,630	146.5
24,440	122.125	26,000	122.5	24,500	248	25,600	152
26.320	127.25	28,000	127.75	26,380	256.5	27,570	157.75
28,200	134	30,000	133	28,260	270.5	29,550	164
30,084	145.75	32,000	138.75	30,150	283	31,510	170.5
32,020	158.375	34,000	144.25	31,090	290		
33,840	186.5	36,000	150.5	31.560	294.5		
34.780	249	37,000	154.5	32,030	299.25		
35,250	263			32.500	305		
35,720	287.5						

7" x $1\frac{7}{16}$" BARS.

TEST No. 718.		TEST No. 720.		TEST No. 724.		TEST No. 747.	
Area (.5166) Load per sq. in.	Heat No. (5,349 B) Average Microm. Reading.	Area (.5191) Load per sq. in.	Heat No. (5,347 B) Average Microm. Reading.	Area (.5153) Load per sq. in.	Heat No. (5,349 B) Average Microm. Reading.	Area (.5160) Load per sq. in.	Heat No. (5,328 A) Average Microm. Reading.
3,877	.0120.5	3,850	.0071.5	3,880	.0180.75	3,870	.0097
5,807	124 25	5,780	75.5	5,820	185.25	5.808	101.75
7,743	129	7,700	79.5	7,760	189.25	7,740	106
9,680	133.5	9,630	83.5	9,700	193.5	9,680	111
11,610	139.25	11.560	87.5	11,640	200	11,610	116.5
13,550	142.25	13,480	92	13,580	202	13,550	121
15,480	147	15,410	97	15,520	206.5	15,480	126
17.420	151.75	17,340	101.25	17,460	209.25	17,420	131.25
19,350	156.25	19,260	106.75	19,450	216.75	19,360	136.5
21,290	161.25	21,190	112.75	21,340	221.75	21,300	141.5
23,230	167.5	23,110	120.75	23,280	226.75	23,230	147
25,160	169.75	25,040	129	25,220	232.75	25,160	152.25
27,100	177	26.970	137.5	27,170	241.5	27,100	157.75
29,030	183.25	28,901	153.75	29,110	253	29,040	163.25
30,970	193.75	29,860	163	30,080	260.5	30,970	171.5
31,940	204 5	30,340	170.5	31,050	270	31,940	179.5
32.400	213.5	30,820	177	31,530	277.5	32,970	192.5
32,900	228.75	31,300	183.5	32,020	285.5	33,200	196
		31,780	192	32,500	295	33,390	199.75
		32,270	202			33,680	202.5
		32,750	212				

7" x $1\frac{7}{16}$" BARS.

TEST No. 761.		TEST No. 763.		TEST No. 826.		TEST No. 828.	
Area (.5178) Load per sq. in.	Heat No. (5,299 A) Average Microm. Reading.	Area (.5204) Load per sq. in.	Heat No. (5,441 A) Average Microm. Reading.	Area (.5229) Load per sq. in.	Heat No. (5,290 B) Average Microm. Reading.	Area (.5229) Load per sq. in.	Heat No. (5,426 B) Average Microm. Reading.
3,860	.0141.75	3,840	.0124	3,820	.0119	3,820	.0046.5
5,790	146	5,760	129.75	5.736	124	5,736	50.5
7,720	151	7,680	135.25	7,650	128.75	7,650	56
9,650	154.75	9,600	140	9,560	137.5	9,560	61.25
11,590	161.25	11,520	144.75	11,470	138.5	11,470	65.5
13,520	166.25	13.450	149.5	13.380	143	13,380	70.25
15,450	171.5	15.370	154	15,300	148.75	15,300	75.25
17,380	176.5	17,290	159	17,210	153.75	17,210	80.25
19,310	181.75	19,210	164.25	19,120	158.5	19,125	89
21,240	187.75	21,140	169.25	21,030	164.25	21,030	95.25
23,170	192.5	23,060	174.5	22,950	169	22,950	100.25
25,110	198.5	24,980	179.5	24,860	174	24.860	105
27,040	204.5	26,900	184.75	26,770	180.25	26,770	107.25
28,970	213.5	28.820	190	28,680	184.75	28,680	115.25
29,930	219.5	30,740	195.5	30,600	190.25	31,550	120
30,420	224	32,660	200.5	32,510	217.5	32,030	122
		33,630	204.25			30.600	124.5
		34,110	205.5			32,980	127.75
		34,590	208.5				
		35,070	211.25				

7" x $1\frac{7}{16}$" Bars.

Not Annealed.				Annealed.			
Test No. 851.		Test No. 865.		Test No. 886.		Test No. 898.	
Area (.5217) Load per sq. in.	Heat No. (5,607 B) Average Microm. Reading.	Area (.5166) Load per sq. in.	Heat No. (5,324 B) Average Microm. Reading.	Area (.5052) Load per sq. in.	Heat No. (5,307 A) Average Microm. Reading.	Area (.5153) Load per sq. in.	Heat No. (5,324 A) Average Microm. Reading.
3,830	.0124.75	3,870	.0147.75	3,960	.0131	3,880	.0218
5,750	129.75	5,800	152.75	5,940	135.25	5,820	223
7,660	134.75	7,740	158	7,920	141	7,760	227.5
9,580	140	9,680	163	9,900	146	9,700	232.5
11,500	145	11,610	168	11,880	151	11,640	238
13,410	149.5	13,550	173.25	13,850	156.5	13,580	241.75
15,330	154.75	15,480	178.25	15,830	161.75	15,520	246.5
17,250	159.75	17,420	187.75	17,810	166.5	17,460	251.5
19,160	164.75	19,350	189	19,790	172	19,400	256.5
21,080	169.75	21,290	124.5	21,770	177.5	21,340	261.5
23,000	175.25	23,270	201	23,750	182.5	23,280	266
24,920	181	25,160	207.75	25,730	187.5	25,230	271.25
26,830	188.75	27,100	214.5	27,710	193	27,170	276.25
28,750	196	29,030	222.5	29,690	198.5	29,110	281.5
29,710	213.75	30,000	227.5	31,670	203.5	31,050	287
30,670	221.5	30,970	236.5	33,650	208.5	32,990	294
31,150	228			34,640	211.5	34,930	324
				35,630	214.5		
				36,620	217.25		
				37,610	221		
				38,600	228		

7" x $1\frac{7}{16}$" Bars.

Annealed.

Test No. 899.		Test No. 902.		Test No. 905.		Test No. 906.	
Area (.5153) Load per sq. in.	Heat No. (5,381 A) Average Microm. Reading.	Area (.5281) Load per sq. in.	Heat No. (5,426 A) Average Microm. Reading.	Area (.5229) Load per sq. in.	Heat No. (5,343 A) Average Microm. Reading.	Area (.5140) Load per sq. in.	Heat No. (5,414 A) Average Microm. Reading.
3,880	.0119	3,780	.0117.5	3,820	.0113.5	3,890	.0134
5,820	124	5,680	121.5	5,776	119	5,830	138.75
7,760	129.5	7,570	127.5	7,650	124	7,780	144
9,700	134	9,470	131.5	7,560	128.5	9,720	149.25
11,640	139	11,360	137	11,470	133.5	11,670	155
13,580	143.5	13,250	141.75	13,380	138.5	13,620	160.5
15,520	149	15,150	146.75	15,300	143.5	15,560	166
17,460	154	17,040	152.5	17,210	148.5	17,511	171.5
19,400	159.25	18,930	157.5	19,120	153.5	19,450	177
21,340	164.5	20,830	163	21,030	159	21,400	182.5
23,280	169.5	22,720	168.5	22,950	164	23,340	189.5
25,230	174.5	24,620	173.75	24,860	169	25,290	194
27,170	179	26,510	179	26,770	174.5	27,240	199.75
29,110	185.5	28,400	184.5	28,680	179	29,180	205.75
31,050	193.5	30,300	190	30,600	184	31,130	211
32,020	198.5	32,190	195.5	32,510	189.5	32,100	214.5
		33,140	198	34,420	194	33,070	217.25
		34,080	201	35,380	197.25	34,050	220
		35,030	203.75	36,330	200.5	35,020	223
		35,980	207	37,290	203	35,500	224.5
		36,920	214.5	38,250	207	35,990	226
						36,480	227.5
						36,960	229
						37,450	232
						37,940	233.5

7" x 1$\frac{1}{16}$" Bars.

Annealed.		Not Annealed.					
Test No. 914.		Test No. 1,100.		Test No. 1,101.		Test No. 1,102.	
Area (.5217) Load per sq. in.	Heat No. (5,290 A) Average Microm. Reading.	Area (.5115) Load per sq. in.	Heat No. (5,328 B) Average Microm. Reading.	Area (.5358) Load per sq. in.	Heat No. (5,441 B) Average Microm. Reading.	Area (.5178) Load per sq. in.	Heat No. (5,499 B) Average Microm. Reading.
3,830	.0120.5	3,910	.0110.5	3,730	.0150.5	3,860	.0170
5,750	125	5,860	114 75	5,600	155	5,790	174. 5
7,660	129.5	7.820	119.5	7,460	160	7,720	179
9,580	134	9,770	124.5	9,330	164.5	9,650	184.25
11,500	138.75	11,730	129.5	11,200	169.25	11,590	190
13,410	143.5	13.680	134.5	13,060	174	13,570	194
15,330	148	15,640	139.5	14,930	179	15,450	198
17,250	153	17,600	145	16,800	184.5	17,380	209
19,160	157.5	19,550	149.5	18,660	189	19,310	204
21,080	161.75	21,500	154.5	20,530	194.5	21,240	214.25
23,000	166.5	23,460	160	22,400	200	23,170	219
24,910	172	25,420	165	24,260	205.25	25,100	224
26,830	178.5	27.370	170.5	26,130	211	27,040	228.25
28,750	180.5	29,330	175.25	28,000	216.75	28,970	235
		31,280	180.75	29,860	223.5	30,900	240.25
		33,[illegible]40	186	31,730	230.5	32,830	245.5
		34,210	189.25	32,660	234.75	33,800	249.5
		35,180	192	33,600	239.5		
		36,170	195.75				

7" x 1$\frac{1}{16}$" Bars.

Annealed.

Test No. 1,648.		Test No. 1,654.		Test No. 1,655.	
Area (.5217) Load per sq. in.	Heat No. (5,454 A) Average Microm. Reading.	Area (.5242) Load per sq. in.	Heat No. (5,300 A) Average Microm. Reading.	Area (.5306) Load per sq. in.	Heat No. (5,349 A) Average Microm. Reading.
3,830	.0198.25	3,810	.0201.5	3,770	.0150.5
5,750	203.25	5,720	206.25	5.650	155.25
7,660	208	7.630	211.5	7,530	160.25
9,580	213.5	9,540	216.25	9,420	165.5
11,500	218.75	11,450	221.25	11,310	170.25
13,410	223.5	13,350	226.5	13,190	175.25
15,330	228.5	15.260	231.75	15,070	180
17,250	233.5	17.170	237.25	16.960	185.25
19,160	238.5	19,070	242.5	18,840	190
21,080	243.75	20,980	247.5	20.730	195.25
23,000	248.25	22,890	252.25	22,610	200
24,920	253	24,800	257.75	24,500	205.25
26,830	259	26,710	262.75	26,380	210.25
28,750	263.25	28,610	268.5	28,770	215.25
30,670	267.5	30,520	275	30,150	220.5
32,580	273.5	31,470	291	32,040	225.25
33,540	281			32,980	228
				33,920	231.5
				34,390	233
				34.861	237.5

7" x 1½" BARS.

Not Annealed.		Annealed.		Not Annealed.	
Test No. 792.		Test No. 1,646.		Test No. 776.	
Area (.5191) Load per sq. in.	Heat No. (5,406 B) Average Microm. Reading.	Area (.5345) Load per sq. in.	Heat No. (5,406 A) Average Microm. Reading.	Area (.5178) Load per sq. in.	Heat No. (5,383 B) Average Microm. Reading.
3,850	.0126.25	3,740	.0161	3,860	2.0137.5
5,790	130.5	5,610	165.5	5,790	141.5
7,700	135.25	7,480	170	7,720	147
9,630	140.25	9,350	173.75	9,650	151
11,560	145	11,220	178	11,590	156
13,480	149.75	13,100	183.25	13,570	160.25
15,410	155	14,970	189.5	15,450	165
17,340	159.5	16,840	194.5	17,340	170
19,260	164.75	18,710	199	19,310	174.75
21,190	169.5	20,580	203.5	21,240	180
23,110	174.75	22,450	207.5	23,170	185.25
25,040	179.75	24,320	213.5	25,100	190.25
26,970	185	26,190	217.5	27,040	195.25
28,900	190	28,060	222.5	28,970	200.75
29,860	192.5	29,930	228	30,900	206.5
30,820	195	31,810	233	32,830	212.25
31,780	197.5	33,680	238.5	33,800	215.5
32,750	200.25	34,610	242	34,280	217.5
33,710	203.25	35,080	246	34,760	222
34,190	205.5				
34,670	209.5				
35,160	515.5				

7" x $\frac{11}{16}$" BARS.

Test No. 790.		Test No. 791.		Test No. 827.	
Area (.5191) Load per sq. in.	Heat No. (5,412 B) Average Microm. Reading.	Area (.5191) Load per sq. in.	Heat No. (5,146 B) Average Microm. Reading.	Area (.5178) Load per sq. in.	Heat No. (5,347 B) Average Microm. Reading.
3,852	.0148.5	3,852	.0138	3,860	.0123.75
5,778	153	5,778	142	5,790	128.25
7,704	158.25	7,704	146.75	7,720	132.5
9,630	162.5	9,630	151.5	9,650	137.25
11,560	167	11,560	156	11,590	142.25
13,480	172	13,480	161	13,570	146.5
15,410	177	15,410	164.5	15,450	151.25
17,340	181.75	17,340	170	17,380	136
19,260	186.5	19,260	175.25	19,310	160.75
21,160	192	21,160	180	21,240	165.5
23,110	197.25	23,110	184.75	23,170	171
25,040	203	25,040	189.5	25,100	176
26,970	208.75	26,970	194.75	27,040	183
28,900	216	28,900	202.25	28,970	188.5
30,820	231	30,820	216	30,900	197
32,750	247.5	31,300	218.5	31,880	202
		31,780	224.5		

7″ x $1\frac{1}{8}$″ Bars.

Test No. 830.		Test No. 840.		Test No. 863.		Test No. 866.	
Area (.5204) Load per sq. in.	Heat No. (5,411 B) Average Microm. Reading.	Area (.5217) Load per sq. in.	Heat No. (5,426 B) Average Microm. Reading.	Area (.5242) Load per sq. in.	Heat No. (5,454 B) Average Microm. Reading.	Area (.5077) Load per sq. in.	Heat No. (5,406 B) Average Microm. Reading.
3,840	.0135.5	3,830	.0140	3,810	.0109.25	3,940	.0121
5,760	140	5,750	144	5,720	113.75	5,910	125.5
7,680	144.75	7,668	148.5	7,620	118.5	7,880	130.5
9,600	149	9,580	153	9,540	123.5	9,850	135.5
11,520	154.5	11,500	157.5	11,450	129	11,820	140
13,450	159	13,420	162	13,350	133	13,790	145.25
15,370	163.75	15,330	166.5	15,260	137.25	15,750	149.75
17,290	168.5	17,250	171.75	17,170	142.5	17,720	154.5
19,210	173.5	19,170	176.5	19,070	146.5	19,700	159.5
21,140	178.5	21,080	182	20,980	151.75	21,660	164.75
23,060	184	23,000	187.5	22,890	156.5	23,630	169.75
24,980	190	24,920	193.5	24,800	161.5	25,600	174.75
26,900	195.75	26,830	201.5	26,710	166	27,570	179.75
28,820	202.5	28,750	210.75	28,610	171.25	29,540	185
30,740	214	29,800	245	30,520	176.25	31,510	190.5
31,710	216.75			32,430	181.5	32,010	191.5
32,660	221.5			33,380	184.5	32,500	193
33,630	225.5			33,860	186	32,990	194.5
				34,340	187.25	33,480	195.75
				34,810	189.25	33,980	197.25
				35,290	191	34,470	198.75
				35,770	191.75	34,960	200
				36,240	194	35,450	201.75
				36,720	197	35,950	210
				37,200	206		

7″ x $1\frac{1}{8}$″ Bars.

Not Annealed.		Annealed.					
Test No. 869.		Test No. 883.		Test No. 891.		Test No. 893.	
Area (.5217) Load per sq. in.	Heat No. (5,357 B) Average Microm. Reading.	Area (.5140) Load per sq. in.	Heat No. (5,406 A) Average Microm. Reading.	Area (.5204) Load per sq. in.	Heat No. (5,374 A) Average Microm. Reading.	Area (.5191) Load per sq. in.	Heat No. (5,357 A) Average Microm. Reading.
3,830	.0163.75	3,890	.0113	3,840	.0110.5	3,850	.0088.5
5,750	168	5,830	118	5,760	113.5	5,780	93
7,668	173.25	7,780	122.5	7,680	118.5	7,700	97.5
9,580	178.25	9,720	127.5	9,600	123	9,630	102.25
11,500	183.25	11,670	132.5	11,520	128.5	11,560	107.5
13,420	188.25	13,620	137	13,450	133	13,480	112
15,330	193	15,560	142	15,370	138.5	15,410	117
17,250	198.25	17,510	146.75	17,290	143.5	17,340	122
19,170	203	19,450	151.5	19,210	148.5	19,260	127
21,080	207.75	21,400	157	21,140	153.75	21,190	131
23,000	213.25	23,340	161.5	23,060	159.5	23,110	136.5
24,920	218.5	25,290	166.5	24,980	163.5	25,040	141.75
26,830	223.5	27,240	172.5	26,900	169	26,970	146
28,750	228.5	29,180	177.5	28,820	174.5	28,900	151.5
30,670	233.75	31,130	182	30,740	179.5	30,820	156.25
31,630	236.5	33,270	187	32,660	184.5	32,750	161.25
32,580	239.5	34,050	190.5	34,590	189.5	34,670	166.75
33,540	242.5	35,020	193.25	36,510	195	35,640	169.5
34,500	245.5	35,990	196.5	38,430	200.5	36,600	172
35,460	254.5	36,960	200.5	39,390	203	37,560	175.5
						38,530	179
						39,010	183.5

7″ x 1 1/8″ Bars.

Annealed.

Test No. 894.		Test No. 902*a*.		Test No. 904.		Test No. 909.	
Area (.5242) Load per sq. in.	Heat No. (5,411 A) Average Microm. Reading.	Area (.5166) Load per sq. in.	Heat No. (5,412 A) Average Microm. Reading.	Area (.5204) Load per sq. in.	Heat No. (5,454 A) Average Microm. Reading.	Area (.5242) Load per sq. in.	Heat No. (5,426 A) Average Microm. Reading.
3,810	.0136.5	3,870	.0107	3,840	.0101.75	3,810	.0110.5
5,720	140.5	5,810	111.25	5,760	105.5	5,720	115.5
7,620	145.5	7,740	116	7,680	109.5	7,630	120.25
9,540	151	9,680	121	9,600	114.5	9,540	125
11,450	156	11,610	125.5	11,520	119	11,450	130
13,350	160.5	13,550	130.5	13,450	123.5	13,350	134.5
15,260	165	15,480	135.5	15,370	128.5	15,260	139
17,170	170	17,420	140.75	17,290	133.5	17,170	144.5
19,070	175.5	19,350	146	19,210	137.5	19,070	148.5
20,980	180.5	21,290	151	21,140	142.5	20,980	154
22,890	185	23,230	156	23,060	147.5	22,890	159
24,800	190.5	25,160	161	24,980	152.5	24,800	163.5
26,710	195.5	27,100	166.5	26,900	157.5	26,710	169
28,610	200	29,030	172	28,820	162.25	28,610	173.75
30,520	205	30,960	177.25	30,740	167.5	30,520	178.75
32,430	210	32,900	183	32,660	172.5	32,430	183.5
34,340	215	33,870	185.75	33,630	175	34,340	190.5
35,290	218	34,840	188	34,590	178	34,810	191.5
36,240	221	35,810	191.5	35,550	182		
37,200	223	36,780	194	36,510	185.5		
38,150	225.75	37,750	198	36,990	187.5		
39,100	228	38,230	200.5	37,470	190.5		
				37,950	194		

7″ x 1 1/8″ Bars.

Annealed.

Test No. 910.		Test No. 912.		Test No. 921.	
Area (.5242) Load per sq. in.	Heat No. (5,446 A) Average Microm. Reading.	Area (.5153) Load per sq. in.	Heat No. (5,347 A) Average Microm. Reading.	Area (.5153) Load per sq. in.	Heat No. (5,383 A) Average Microm. Reading.
3,810	.0109	3,880	.0123.75	3,880	.0124.5
5,720	113.5	5,820	128.5	5,820	129
7,630	118.5	7,760	133.5	7,760	133.5
9,540	124	9,700	138.75	9,700	138
11,450	128.5	11,640	144	11,640	142.5
13,350	133.5	13,580	149	13,580	147.25
15,260	138.5	15,520	154	15,520	151.5
17,170	143.5	17,460	158.75	17,460	156.5
19,070	148.5	19,400	164	19,400	161.5
20,980	153.5	21,340	169	21,350	166.5
22,890	159	23,280	174	23,280	171
24,800	164	25,230	128.5	25,230	176.5
26,710	169.5	27,170	184.5	27,170	181
28,610	175.25	29,110	189.75	29,110	185.75
30,520	180.5	31,050	196	31,050	191
32,430	186	32,990	201.5	32,990	196.5
33,380	189	34,930	209	33,960	199.5
34,340	191.75	35,420	212.25	34,930	202.5
35,290	194.5			35,900	210
36,240	197.5				
37,200	200.5				

7″ x $1\frac{1}{8}$″ Bars.

Not Annealed.

Test No. 1,092.				Test No. 1,093.			
Area (.5242) Load per sq. in.	Heat No. (5,374 B) Average Microm. Reading.	Area (.5242) Load per sq. in.	Heat No. (5,374 B) Average Microm. Reading.	Area (.5115) Load per sq. in.	Heat No. (5,357 B) Average Microm. Reading.	Area (.5115) Load per sq. in.	Heat No. (5,357 B) Average Microm. Reading.
3,810	.0157.5	20,980	201.5	3,910	.0040.25	21.510	87.75
5,720	162	22,890	206.5	5,860	46.5	23,460	93
7,630	166.75	24,800	212	7,820	52.25	25,420	97.5
9,540	171.75	26,710	218	9,770	57.75	27,370	101.5
11,450	176.5	28.610	223.75	11,730	63	29,330	106.5
13,350	181.5	30,520	230.5	13,680	68	31,280	111.5
15,260	186.5	32,430	268	15,640	73	33.240	116.5
17,170	190.75			17,590	78.5	35,190	121.5
19,070	196.5			19,550	83	36,170	126

7″ x $1\frac{3}{4}$″ Bars.

Not Annealed.				Annealed.			
Test No. 696.				Test No. 896.			
Area (.5128) Load per sq. in.	Heat No. (5,447 B) Average Microm. Reading.	Area (.5128) Load per sq. in.	Heat No. (5,447 B) Average Microm. Reading.	Area (.5153) Load per sq. in.	Heat No. (5,447 A) Average Microm. Reading.	Area (.5153) Load per sq. in.	Heat No. (5,447 A) Average Microm. Reading.
3,860	.0108.25	33,800	184.75	3,880	.0163.5	32.990	243.5
5,790	113	34,280	186.25	5,820	167.5	33.960	248
7,720	117.25	34,760	187.5	7.760	172.5	34,930	255.5
9,650	122.5	35,240	189	9,700	177.25		
11,590	127.75	35.730	190.25	11.640	182.5		
13,570	132.75	36,210	190.75	13,580	187.25		
15,450	137.875	36,690	192.75	15,520	192.5		
17,340	141 75	37,170	194	17,460	198.0		
19,310	146.75	37.660	195.5	19,400	202.0		
21.240	151.75	38.140	197	21,350	208.5		
23,170	156.5	38.620	198.25	23.280	215.25		
25.100	161.5	39.110	109.5	25.230	220.5		
27,040	166.5	39,590	201.5	27,170	225.5		
28,970	171.25	40.070	202.5	29,110	230.25		
30,900	177	40,560	204.25	31,050	237		
32.830	182			33,020	240.5		

7″ x 1$\frac{3}{16}$″ Bars.

Test No. 719.		Test No. 744.		Test No. 745.		Test No. 751.	
Area (.5166) Load per sq. in.	Heat No. (5,424 B) Average Microm. Reading.	Area (.5153) Load per sq. in.	Heat No. (5,311 A) Average Microm. Reading.	Area (.5140) Load per sq. in.	Heat No. (5,302 A) Average Microm. Reading.	Area (.5294) Load per sq. in.	Heat No. (5,405 A) Average Microm. Reading.
3,870	.0199.25	3,880	.0206	3,890	.0192.5	3,780	.0110.75
5,810	203.875	5,820	211.5	5,830	197.75	5,670	115
7,740	208.5	7,760	215.5	7,780	202.25	7,550	119.75
9,680	213.5	9,700	221	9,720	208	9,440	124.75
11,610	217.5	11,640	225.75	11,670	211.75	11,330	129.5
13,550	223.25	13,580	231	13,620	216.5	13,220	134
15,480	228	15,520	235.25	15,560	221.75	15,110	139
17,420	233.25	17,460	241.5	17,510	226.75	17,000	144
19,350	237.25	19,400	246	19,450	231.75	18,890	149
21,290	242	21,350	251.5	21,400	237	20,780	154
23,230	247	23,280	256.25	23,340	242	22,660	159
25,160	252	25,230	261.5	25,290	248	24,550	164
27,100	257.75	27,170	267	27,240	253.5	26,440	168.5
28,080	261	29,110	273	29,180	259	28,330	174.25
29,030	264.25	31,050	278.5	31,130	264	30,220	179.75
30,000	269.75	32,990	287	31,610	265.5	31,160	182.5
30,970	277.5	33,960	311	32,100	268	32,110	185.5
31,460	282.25			32,590	270	32,580	187.25
31,940	286.75			33,070	272.5	33,050	191
32,420	291			33,560	275.5		
				34,050	279.25		
				34,580	284.25		

7″ x 1$\frac{3}{16}$″ Bars.

Not Annealed.				Annealed.	
Test No. 752.		Test No. 753.		Test No. 1,650.	
Area (.5268) Load per sq. in.	Heat No. (5,405 A) Average Microm. Reading.	Area (.5242) Load per sq. in.	Heat No. (5,375 A) Average Microm. Reading.	Area (.5294) Load per sq. in.	Heat No. (5,424 A) Average Microm. Reading.
3,800	.0144.25	3,810	.0108.75	3,780	.0188.5
5,690	149	5,720	112.75	5,670	193.25
7,590	153.75	7,630	117.75	7,550	198
9,490	158.25	9,540	122	9,940	204
11,390	163.5	11,450	126.75	11,330	208.5
13,290	168.5	13,350	131.25	13,220	213.5
15,180	173.25	15,260	136	15,110	218
17,080	178	17,170	140.625	17,000	223.75
18,980	183	19,070	145.5	18,890	228.5
20,880	188	20,980	150.5	20,780	233.25
22,780	194	22,890	155.5	22,670	238.25
24,680	198.5	24,800	160	24,560	243
26,580	203.25	26,710	165	26,440	247.75
28,470	208.5	28,610	169.75	28,330	253.5
30,370	214.5	30,520	174.75	30,220	259.75
31,320	217	31,470	177	32,110	264.5
31,800	218.75	32,430	179.5		
32,170	220.5	32,900	181		
32,740	221.75	33,380	182.25		
33,220	223.5	33,670	182.75		
33,690	225	33,860	183.25		
34,170	227.25	34,340	184.5		
34,640	228.5	34,810	186		
35,120	232.5	35,290	187		
		35,770	188.25		
		36,240	189.5		
		36,720	191		
		37,200	192.25		
		37,670	193.5		
		37,960	195		
		38,150	195.75		
		38,630	197.5		
		39,100	199.5		

7″ x 1⅞″ Bars.

Not Annealed. Test No. 713.				Annealed. Test No. 1,649.			
Area (.5191) Load per sq. in.	Heat No. (5,378 B) Average Microm. Reading.	Area (.5191) Load per sq. in.	Heat No. (5,378 B) Average Microm. Reading.	Area (.5345) Load per sq. in.	Heat No. (5,378 A) Average Microm. Reading.	Area (.5345) Load per sq. in.	Heat No. (5,378 A) Average Microm. Reading.
3,850	.0142	21,190	186.75	3,740	.0158.5	20,580	201.75
5,780	146.5	23,110	193.25	5,610	163.25	22,450	206.5
7,700	150.75	25,040	200.5	7,480	168.25	24,320	211.25
9,630	156.25	26,970	209.5	9,350	173	26,190	216
11,560	161.25	28,900	219	11,220	177.5	28,060	221.25
13,480	165.25	30,820	234	13,100	182	29,930	226.75
15,410	170.25	31,780	246.75	14,970	187.5	31,810	232.75
17,340	175.75	32,270	258.25	16,840	192.5	33,680	240
19,260	180.75	32,750	270	18,710	197	34,610	249

24″ x 1⅜″ Plate.

Not Annealed. Test No. 971.				Annealed. Test No. 1,075.			
Area (.5319) Load per sq. in.	Heat No. (5,484 B) Average Microm. Reading.	Area (.5319) Load per sq. in.	Heat No. (5,484 B) Average Microm. Reading.	Area (.5506) Load per sq. in.	Heat No. (5,484 A) Average Microm. Reading.	Area (.5506) Load per sq. in.	Heat No. (5,484 A) Average Microm. Reading.
3,760	.0078.5	30,080	149	3,680	.0165.75	29,060	237.5
5.640	84	31.960	153.5	5.450	170.5	30,870	242
7.520	88.25	33.840	159	7,260	175.5	32.690	247
9,400	93.25	35,720	164.5	9,080	180	34.510	252
11,280	98	37,600	169	10,900	184.75	36,320	256.75
13.160	102.5	39,480	174.25	12,710	189.5	38,140	261
15.040	108.25	40,420	177.5	14,530	200.5	39.050	263.5
16.920	113.5	41,360	180	16,350	205	39.960	266
18.800	118.5	42.300	182.25	18,160	209.5	40,870	267.5
20.680	123.75	43,240	186	19,980	214	41,770	269.5
22.560	129	44,180	188.5	21,790	219		
24.440	134	45,120	190.5	23,610	223.5		
26,320	138.75	45,590	192.5	25.430	228		
28,200	144			27,240	233		

22½" x ¾" PLATE.

Not Annealed.				Annealed.			
Test No. 967.				Test No. 1,071.			
Area (.5070) Load per sq. in.	Heat No. (5,910 B) Average Microm. Reading.	Area (.5070) Load per sq. in.	Heat No. (5,910 B) Average Microm. Reading.	Area (.5168) Load per sq. in.	Heat No. (5,910 A) Average Microm. Reading.	Area (.5168) Load per sq. in.	Heat No. (5,910 A) Average Microm. Reading.
3,940	.0088	27,610	147	3,870	.0163	27,090	223
5,920	92	29,580	152	5,800	168	29,020	228
7,890	97.5	31,560	157	7,740	173	30,960	233.25
9,860	102.5	33,530	162	9,670	179	32,890	238.5
11,830	106.5	35,500	167.5	11,610	183	34,830	243.5
13,810	111.75	37,470	173	13,540	188.5	36,760	249.5
15,780	116.5	39,450	177.75	15,480	193		
17,750	122	41,420	182.5	17,410	198		
19,720	126.75	42,410	185.5	19,350	203		
21,700	132	42,900	187	21,280	208.5		
23,670	136.75	43,400	191.5	23,220	213.25		
25,640	141.5			25,150	218		

20" x ¾" PLATE.

Test No. 964.		Test No. 986.		Test No. 993.		Test No. 997.	
Area (.4848) Load per sq. in.	Heat No. (5,866 B) Average Microm. Reading.	Area (.5106) Load per sq. in.	Heat No. (5,568 B) Average Microm. Reading.	Area (.5066) Load per sq. in.	Heat No. (5,540 B) Average Microm. Reading.	Area (.5043) Load per sq. in.	Heat No. (5,530 B) Average Microm. Reading.
4,125	.0147.25	3,920	.0060	3,950	.0074.75	3,960	.0070.75
6,190	152.5	5,870	645	5,920	795	5,950	75
8,250	157.75	7,830	692.5	7,890	845	7,930	80.5
10,310	163	9,790	745	9,870	89	9,910	85.5
12,380	168	11,750	795	11,840	94	11,900	90.5
14,440	173.5	13,710	85	13,820	995	13,880	95.5
16,500	178.5	15,680	905	15,790	104	15,860	100.5
18,560	184	17,620	95	17,760	109	17,840	105.25
20,620	189	19,580	100	19,740	114.5	19,820	111
22,690	194.5	21,540	104.5	21,710	119.25	21,810	116
24,750	200	23,500	110	23,690	124.5	23,800	121
26,820	205.5	25,460	115.5	25,660	129.75	25,780	125.5
28,880	211	27,420	120.5	27,630	134.75	27,760	131.25
30,940	216.5	29,380	125.5	29,610	140	29,740	136.25
33,000	221.5	31,330	130	31,580	145.25	31,720	142
35,070	227.5	33 290	135.5	33,560	150.5	33,710	146.5
37,130	232.75	35,250	140.25	35,530	155.5	35,690	152
39,190	238	37,210	145.5	37,510	161	37,680	157
41,250	243.5	39,170	151	39,470	166.5	39,660	162.25
		41,130	156	41,460	171	41,640	167.5
		43,080	161.5	42,440	173.5	43,620	172.5
		44,060	164.5	43,430	176.5	44,620	175.5
		45,050	167	44,420	179	45,610	178.5
		46,030	169.5			46,600	181.25

20″ x ¾″ PLATE.
ANNEALED.

Test No. 1,077.		Test No. 1,080.		Test No. 1,106.		Test No. 1,109.	
Area (.5185) Load per sq. in.	Heat No. (5,530 A) Average Microm. Reading.	Area (.4693) Load per sq. in.	Heat No. (5,540 A) Average Microm. Reading.	Area (.5015) Load per sq. in.	Heat No. (5,806) Average Microm. Reading.	Area (.4995) Load per sq. in.	Heat No. (5,568) Average Microm. Reading.
3,850	.0040	4,260	0194.5	3,990	.0159	4,000	.0133
5,780	455	6,390	200.75	5,980	164.25	6,000	137
7,710	507.5	8,520	207	7,979	169	8,000	142
9,640	555	10,650	212	9,970	173.5	10,000	147.5
11,560	602.5	12,780	217	11,960	178.5	12,000	152.5
13,490	655	14,910	223	13,960	183	14,000	158
15,420	705	17,050	228.5	15,950	188.5	16,000	162.5
17,350	75	19,180	233.5	17,940	194	18,000	168
19,270	80	21,310	239	19,940	199	20,000	173
21,200	855	23,440	245	21,930	204.5	22,000	178
23,130	902.5	25,570	250	23,930	208.5	24,000	184
25,060	955	27,700	255.5	25,920	213.5	26,000	189
26,980	100.5	29,830	261	27,910	218.5	28,000	194
28,910	105.5	31,960	266.5	29,910	224	30,000	199.25
30,840	110.5	34,090	272	31,900	228.75	32,000	204.5
32,760	116	35,160	275.5	33,900	234.5	34,000	210
34,700	121.5	36,220	278	35,890	240	35,000	213
36,620	126.5			36,890	242	36,000	216
						37,000	219

20″ x ¾″ PLATE.
ANNEALED.

Test No. 1,656				Test No. 1,657.			
Area (.4638) Load per sq. in.	Heat No. (5,530) Average Microm. Reading.	Area (.4638) Load per sq. in.	Heat No. (5,530) Average Microm. Reading.	Area (.4646) Load per sq. in.	Heat No. (5,568) Average Microm. Reading.	Area (.4646) Load per sq. in.	Heat No. (5.568) Average Microm. Reading.
4.310	.0050.75	25,870	106	4,330	.0196.25	26,000	252
6,470	56	28,030	111	6,500	202	28,160	257.75
8.620	61	30,180	116.25	8,660	207.25	30,330	263.5
10,780	66.75	32,340	121.75	10,830	212.75	32,500	269
12.940	71 75	34.500	127	13,000	218.5	34,660	274.25
15,090	77.5	36.650	133	15.160	224.25	36,830	280
17,250	82.75	38,810	138	17.330	229.75	39,000	286
19.400	88	40.970	144.5	19,500	235.25	41,160	291.25
21,560	93.75	43,120	150.5	21,660	240.75	42,240	293.75
23,710	99			23,830	247		

24" x ⅝" PLATE.

Test No. 965.		Test No. 968.		Test No. 998.		Test No. 1,076.	
Area (.4843) Load per sq. in.	Heat No. (5,863 B) Average Microm. Reading.	Area (.4800) Load per sq. in.	Heat No. (5,494 B) Average Microm. Reading.	Area (.5108) Load per sq. in.	Heat No. (5,484 B) Average Microm. Reading.	Area (.4985) Load per sq. in.	Heat No. (5,484 A) Average Microm. Reading.
4,130	.0106	4,160	.0210	3,910	.0176	4,010	.0169.5
6,190	111.75	6,250	215	5,870	180.75	6,020	174.5
8,260	117	8,330	220	7,830	186.5	8,020	180
10,320	122.5	10,410	225.5	9,790	190	10,030	184.5
12,390	128.5	12,500	231	11,740	195.75	12,030	190
14,450	133.25	14,580	236.25	13,700	199.5	14,040	196
16,520	138.25	16,660	241	15,660	205.5	16,040	201
18,580	144	18,750	246.5	17,620	210	18,050	207
20,650	149	20,830	252	19,600	215	20,050	212.5
22,710	154.5	22,920	257.5	21,530	220	22,060	217.5
24,780	160.5	25,000	264	23,490	225.25	24,070	222.5
26,840	164.5	27,080	268.5	25,450	230.25	26,070	227
28,910	170.5	29,170	273	27,410	245.5	28,080	232
30,970	175.5	30,200	276.5	29,360	240.75	30,090	237
33,040	181	31,250	278.5	31,320	246	32,100	242.5
35,100	186.5	32,290	281.5	33,280	252	34,100	247.5
37,170	191.5	33,330	284	35,240	256.25	35,100	250.5
39,230	197.5	34,370	287	37,200	261.75	36,110	253
41,300	203	35,470	289.5	39,150	266.5	37,110	256
42,330	206	36,460	292.5	41,110	272.25	38,120	258.5
43,360	208	37,500	295	43,070	277.5		
44,440	211.5	38,540	298	43,560	279		
45,430	214	39,580	300.75				
46,460	217	40,620	303.5				
47,490	221	41,670	306				
		42,710	309				
		43,750	312				
		44,790	315				
		45,830	320.75				

24" x ⅝" PLATE.

ANNEALED.

Test No. 1,079.		Test No. 1,090.		Test No. 1,660.	
Area (.4752) Load per sq. in.	Heat No. (5,494 A) Average Microm. Reading.	Area (.4806) Load per sq. in.	Heat No. (5,863 A) Average Microm. Reading.	Area (.5078) Load per sq. in.	Heat No. (5,494) Average Microm. Reading.
4,210	.0156	4,160	.0141	3,940	.0049
6,310	161.5	6,240	147	5,910	54
8,420	167.5	8,320	153	7,880	59
10,520	173	10,400	158	9,850	64
12,630	178.5	12,480	163	11,820	69
14,730	184	14,560	168	13,790	74
16,840	189	16,650	174	15,760	79.25
18,940	194.5	18,730	179.5	17,730	84.25
21,040	200	20,810	184	19,700	89.5
23,150	206	22,890	189	21,670	94.75
25,260	212	24,970	195	23,640	100
27,360	216.5	27,050	200	25,610	105
29,470	222.5	29,130	205	27,580	110.5
31,570	228	31,210	210.5	29,550	115.5
33,680	231.5	33,300	216	31,520	120.5
35,780	239.5	35,370	221.5	33,490	126
36,830	242.5	36,420	224.5	35,460	130.75
37,880	247	37,460	227	37,430	136
		38,500	229.5	39,400	141.25
		39,540	232.5	41,370	147
				42,360	149.75
				43,340	152.75
				44,330	155
				45,310	159

22″ x ⅝″ Plate.

Not Annealed.		Annealed.			
Test No. 988.		Test No. 1,073.		Test No. 1,659.	
Area (.5002) Load per sq. in.	Heat No. (5,566 B) Average Microm. Reading.	Area (.4913) Load per sq. in.	Heat No. (5,566) Average Microm. Reading.	Area (.5063) Load per sq. in.	Heat No. (5,566) Average Microm. Reading.
4,000	.0052.25	4,070	.0037.5	3,950	75.25
6,000	57	6,100	42.5	5,930	80.5
8,000	62	8,140	47.5	7,900	85.25
10,000	67	10,180	53	9,870	90.25
12,000	72.5	12,210	58.5	11,850	95.5
14,000	77	14,250	63.75	13,820	101.25
16,000	82.5	16,280	69	15,800	106.25
18,000	87.5	18,320	74.5	17,770	111.25
20,000	92.25	20,350	80	19,750	116.5
22,000	97.5	22,390	85.5	21,720	122
24,000	103.5	24,420	92	23,700	127.25
26,000	107.5	26,460	96.5	25,670	132.25
28,000	113	28,490	101.5	27,650	137.5
30,000	118.5	30,530	107.5	29,620	142.75
32,000	123.5	31,550	110.5	31,600	147.75
34,000	129.5			33,580	153.75
36,000	134			35,550	158.25
38,000	139.75			37,520	163.25
40,000	144.5			39,500	169.75
42,000	149.75			40,490	173.5
44,000	155.5				

18″ x ¾″ Plate.

Not Annealed.				Annealed.	
Test No. 979.		Test No. 980.		Test No. 1,107.	
Area (.4977) Load per sq. in.	Heat No. 5,419) Average Microm. Reading.	Area (.5050) Load per sq. in.	Heat No. (5,472) Average Microm. Reading.	Area (.4665) Load per sq. in.	Heat No. (5,419) Average Microm. Reading.
4,020	.0194.5	3,960	.0177.5	4,280	.0126
6,030	200	5,940	181.5	6,430	131.5
8,040	205.25	7,920	186.5	8,570	137
10,050	210.5	9,900	191.5	10,710	143.25
12,050	216.5	11,880	196.5	12,850	149
14,060	221.5	13,860	202	15,000	155
16,070	227.5	15,840	207	17,140	161
18,080	232.5	17,820	212.5	19,280	166.5
20,090	238	19,800	217.5	21,420	172
22,100	243.5	21,780	223	23,560	178.5
24,110	249	23,760	228	25,710	184.5
26,120	254.5	25,740	233.5	27,850	190.5
28,130	260	27,720	238.5	29,990	196
30,140	265.5	29,700	245	32,130	202
32,150	272	31,680	249	33,210	204.75
34,160	277.5	33,660	254.5	34,280	208
36,170	283	35,640	260	35,350	110.5
38,180	289.25	37,620	265	36,420	214
40,190	294.5	39,600	273	37,490	216.5
42,200	301	41,580	277	38,560	219.5
44,200	306.5	43,560	281.5		
45,210	309.5	45,540	288		
46,210	312	46,530	291.5		
47,220	314	47,520	295		

18" x ⅝" Plate.

Annealed.

Test No. 1,108.				Test No. 1,658.			
Area (.4907) Load per sq. in.	Heat No. (5.472) Average Microm. Reading.	Area (.4907) Load per sq. in.	Heat No. (5,472) Average Microm. Reading.	Area (.4552) Load per sq. in.	Heat No. (5,419) Average Microm. Reading.	Area (.4552) Load per sq. in.	Heat No. (5,419 A) Average Microm. Reading.
4,070	.0176	24,450	229	4,390	.0170.5	26,360	228
6,110	181	26,490	234.5	6,590	176.25	28,560	234
8,150	186.5	28,530	240	8,790	182.0	30,760	240
10,190	191.5	30,560	246	10,990	187.5	32,950	246
12,220	196.5	32,600	251.5	13,180	193	35,150	251.75
14,260	202	34,640	256.5	15,380	199.25	37,350	257.5
16,300	202.5	36,680	262	17,570	205	39,550	263.25
18,340	213	37,700	265	19,770	210.75	41,740	269
20,370	219	38,720	268	21,970	216.5	43,940	275
22,410	224	39,730	270.5	24,170	222.5	45,040	280.5

24" x ½" Plate.

Test No. 963.		Test No. 966.		Test No. 994.		Test No. 996.	
Area (.4668) Load per sq. in.	Heat No. (5,444) Average Microm. Reading.	Area (.4975) Load per sq. in.	Heat No. (5,483) Average Microm. Reading.	Area (.5056) Load per sq. in.	Heat No. (5,568) Average Microm. Reading.	Area (.4905) Load per sq. in.	Heat No. (5,534) Average Microm. Reading.
4,280	.0142.5	4,020	.0096.5	3,950	.0181.5	4,070	.0180.5
6,430	148.5	6,030	100.25	3,930	187	6,110	185.5
8,570	153.5	8,040	106.5	7,910	191.75	8,150	191.75
10,710	159	10,050	111.75	9,890	197	10,190	197
12,850	164.75	12,060	117	11,870	202	12,230	201.5
15,000	170.5	14,070	122.5	13,840	207.25	14,270	206.5
17,140	176.75	16,080	127.5	15,820	212.25	16,310	213
19,280	182.5	18,090	132.5	17,800	217.5	18,350	217.5
21,420	188	20,100	138	19,780	223.25	20,390	223
23,560	194	22,110	144	21,750	228.5	22,430	228.5
25,710	199.5	24,120	149	23,730	234.25	24,470	233.25
27,850	205.5	26,130	154.5	25,710	239	26,510	239
29,990	211	28,140	159.5	27,690	244	28,550	244.5
32,130	217.5	30,150	164.5	29,660	249.25	30,590	250
34,280	222	32,160	171	31,640	254	32,630	255
36,420	228	34,170	176.25	33,620	259.5	34,660	260.5
38,560	233.5	36,180	181.5	35,600	265.25	36,700	265.5
40,700	240	38,190	186.5	37,580	270	38,740	271.5
42,850	245	40,200	192	39,550	275.5	40,780	276
43,920	248	42,210	198	41,530	281	42,820	282.25
44,450	250.5	44,220	202.5	43,510	287	44,850	288.5
44,990	251	46,230	209	45,490	291.5		
45,520	252.5	48,240	211.5	46,480	295		
46,060	254.5	49,250	217				
46,600	255.5						
47,130	256.5						
47,670	258.5						
48,200	260						
48,740	262						

22" x ¾" Plate.

Not Annealed.		Annealed.			
Test No. 983.		Test No. 1,073.		Test No. 1,659.	
Area (.5002) Load per sq. in.	Heat No. (5,566 B) Average Microm. Reading.	Area (.4918) Load per sq. in.	Heat No. (5,566) Average Microm. Reading.	Area (.5063) Load per sq. in.	Heat No. (5,566) Average Microm. Reading.
4,000	.0052.25	4,070	.0037.5	3,950	75.25
6,000	57	6,100	42.5	5,920	80.5
8,000	62	8,140	47.5	7,900	85.25
10,000	67	10,180	53	9,870	90.25
12,000	72.5	12,210	58.5	11,850	95.5
14,000	77	14,250	63.75	13,820	101.25
16,000	82.5	16,280	69	15,800	106.25
18,000	87.5	18,320	74.5	17,770	111.25
20,000	92.25	20,350	80	19,750	116.5
22,000	97.5	22,390	85.5	21,720	122
24,000	103.5	24,420	92	23,700	127.25
26,000	107.5	26,460	96.5	25,670	133.25
28,000	113	28,490	101.5	27,650	137.5
30,000	118.5	30,530	107.5	29,620	142.75
32,000	123.5	31,550	110.5	31,600	147.75
34,000	129.5			33,520	153.75
36,000	134			35,550	158.25
38,000	139.75			37,520	163.25
40.000	144.5			39,500	169.75
42,000	149.75			40,490	173.5
44,000	155.5				

18" x ¾" Plate.

Not Annealed.				Annealed.	
Test No. 979.		Test No. 980.		Test No. 1,107.	
Area (.4977) Load per sq. in.	Heat No. (5,419) Average Microm. Reading.	Area (.5050) Load per sq. in.	Heat No. (5,472) Average Microm. Reading.	Area (.4665) Load per sq. in.	Heat No. (5,419) Average Microm. Reading.
4,020	.0194.5	3,960	.0177.5	4,280	.0126
6,080	200	5,940	181.5	6,480	131.5
8,040	205.25	7,920	186.5	8,570	137
10.050	210.5	9,900	191.5	10,710	143.25
12,050	216.5	11,880	196.5	12,850	149
14,060	221.5	13,860	202	15,000	155
16,070	227.5	15,840	207	17,140	161
18,080	232.5	17,820	212.5	19,280	166.5
20,000	238	19,800	217.5	21,420	172
22,100	243.5	21,780	223	23,560	178.5
24,110	249	23,760	228	25,710	184.5
26,120	254.5	25,740	233.5	27,850	190.5
28,130	260	27,720	238.5	29,990	196
30,140	265.5	29,700	245	32,130	202
32,150	272	31,680	249	33,210	204.75
34,160	277.5	33,660	254.5	34,280	208
36,170	283	35,640	260	35,350	110.5
38,180	289.25	37,620	265	36,420	214
40,190	294.5	39,600	273	37,490	216.5
42,200	301	41,580	277	38,560	219.5
44,200	306.5	43,560	281.5		
45,210	309.5	45,540	288		
46,210	312	46,530	291.5		
47,220	314	47,520	295		

18" x $\frac{5}{8}$" PLATE.

ANNEALED.

TEST No. 1,108.				TEST No. 1,658.			
Area (.4907) Load per sq. in.	Heat No. (5,472) Average Microm. Reading.	Area (.4907) Load per sq. in.	Heat No. (5,472) Average Microm. Reading.	Area (.4552) Load per sq. in.	Heat No. (5,419) Average Microm. Reading.	Area (.4552) Load per sq. in.	Heat No. (5,419 A) Average Microm. Reading.
4,070	.0176	24,450	229	4,390	.0170.5	26,360	228
6,110	181	26,490	234.5	6,590	176.25	28,560	234
8,150	186.5	28,530	240	8,790	182.0	30,760	240
10,190	191.5	30,560	246	10,990	187.5	32,950	246
12,220	196.5	32,600	251.5	13,180	193	35,150	251.75
14,260	202	34,640	256.5	15,380	199.25	37,350	257.5
16,300	202.5	36,680	262	17,570	205	39,550	263.25
18,340	213	37,700	265	19,770	210.75	41,740	269
20,370	219	38,720	268	21,970	216.5	43,940	275
22,410	224	39,790	270.5	24,170	222.5	45,040	280.5

24" x $\frac{1}{2}$" PLATE.

TEST No. 963.		TEST No. 966.		TEST No. 994.		TEST No. 936.	
Area (.4668) Load per sq. in.	Heat No. (5,444) Average Microm. Reading.	Area (.4975) Load per sq. in.	Heat No. (5,483) Average Microm. Reading.	Area (.5056) Load per sq. in.	Heat No. (5,568) Average Microm. Reading.	Area (.4905) Load per sq. in.	Heat No. (5,534) Average Microm. Reading.
4,280	.0142.5	4,020	.0096.5	3,950	.0181.5	4,070	.0180.5
6,430	148.5	6,030	100.25	3,930	187	6,110	185.5
8,570	153.5	8,040	106.5	7,910	191.75	8,150	191.75
10,710	159	10,050	111.75	9,890	197	10,190	197
12,850	164.75	12,060	117	11,870	202	12,230	201.5
15,000	170.5	14,070	122.5	13,840	207.25	14,270	206.5
17,140	176.75	16,080	127.5	15,820	212.25	16,310	213
19,280	182.5	18,090	132.5	17,800	217.5	18,350	217.5
21,420	188	20,100	138	19,780	223.25	20,390	223
23,560	194	22,110	144	21,750	228.5	22,430	228.5
25,710	199.5	24,120	149	23,730	234.25	24,470	233.25
27,850	205.5	26,430	154.5	25,710	239	26,510	239
29,990	211	28,140	159.5	27,690	244	28,550	244.5
32,130	217.5	30,150	164.5	29,660	249.25	30,590	250
34,280	222	32,160	171	31,640	254	32,630	255
36,420	228	34,170	176.25	33,620	259.5	34,660	260.5
38,560	233.5	36,180	181.5	35,600	265.25	36,700	265.5
40,700	240	38,190	186.5	37,580	270	38,740	271.5
42,850	245	40,200	192	39,550	275.5	40,780	276
43,920	248	42,210	198	41,530	281	42,820	282.25
44,450	250.5	44,220	202.5	43,510	287	44,850	288.5
44,990	251	46,230	209	45,490	291.5		
45,520	252.5	48,240	211.5	46,480	295		
46,060	254.5	49,250	217				
46,600	255.5						
47,130	256.5						
47,670	258.5						
48,200	260						
48,740	262						

24″ x ½″ Plate.

Annealed.

Test No. 1,070.		Test No. 1,081.		Test No. 1,083.		Test No. 1,084.	
Area (.4950) Load per sq. in.	Heat No. (5,534) Average Microm. Reading.	Area (.4944) Load per sq. in.	Heat No. (5,566) Average Microm. Reading.	Area (.5040) Load per sq. in.	Heat No. (5,483) Average Microm. Reading.	Area (.5050) Load per sq. in.	Heat No. (5,568) Average Microm. Reading.
4,040	.0132	4,045	0174.5	3,960	.0033.5	3,960	.0040.75
6,060	137.5	6,068	180.5	5,950	43	5,940	45.5
8,080	143.25	8,091	185.5	7,930	49	7,920	50.75
10,100	149	10,110	190.25	9,920	54.5	9,900	56
12,120	153.75	12,130	196	11,900	60.25	11,880	60.5
14,140	160.5	14,160	201	13,890	64.5	13,860	66.5
16,160	166.5	16,180	206	15,870	70.5	15,840	70.5
18,180	172	18,200	212	17,850	75.5	17,820	76
20,200	178	20,200	217.5	19,840	81	19,800	81
22,220	184	22,250	223	21,820	86.5	21,780	86.5
24,240	189	24,270	228.5	23,810	91.5	23,760	91.5
26,260	196.5	26,300	233.5	25,790	96.5	25,740	96
28,280	202	28,320	239.5	27,770	101.75	27,720	102
30,300	206.5	30,340	244.5	29,760	107.5	29,700	107
32,320	213.5	32,360	250	31,740	111.5	31,680	112
34,340	219	34,380	255.5	33,730	118.25	33,660	117.5
36,360	225	36,410	261.5	34,720	120	34,650	120
38,380	232					35,640	122.5
39,390	234					36,630	125
40,400	237.5					37,130	126.5
						37,620	127
						38,120	129
						38,610	130.25
						39,010	131.75

24″ x ½″ Plate.

Annealed.

Test No. 1,087.		Test No. 942.		Test No. 946.	
Area (.4713) Load per sq. in.	Heat No. (5,444) Average Microm. Reading.	Area (.4924) Load per sq. in.	Heat No. (5,534) Average Microm. Reading.	Area (.5100) Load per sq. in.	Heat No. (5,534) Average Microm. Reading.
4,240	.0053	4,070	.0109.5	3,920	122.25
6,360	58	6,110	115	5,880	127
8,480	63	8,130	120.75	7,840	131.5
10,610	68.75	10,170	125.25	9,800	136.25
12,730	74	12,210	129.75	11,760	140.75
14,850	79.5	14,230	135.5	13,720	146.25
16,970	84.5	16,250	140.75	15,680	151.75
19,100	90	18,270	145.75	17,640	157.25
21,210	95.5	20,310	150	19,600	163
23,340	101.5	22,330	157	21,560	167.25
25,460	106	24,370	162	23,530	172.5
27,580	111.5	26,400	167.5	25,490	177.25
29,700	117	28,430	173.5	27,450	185
31,830	122.5	30,460	179	29,410	190
33,950	128	32,490	184.5	31,370	195.75
36,070	133.5	34,520	190.5	33,330	201
37,130	136	35,540	193.25	35,290	206.5
38,190	139	36,550	196.5	36,270	209.25
		37,570	199.25	37,250	211.75
		38,580	203	37,740	213
				38,230	214.5
				38,720	216
				39,210	217.5
				39,700	218.5
				40,200	220
				40,690	222

24" x ½" PLATE.

NOT ANNEALED.		ANNEALED.			
TEST NO. 1,003.		TEST NO. 1,662.		TEST NO. 1,663.	
Area (.4910) Load per sq. in.	Heat No. (5,566) Average Microm. Reading.	Area (.4860) Load per sq. in.	Heat No. (5,430) Average Microm. Reading.	Area (.4906) Load per sq. in.	Heat No. (5,444) Average Microm. Reading.
4,073	.0041	4,110	.0147.5	4,070	.0123.75
6,110	44.5	6,170	152.5	6,110	128.5
8,140	50	8,230	158.25	8,140	133.75
10,180	55	10,290	163.75	10,180	138.25
12,220	58.25	12,340	169	12,220	144.5
14,250	62.5	14,400	174.25	14,260	150.25
16,290	68	16,460	180	16,300	155.75
18,330	73	18,520	185.75	18,340	161
20,370	79	20,580	191.5	20,380	166.75
22,410	84	22,630	197	22,420	172.25
24,440	89.5	24,690	202.5	24,470	177.75
26,480	94.5	26,750	207.75	26,510	183
28,510	100.25	28,810	213.75	28,550	188.5
30,540	105.5	30,870	219.25	30,590	194.5
32,580	112	32,920	225.5	32,630	199.75
34,620	116.5	34,980	231	34,670	205.75
36,660	122	37,040	236.5	36,710	211
38,700	128	39,100	242	38,750	216.5
40,740	132.5	41,160	247.5	40,790	222.75
42,770	138.5	43,210	253.75	41,810	225.5
44,810	144	45,270	259.25	42,820	228
46,850	149.5	46,300	262.75	43,830	231
47,870	152	46,820	264		
48,880	155	47,330	266		

DISCUSSION.

Prof. D. S. Jacobus.—I wish to make a remark about the method of using the micrometers devised by Mr. Henning, for the simple reason that many people have tried to use electric micrometers, and have failed to obtain the precise and uniform measurements given in this paper. The secret of success lies in employing a very weak electric current so that it will not spark at the contact points. The best method is to put in a low resistance relay in the circuit, and make this relay open and close the bell circuit. A very weak current may then be passed through the contact points, and a strong one through the bell. Another precaution which has to be employed is to make sure that there is no oil or grease on the ends of the micrometer contact pieces. Before making a test we polish up the ends of the contact pieces with a thin piece of clean pine wood. In this way measurements may be readily taken to one-quarter of a ten-thousandth of an inch, or .000025 inch.

CCCCXCVII.*

A NOVEL FLY-WHEEL.

BY CHAS. H. MANNING, MANCHESTER, N. H.
(Member of the Society.)

On October 15, 1891, the fly-wheel of a large pair of Corliss engines belonging to the Amoskeag Mfg. Co., of Manchester, N. H., exploded from centrifugal force, causing great destruction of property and the loss of three valuable lives. It was to replace this wheel that the one about to be described was built; but before speaking of the new wheel, it may be well to say something of the engine and the old wheel.

The engine was built by the Corliss Steam Engine Co., of Providence, R. I., in 1883, and has two cylinders 36 inches diameter, with 6 feet stroke of piston, and was running at the time of the accident at a little over 61 revolutions per minute. One cylinder was connected with a Bulkley condenser, and the other was working non-condensing. The fly-wheel of cast-iron and "built up," was 30 feet diameter and 110 inches face, with one set of 12 arms, and weighed 116,000 lbs. This wheel was turned for three belts, the two outside ones being 40 inches wide each and running back between the cylinders to a jack-shaft running to Mill No. 5; the central belt, 24 inches wide, leaving the fly-wheel in the opposite direction and connecting to a jack-shaft running to No. 7 Mill.

Figure 166 shows the general situation of all this, although it should be mentioned that the two jack-shafts do not stand in line with each other but diverge about two and three-eighths feet per hundred, and the engine shaft bisected this angle, which necessitated the use of guide pulleys on all the belts.

The engine had been run under precisely the same conditions for several days, including a water-wheel at part-gate attached to No. 7 jack-shaft and furnishing about 200 H.P., and one on

* Presented at the San Francisco Meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

No. 5 jack-shaft with gate closed to one-tenth and giving no power.

The indicator cards are on record for every day up to that of the accident, and show for several days previous about 1,950 H.P., and of this probably 1,850 was transmitted to the belts. The belts were heavy, double leather, and the total number of days' work they had done was 879, less than three years, as this

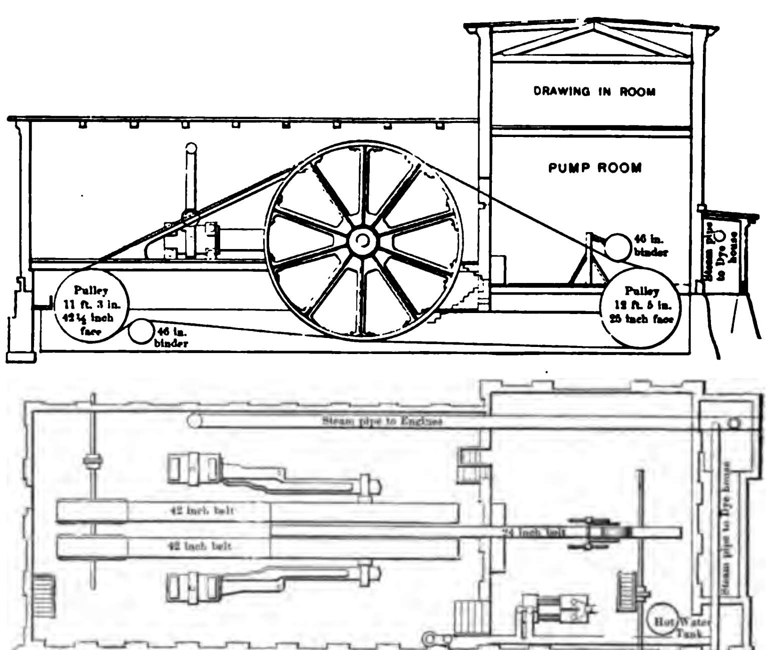


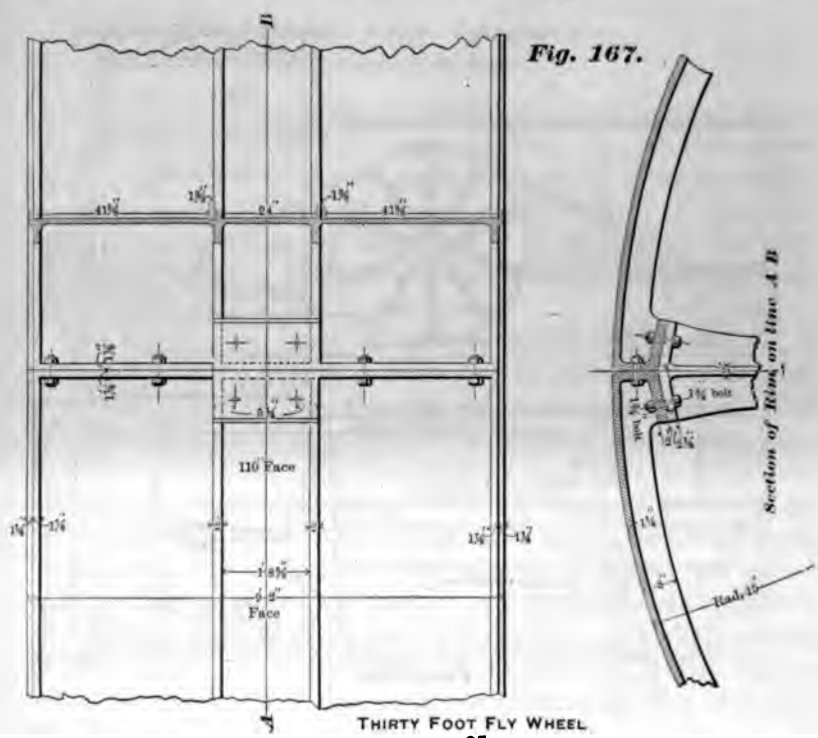
Fig. 166.

engine is only used in times of low or back water, when the water-wheels will not furnish sufficient power.

The belts had been carefully examined within a few days and were in good condition, and though heavily loaded were not overtaxed.

The first unusual occurrence was a loss of speed in No. 5 Mill, doubtless caused by slipping of the twin belts on the jack-pulleys ; the operatives, thinking the engine about to stop from some cause, threw off their work, when speed immediately came up once more and the machines were started ; this caused the belts to slip again and work was more generally thrown off, thus relieving the engine suddenly, the speed increasing sufficiently to attract the attention of the engineers, who immediately closed the throttles, but too late to save the wheel, which went to pieces at a speed of seventy-three or four revolutions per minute. On the No. 7 jack-shaft drive no irregularity of speed was noticed until the acceleration a few seconds previous to the explosion, when many of the looms "knocked off" automatically, but

Fig. 167.



THIRTY FOOT FLY WHEEL
OF
CORLISS ENGINE
AMOSKEAG M'F'G. CO.

one particular loom ran until the speed stopped after the explosion. Experiment demonstrated that this loom would "knock off" before it attained a speed corresponding to 75 revolutions of the engine.

The governor and its belt were intact and in excellent order after the explosion, with the exception of the trip rods, which were broken.

At this speed, had the castings been ordinarily good, the wheel should have been perfectly safe and, as will be seen by reference to Fig. 167, the bolting of the sections together with five $1\frac{3}{4}$-inch iron bolts, made these joints theoretically the weakest places; but in fact there were very few of the bolts broken, some of them even pulling out large washers of cast-iron; but the rim castings, as well as the ends of the arms, were full of flaws caused chiefly by the drawing and shrinking of the metal. Fig. 168 is a fair representation photographed from pieces of the wheel.

FIG. 168.

Specimens of the metal were tested for tensile strength and varied from 15,000 lbs. per square inch in sound pieces to 1,000 lbs. in spongy ones. None of these flaws showed on the surface, and a rigid examination of the parts before they were erected failed to give any cause to suspect their true nature or they never would have been accepted. Experiments were carried on for some time after the accident in the Amoskeag Company's foundry in attempting to duplicate the flaws, but with no success in approaching the badness of these castings.

DESIGN OF THE WHEEL.

It has been the practice of this company for some years past to make all pulleys having a rim speed of 3,000 feet or over with wood rims on cast-iron spiders, and after due consideration it

Fig. 170.

was determined to replace this fly-wheel with a wooden-rimmed one on two sets of arms, and the design shown in Fig. 169 was the result of this decision.

As it was not desirable to remove either of the cranks, the hubs were made in halves, and in erecting the wheel the design should have been perfectly safe and, as will be seen by reference to Fig. 167, the bolting of the sections together with five $1\frac{3}{4}$-inch iron bolts, made these joints theoretically the weakest places; but in fact there were very few of the bolts broken, some of them even pulling out large washers of cast-iron; but the rim castings, as well as the ends of the arms, were full of flaws caused chiefly by the drawing and shrinking of the metal. Fig. 168 is a fair representation photographed from pieces of the wheel.

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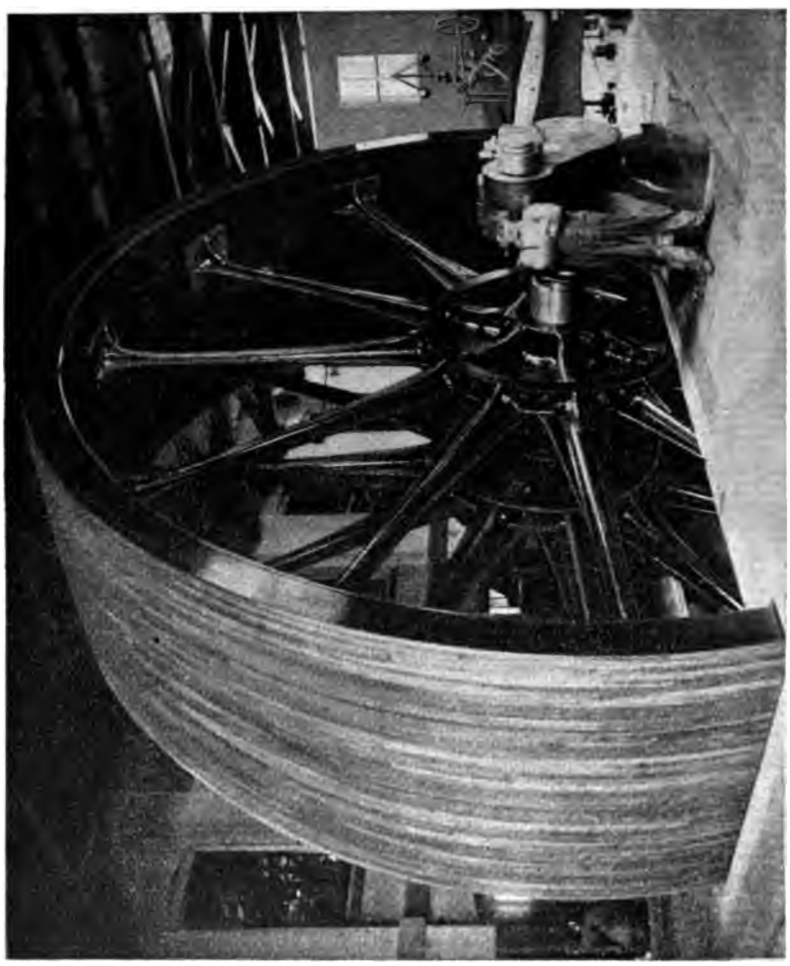

FIG. 170.

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as shown was departed from to the extent of placing the joints of the two hubs at right angles to each other on the shaft. The bolting together of the half hubs was made ample to hold the halves of the wheel together when running at normal speed of 61 revolutions per minute, independent of any other strength in the wheel. The arms were so placed that the inner end served as a butt-strap over the end of this joint. The arms were so designed that each one would safely carry the belt strain coming on its half the wheel, if the other eleven arms were entirely relieved, and the bolting to the hub is sufficient to carry its portion of the rim if cut adrift from the remainder of the rim. Sharp angles and changes of section in the arms were avoided as much as possible, as will be seen in the arm section, both where it joins the hub and the rim. The circular cross section was selected in preference to the elliptic on account of the greater certainty of uniform distribution of metal.

The counterbalancing of cranks and connecting rods was obtained by placing heavy cast-iron plugs in the hollows at the outer end of the three arms directly opposite each crank, and these plugs were secured in place by 1-inch bolts running through the centre of the arms, and large washers fitted to the inner ends of the arms. Though the total weight in the wheel is not much less than that of the old one, the weight in the rim is only about one-half, but it has shown itself to be ample for a very steady speed.

As for safety, the speeds being the same in both cases, the hoop tension in the rim per unit of cross section would be directly as the weight per cubic unit, and its capacity to stand the strain directly as the tensile strength per square unit, therefore the tensile strengths divided by the weights will give relative values of different materials.

Cast-iron weighing 450 lbs. per cubic foot and with a tensile strength of 1,440,000 lbs. per square foot would give a value of $\frac{1,440,000}{450} = 3,200$, whilst ash, of which the rim was made, weighing 34 lbs. per cubic foot, and with 1,152,000 lbs. tensile strength per square foot, gives a result $\frac{1,152,000}{34} = 33,882$, and $\frac{33,882}{3,200} = 10.58$, or the wood-rimmed pulley is ten times safer than the cast-iron when the castings are good.

This would allow the wood-rimmed pulley to increase its speed to $\sqrt{10.58} = 3.25$ times that of a sound cast-iron one with equal safety.

CONSTRUCTION.

In the foundry work on hub and arms great care was taken to produce perfectly sound castings, and as tough as possible. When poured, the metal was allowed to cool in ladles, as much as safe to avoid "cold shuts," and after pouring, it was thoroughly churned to get rid of all gas and avoid blow-holes, and no casting was uncovered for forty-eight hours after pouring.

The first arm was tested to destruction for cross-breaking strength, being bolted as to the hub and loaded at outer end, and it stood well above calculation. In the machine shop, after the hubs had been bored and turned, the arms were bored by being placed one at a time in a stationary saddle and the tool carried on the face-plate of the lathe. The arms being clamped to the hubs, they were drilled together.

The wreck of the old wheel being cleared from the shaft, the key-way was continued toward each end by a rotary milling tool travelling on guides clamped to the shaft, the tool being driven from a small stationary engine placed for the purpose. One hub was then put on, keyed fast, and its arms put in place and bolted on. The shaft was then revolved and marks scribed on the ends of the arms by which to place the first course of wood.

The wood was Western ash plank, thoroughly kiln-dried, eight courses being of 4-inch plank, seven of 2-inch, and the remainder of 3-inch, all of which was reduced about one-half inch in dressing, so that there are forty-four courses in all, each course or ring made of twelve pieces. The first ring of 3½-inch plank was set by the scribe marks so that the butts came over centre of arms, and with bolt heads let in 2½ inches, with washers under the heads and the nuts on the inner end. The butts were all left three-quarters of an inch open at the outer ends. The next course was applied with the butts breaking joints about 22 inches, and so on with each course, and the thickness of the courses was so arranged as to bring the arm-bolts through the centre of a 3½-inch course. Each course was carefully dressed to exact thickness on a buzz planer, and each piece was thoroughly glued and secured in place with sixteen 3½-inch lag bolts. The holes for these bolts and the countersinks for the heads were bored to template before leaving the wood-working shop; then each piece was clamped in the place it was to go, on the part already erected, the bolts were scribed through, and then the piece being removed, the holes were bored by a gauged auger, set on a flexible shaft and run by power. The piece was then glued, put in position, and the lag bolts screwed in by a socket-wrench set in the same flexible shaft. This shaft was driven with the usual rope drive, and the weight on the tension-pulley was so adjusted that when all the strain was on the lag bolt that it would safely stand, the rope would slip.

This allowed the lag screws to be set up very rapidly and to an even strain whilst the glue was still hot.

The wheel was turned one-twelfth of a revolution each time, and every ring completed before another was commenced. After the face had grown to about two feet in breadth, several round turns of manila rope about one inch in diameter were taken around it, and the ends then led off to the opposite sides around a suitable drum several times, and the ends spliced. This drum was revolved by engine power in either direction with friction clutches, so that the wheel as it grew and increased in weight was very readily handled. At the proper time the second hub and set of arms were put on, and the rim continued over this.

By reference to Fig. 169 it will be seen that between each pair of arms there are three seven-eighths inch bolts extending through the rim from side to side, and the holes for these bolts were bored in each piece before it was put in place, but left small and then rimmed to size after the pieces were all in place. The courses being completed, the next step was to drive well-fitted hard-wood wedges in the 504 spaces between the butts. These were driven on hot glue with a heavy sledge, as were also the hard-wood keys between the outer ends of the arms and the rim in the key-ways shown in the design. The wheel was then belted to the jack-shaft which was driven by a water-wheel, and slide-rests being set up, it was turned in position on the outside, and as far as the arms would permit on the inside, the remainder of which was finished by hand. All but the driving face was thoroughly painted, and into the driving face a finish of raw linseed oil and beeswax was worked under heavy friction with the wheel revolving, this being done to exclude moisture.

When completed the wheel was run up to a speed of 76

revolutions per minute, being a surface speed of nearly 7,200 feet per minute, and at this speed it ran absolutely true, and when stopped failed to show the least hair crack in varnish on inside of rim or where secured to arms. Since its completion, there having been an ample supply of water, it has been used but little, but so far has given entire satisfaction.

The cost of this wheel complete was $7,000 nearly, which is less than that of its sinful predecessor. In this cost is included all patterns and special appliances, and it could be duplicated for much less money. Safety was the greatest consideration, and it is firmly believed that this wheel is as safe and durable as any in existence.

DISCUSSION.

Mr. A. K. Mansfield.—**This paper deals with a new wheel, and, incidentally, with the old one which it replaced. I think we will all agree that Mr. Manning has designed the wheel in a thoroughly mechanical manner, and that it will probably stand the test required of it. But we ought to be, I think, concerned chiefly with the old wheel, from the fact that in the Eastern part of the country there have been quite a number of accidents from wheels bursting lately, which makes this a very important subject to engineers.** Quite a number of our large wheels have gone to pieces to very nearly the same time; so much so that I fear some users of large wheels which have not gone to pieces will naturally feel great anxiety.

Mr. Manning, on page 620, says: "At this speed, had the castings been ordinarily good, the wheel should have been perfectly safe, and, as will be seen by reference to Fig. 167, the bolting of the sections together with five $1\frac{3}{4}$-inch iron bolts made these joints theoretically the weakest places." I think perhaps it would have been better for him to say, "made these joints very weak places." He shows us that he has designed a wheel to perform the same service as the one that went to pieces, several times stronger than that was theoretically. Yet he seems to consider that the wheel which went to pieces was theoretically strong enough. This seems a little inconsistent; and suspecting it to be so, I have made a few figures which may be interesting.

It is very easy for us to calculate what the centrifugal tension is, in a wheel running at any speed, tending to part the wheel through one of its sections—to part it in two halves. Applying the well-known formula, I found that the centrifugal force tending to part this wheel—that is, to divide it in two halves—was about 360,000 lbs. at 60 revolutions, which speed, I take it, the engine was designed to run at. This is based on the weight of the rim alone; arms ignored. Adding one-half of the total weight of wheel to this, which is proper, for the force of gravity of this half also tends to divide the wheel in halves, this figure becomes 418,000 lbs. If we assume that the hub where the wheel is bolted on the shaft carries one-third of the strain, which may be proper to assume in the case of this wheel, it being very easy to get large bolts in at the hub (the paper does not show how it was bolted together there)—if we assume, therefore, that at the hub the strain taken by the bolts was one-third the centrifugal force of the rim, in addition to the whole centrifugal force of arms and hub, it leaves one-third of the former force to be accounted for, or to be resisted, at the rim on each side. One third of this strain is 139,000 lbs. There are five bolts. One-fifth of this figure is 27,900 for each bolt. These bolts inside of the thread have a cross section of about 1¾ inches, which makes the force per square inch of bolt 15,920 lbs., or, roughly, 16,000 lbs. Now this means that if these bolts were placed in such a position that they directly resist the force tending to separate the wheel, the strain on them would be, roughly, 16,000 lbs. to the square inch, which is a greater strain than we ordinarily allow in practice in such parts of machinery. But the bolts are not placed where they resist the strain directly; they would have to be placed directly in the rim, which cannot be; they are placed inside of the rim, some little distance away. In designing wheels I have generally placed these bolts in such place that I consider there is a leverage of 2 to 1 on the bolts, and I think in the present case it is fair to assume there was a leverage of 2 to 1. If so, they were under a strain of about 32,000 lbs. per square inch. Now, at once you may think the bolts should have given out and not the casting. Mr. Manning states that not many of the bolts gave out, or something to that effect. It is quite sufficient if any of them gave out. An examination of the diagram of the old wheel will show that, considered roughly—bearing in mind that we have no figures in some parts on which to base calculation—it would seem that the remainder of the wheel was designed to conform in strength to the strength of the bolts, or in weakness, I should rather say. But besides that, there were flaws in the casting. I doubt if there are many wheels of that sort made in which there are not flaws. I think if you break up wheels of that kind, or other large castings of that sort, you will nearly always find flaws of the kind mentioned. The fact is, as near as I can judge—I think, perhaps, you will agree with me if you do this figuring over after me—that in this particular case the wheel is a badly designed wheel; the bolts should have been three times as strong, and the wheel casting about the bolts about three times as strong as it was This is an interesting fact—it seems to me an interesting fact—and it is a manifestly important matter to us, considering the number of wheels which have gone to pieces lately. What caused the other wheels to go to pieces I don't know. This wheel went to pieces under a higher speed than it was intended to run at; it seems to have been too weak to run at the speed at which it was designed to run.

Mr. James McBride.—There is one very remarkable thing in regard to these two wheels. If I understand the paper, the first wheel, the writer stated, weighed 160,000 lbs.; the new wheel weighs only about half that much, and regulates the engine perfectly.

The first wheel was twice as heavy as there was any necessity for. Now is it not possible if the first wheel had been of less weight—half as heavy as it was—and made of good material, it would not have gone to pieces? There seems to be considerable discrepancy between the opinions of the designers of the two wheels. As Mr. Mansfield says, I think it was badly designed. If an engine-builder designs a wheel for an engine and it breaks, and another man comes along and puts one on half as heavy, that does the work and runs the engine steadily, evidently the first man was at fault.

* The breaking of this *heavy weak* cast-iron wheel, and the substitution of a *strong light* wooden wheel emphasizes the fact not generally known, that nearly all very large cast-iron band wheels which serve as balance wheels on steam-engines are much heavier than is really necessary, owing to the great difficulty of constructing such large wheels from light castings. Builders are obliged to make the segments very thick to insure good castings, and to allow sufficient stock for finishing, and are generally sure to

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*Added after adjournment.

make all parts heavy enough to meet those ends. Hence they get wheels that are too heavy.

Mr. Manning has done a good thing in producing a *light* wheel which is heavy enough for regulation, and yet many times stronger than an all cast-iron wheel.

Mr. Mansfield.—I would like to add a little to what I have said. According to the figuring I made on the wheel, I found the factor of safety might be considered to be about 2, referring to the ultimate strength of the material. A wheel may run twenty or any number of years with perfect safety, provided it is subject to no undue strain, but only the strain which it is designed to bear. We know in this case the wheel received those undue strains. The conclusion I come to would be, naturally, I think, the factor of safety was not great enough—that is, it was not as great as the best judgment of engineers teaches them to apply in such cases.

Mr. T. J. Borden.—Twenty-three years since I put in a wheel 30 feet in diameter, 9 feet 2 inches face, from the same parties who made the Amoskeag wheel, and a year earlier another mill in the same city put in one of the same diameter, with only two inches less face, from the same makers. There are about fifty wheels in the city of Fall River, ranging from 24 feet to 30 feet in diameter, and most of them from 7 feet to 9 feet 2 inches face. Most of them are of the same general construction and proportion as the Amoskeag wheel, but are usually run at about ten per cent. less surface velocity. The average period of service of the entire number is probably about fifteen years.

I have known of no failure to stand the service to which they have been subjected. I have some doubts as to the giving way of the Amoskeag wheel having been due to centrifugal force. Other conditions existed in that case which may have been the prime cause of the accident. Some reference has been made to other wheels going to pieces recently. I know of one quite recently which, from facts that have come to my knowledge, I have no doubt gave way from the weakness of the shaft, it having been in use but a very short time and had shown weakness from the beginning of its operation.

The wheel should not depend solely upon the bolts in the flanges of the rim segments to hold it together. Each arm of the wheel should be able to sustain itself and the section of the rim attached to it independent of any other arm. The end of each arm in the Amoskeag wheel, and all other wheels from the same makers, is attached to the centre of a segment of the rim, and the tensile strength of each of those arms and the four bolts securing the rim segment to it was sufficient to afford a large margin of safety. Hence the wheel should have been able to withstand the centrifugal force regardless of the bolts in the end flanges of the rim segments. If dependence was to be placed upon the segments being bolted sufficiently to withstand the centrifugal force regardless of the arms, there should undoubtedly have been a greater number of bolts at each joint, or a stronger form of flanges used.

I am inclined to think that if that wheel had been run with half the bolts taken out of the flanges of the segments, the wheel would have held together, from the fact that each segment would have been held by the arm to which it was attached.

Mr. F. H. Laforge.—I would like to ask the speaker if he knows of any wheels running with as wide a face as the wheel in question had when it broke. I have not read the paper carefully to say just what the width of the face is, but my impression is that this was an extraordinarily wide-faced wheel.

Mr. Borden.—The two 30-foot wheels to which I previously alluded as having been in service twenty-three and twenty-four years were of the same diameter as the Amoskeag wheel; one was 108 inches face and the other 110 inches face. I think the Amoskeag wheel was 110 inches face.

Mr. George W. Dickie.—There is one question which has not come up in this discussion, nor in the reading of the paper, which is quite a prominent factor; that is, that this wheel burst under a pressure that it was not designed for. Wheels, like boilers, are built to stand a certain pressure; here was a wheel built to run 61 revolutions, and it was run at 73. There is no explanation given of why that engine was running at 73 revolutions when it was designed to run at 61. The pressure on the wheel would increase as the square of the revolutions, and consequently it was subjected to a far greater strain than it was designed for. It seems to me that there was something else than the wheel at fault, and that the wheel here branded as the sinful predecessor of the one which took its place was more sinned against than sinning. The governor of the engine had evidently been out of place or inoperative. In condemning structures of that kind, all the causes should be taken into account, and if the structure was subjected to a strain that it was never designed for, why, condemnation would not be proper.

Mr. Wilfred Lewis.—It looks to me as though the broken wheel was very badly designed. The bolts to unite the sections are put through the projecting flanges without any supporting ribs, and although the tension in the rim was not excessive, the eccentric loading might strain the flanges up to the breaking point very easily. At 7,200 feet a minute the centrifugal tension would be about 1,400 lbs. to the square inch, and the application of that load on flanges without supporting ribs seems to be a very good reason why the wheel should have broken as it did.

*Mr. C. H. Manning.**—In closing this argument, I have very little to say in way of criticism of the remarks that have been made. I differ entirely from Mr. Mansfield in doubling the strain on the bolts on account of any leverage on them, as it could only be true if the cast-iron was perfectly rigid in one direction and very pliable in the other, and any increase of strain due to this cause is very slight, and I see no reason to change the language of my paper, especially as Mr. Mansfield practically agrees with my statement, that, theoretically, the bolts were weaker than good castings.

If Mr. Mansfield will examine the wreck of the wheel, he will be convinced that the bad casting did the mischief.

According to Mr. Mansfield's figures, the bolts would have been taxed beyond their elastic limit every time the wheel went to speed; consequently would have stretched, which they did not do.

There was no inconsistency in making the wheel stronger than the old one was theoretically, as, from the nature of the material used, I could not well do otherwise, as the strength is a function of speed and $\frac{\text{tensile strength}}{\text{weight}}$, and making the rim thicker or thinner, as long as rigidity is retained, would not affect its strength. I have probably broken up as many large castings as most men with thirty odd years spent amongst machinery, and I never saw such flaws or so many of them; and a distinguished ex-president of our Society, with still more years of experience behind him, said he had never seen their like, and only wondered that the wheel ever held together to be turned.

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Mr. McBride was mistaken as to the figures in the paper: the old wheel weighed 116,000 lbs., and the new one 104,000, but the latter having two sets of arms, against one in the old, made a much greater difference in rim weights, they being about as 74,000 to 32,000 in round numbers. Mr. McBride is undoubtedly right that most builders of fly-wheel pulleys, in order to get sufficient face, get too much weight.

Mr. Borden's statement that "other conditions existed in that case which may have been the prime cause," doubtless means that the jack-pulley may have given out first; but that is simply impossible, as several tons of the fly-wheel wreck were under the wreck of the jack. The other wheel, which went to pieces on account of a weak shaft, at Willimantic, was of about the same size, and also had but one set of arms, which I consider a very grave fault, as by applying the weight to the shaft through two sets of arms it is brought much nearer the bearing, and the bevelling of the shaft avoided. By reference to Fig. 167, Mr. Borden will see that he is mistaken about the arm bolting to the middle of the segment, as the segments abut on the arm.

If Mr. Dickie has ever had any experience with Corliss engines he must know that when the governor is set for a heavy load the engine will overrun at a light load. The governor was not only in excellent working order, but it was aided by a Gale regulator in addition, but yet it failed to perform its office. To be absolutely reliable, a cut-off governor should fail to engage the valve at all when the engine is over speed, but with single eccentric and any considerable range of cut-off this is impossible with the Corliss engine.

CCCCXCVIII.*

AUTOGRAPHIC RECORDING APPARATUS FOR USE IN THE TESTING OF MATERIALS.

BY THOMAS GRAY, TERRE HAUTE, IND.
(Member of the Society.)

IT has seemed to be important that a reliable and convenient method should be obtained for automatically recording the behavior of different kinds of materials while they are being tested. Considerable attention has therefore been devoted by the author to this subject for several years, and it is proposed in this paper to give a brief description of the latest form of apparatus used by him for recording tests on tensile and compressive strength. This, like several other forms of apparatus which have been contrived for the purpose, draws a double diagram of the change of length of the specimen in comparison with the stress applied during the test. One of these diagrams magnifies the change of length in a ratio varying from 100 to 500 times, according to the adjustment of the apparatus, and is intended only to show the behavior of the specimen for strains below its elastic limit. This range of adjustment for the sensibility enables the full size of the diagram sheet to be taken advantage of for materials varying considerably in their capacity for elastic strain, such, for example, as timber and soft iron. The other diagram is intended to show the relation of the permanent changes of length to the forces producing them. The scale of this diagram can also be varied to suit the circumstances of the case.

The apparatus is illustrated in the accompanying diagram (Figs. 171, 172), in which *M M* are the two grip crossheads of one of Riehle's testing machines of the Harvard type. The lower of these crossheads is adjustable in height by means of two screwed rods (one on each side of the grip), which connect the crosshead to the piston of the hydraulic apparatus used for applying the stress to the specimen. To the upper crosshead a

* Presented at the San Francisco Meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

Mr. McBride was mistaken as to the figures in the paper: the old wheel weighed 116,000 lbs., and the new one 104,000, but the latter having two sets of arms, against one in the old, made a much greater difference in rim weights, they being about as 74,000 to 32,000 in round numbers. Mr. McBride is undoubtedly right that most builders of fly-wheel pulleys, in order to get sufficient face, get too much weight.

Mr. Borden's statement that "other conditions existed in that case which may have been the prime cause," doubtless means that the jack-pulley may have given out first; but that is simply impossible, as several tons of the fly-wheel wreck were under the wreck of the jack. The other wheel, which went to pieces on account of a weak shaft, at Willimantic, was of about the same size, and also had but one set of arms, which I consider a very grave fault, as by applying the weight to the shaft through two sets of arms it is brought much nearer the bearing, and the bevelling of the shaft avoided. By reference to Fig. 167, Mr. Borden will see that he is mistaken about the arm bolting to the middle of the segment, as the segments abut on the arm.

If Mr. Dickie has ever had any experience with Corliss engines he must know that when the governor is set for a heavy load the engine will overrun at a light load. The governor was not only in excellent working order, but it was aided by a Gale regulator in addition, but yet it failed to perform its office. To be absolutely reliable, a cut-off governor should fail to engage the valve at all when the engine is over speed, but with single eccentric and any considerable range of cut-off this is impossible with the Corliss engine.

CCCCXCVIII.*

AUTOGRAPHIC RECORDING APPARATUS FOR USE IN THE TESTING OF MATERIALS.

BY THOMAS GRAY, TERRE HAUTE, IND.
(Member of the Society.)

IT has seemed to be important that a reliable and convenient method should be obtained for automatically recording the behavior of different kinds of materials while they are being tested. Considerable attention has therefore been devoted by the author to this subject for several years, and it is proposed in this paper to give a brief description of the latest form of apparatus used by him for recording tests on tensile and compressive strength. This, like several other forms of apparatus which have been contrived for the purpose, draws a double diagram of the change of length of the specimen in comparison with the stress applied during the test. One of these diagrams magnifies the change of length in a ratio varying from 100 to 500 times, according to the adjustment of the apparatus, and is intended only to show the behavior of the specimen for strains below its elastic limit. This range of adjustment for the sensibility enables the full size of the diagram sheet to be taken advantage of for materials varying considerably in their capacity for elastic strain, such, for example, as timber and soft iron. The other diagram is intended to show the relation of the permanent changes of length to the forces producing them. The scale of this diagram can also be varied to suit the circumstances of the case.

The apparatus is illustrated in the accompanying diagram (Figs. 171, 172), in which *M M* are the two grip crossheads of one of Riehle's testing machines of the Harvard type. The lower of these crossheads is adjustable in height by means of two screwed rods (one on each side of the grip), which connect the crosshead to the piston of the hydraulic apparatus used for applying the stress to the specimen. To the upper crosshead a

* Presented at the San Francisco Meeting, May, 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions*.

beam, P, is bolted, and on the top of this beam the recording apparatus is mounted and enclosed in a case. The outer end of the beam P is braced by means of stay rods from the lower part of the platform, carried by the steelyard levers of the machine, so that any distortion of the crosshead does not affect the record. To the specimen A two frames of the form shown in the figure are pivoted at d and g of the main diagram and at B in the detail (Fig. 172). The bars b b are round, with the exception of the central block, and are made long enough to allow the side bars to be separated far enough to enclose the largest specimen which the machine is intended to test. The pivots are held firmly in two centre-punch holes by means of two flat springs, to which the ends of the chains a a are attached. This makes a form of frame which is very easily put in position on the specimen, and which is at the same time rigid in the direction of the forces applied to it. The pivots being placed across a diameter of the specimen, and the recording apparatus worked from the centre of the end bars of the frames, insures that bending of the specimen will produce little effect on the diagram. The upper of these two frames is pivoted at c to a link hinged to the upper crosshead M, then to the specimen at d, and finally to a rigid rod, acting as a strut between the two frames, at e. The lower frame is hinged at f to the strut connecting the two frames, is pivoted to the specimen at g, and carries on its free end a link h i, which connects it to the vertical rod R. The length of the link h i is the same as the length from e to f of the strut connecting the ends of the two frames, so that if g moves relatively to a the rod R will move just twice as much without any error due to the arcs round which the points f and h turn. The block at i is rigidly fixed to the rod R, but is pivoted to a second block, which slides on the link so as to give the necessary freedom of motion. The rod R slides in guide blocks fixed to the machine near the upper and lower ends of the rod. It is evident that by clamping the frames at the proper distance apart on the rod e f, any length of specimen may be used. It will be readily seen, also, that since the point c is practically fixed relatively to the machine, the point h will also be fixed so long as the specimen does not change length, and therefore slipping in the grips does not affect the record. If, however, the specimen stretches, the rod R will move down through a distance equal to twice the change of length between the points d and g. The record of

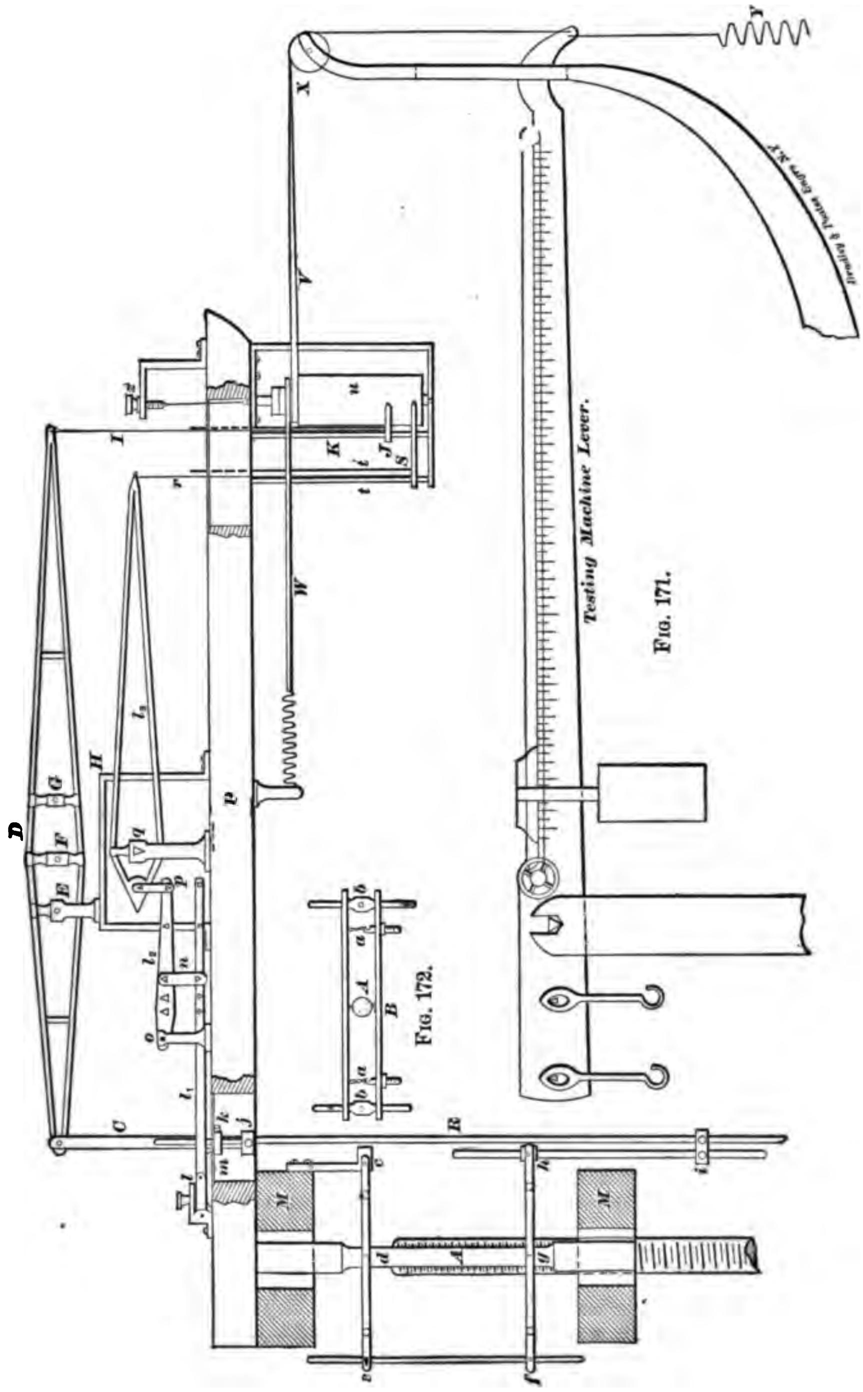

FIG. 171.

FIG. 172.

the elastic elongation of the specimen is made by means of the system of levers, l_1, l_2, l_3. The left-hand end of the lever l_1 is pivoted to a hinged piece, l, which can be raised or lowered by means of a screw so as to adjust the position of the record pen. Passing from the hinged piece l, the lever l_1 rests through a knife edge and a vertical strut, m, on a block, k, clamped to the rod R, and its forward end is carried by the lever l_2, to which it is connected by the link n. The lever l_2 rests by a knife edge on the supporting pillar at o, and its forward end is carried by the lever l_3, which it helps to counterpoise. The lever l_3 rests by a knife edge bearing on a pillar at q, and carries on its forward end a pen, s, which is connected to the lever by a light thread. The pen s consists of a very small ink-well, furnished with a capillary opening opposite the paper on which it writes. This pen is carried on the end of an arm, s, and is guided to move vertically by means of the fixed rod t and a light rod t' fixed to s and guided at its upper end. The length of the supporting thread r is made about in the same ratio to that of the link p as the ratio of the long to the short arm of the lever l, allowance being made for the motion of l_2. The ratio of the arms of the lever l_3 is 10 to 1, but by changing the position of the link n the combined ratio of the other two levers may be made either 5, 10, 15, 20 or 25 to 1, thus enabling the magnification of the record to pass from 100 times to 500 times by convenient steps. This amount of magnification is found amply sufficient for any ordinary material if the specimen be a few inches in length, the observations on a specimen 8 inches long being equivalent to direct observation on a bar 330 feet long. It will be noticed that the method of attachment here described allows the block k on the rod R to move away from the strut m, and thus to leave the sensitive system of levers unaffected by elongations beyond the range of the diagram. Immediately permanent set and flow of the material sets in, the pen s moves suddenly up against a stop on the rod t, and the levers are unaffected by the subsequent stretching. The very beginning of the permanent set is thus obtained with great accuracy.

The permanent elongation of the specimen is recorded by means of the lever D, which is connected to the rod R by means of the connecting rod C, and rests on a round axis at E or F or G, according to the scale of the diagram desired. The first of these positions for the axis magnifies the elongation four times, the second three times, and the third two times. This lever produces its record by means of a pen, j, which is precisely similar to s. With regard to the dimensions of the parts, it may be stated that in the apparatus now being used at the Rose Polytechnic Institute the lever D is four feet long. The shortest lever arm in the whole apparatus is thus about two inches, a length which can be measured with considerable accuracy even if no direct method of standardizing the system could be adopted. The long levers are made of light, hard steel rods, and have the form shown in the diagram, which insures a very light and yet rigid system.

The record is made on a sheet of paper wound on the drum u. This drum is pivoted on straight bearings at top and bottom, and its weight is carried by a thin steel wire attached to the adjusting screw z. The drum thus turns very freely, and care is taken to arrange the driving apparatus to insure little or no pressure on the bearings. The drum is provided with two sets of double pulleys of different sizes placed near the upper end of the axis. Supposing, as shown in the diagram, the larger set to be in use, a silk tape is placed on each pulley, and the end of one of these tapes carried over the pulley x and attached to the end of the testing machine lever through a special slip grip arrangement not shown in the diagram. The other tape w is attached to the end of a light spring capable of elongating by an amount equal to the circumference of the pulley. These two tapes pulling against each other in the same plane produce no pressure on the axis. The position of the end of the machine lever is controlled by a spring of the proper strength, the elongation of which by the pull on the specimen under test causes the drum to revolve in proportion to the stress applied. The horizontal scale of the diagram is thus proportional to the strength of the spring y, and this scale is usually obtained by comparing the vertical motion of the pen for a pull on the specimen measured by the sliding steelyard weight with the motion of the pen for any convenient displacement of the drum round its axis when controlled by the spring. For standardizing tests of this sort a hard steel bar is placed in the machine so as to avoid trouble due to "set" during the test. The stresses indicated by the instrument are thus the same as those which would be obtained by testing with the sliding weight. It will be observed that provision is made in this form of the apparatus for producing a diagram to rectangular coördinates. This is the chief difference between the apparatus here described and that which I have used for several years in which the pens were attached directly to the ends of the levers.

When materials are tested for crushing strength or for elasticity under compressing forces, the specimen is placed below the lower crosshead *m*, which is then pulled down on it, producing compression between the crosshead and the sole plate of the balanced part of the machine. A longer link has then to be used to make the connection to the point *c*, or, as is found rather more convenient, the bar *e c* is placed below, and a strut connection made to the sole plate instead of the link to the upper crosshead. The method of obtaining the diagram is otherwise identical with that already described.

For cross bending tests the same indicating apparatus is used with some simple modifications of the attachments, but as this part of the testing apparatus will probably be modified in the near future in accordance with some new plans which have been thought out, detailed description is left to another opportunity, when the whole subject of cross bending tests can be discussed.

The annexed diagram (Fig. 173) illustrates the record obtained by this apparatus. The curves marked (1) were obtained from a round specimen of tool steel .884 inch in diameter. The specimen was made from bar steel $1\frac{1}{8}$ inches diameter, and was turned down over a length of $9\frac{1}{2}$ inches, the shoulder curving gradually from the larger to the smaller diameter. It will be noticed that in this specimen there is no definite yield point such as is shown in a marked manner in the other curves. The specimen draws down nearly uniformly and shows very little local contraction. The curves marked (2) were obtained from a specimen of soft steel .875 inch diameter and similar in form to the tool steel specimen. In this case there is a marked elongation with nearly constant strength a little past the elastic limit, and there is a considerable falling off of strength before rupture and during the period in which local contraction takes place. The strength at the point of rupture is, however, very much higher than that obtained by taking the ratio of the maximum load to the initial area of cross section. The curves marked (3) were obtained from a specimen of ordinary bar iron. The part under test was initially .844 inch diameter, the shape being the same as that of the other two specimens. This specimen behaves similarly to the soft steel at the yield point, but has a much less gradual falling off of strength just before rupture, a peculiarity common to several diagrams taken from this iron. The strength at rupture, reckoned on the ruptured area, is in this case less than that obtained by taking the ratio of the maximum load to the initial section. The local contraction of section before rupture

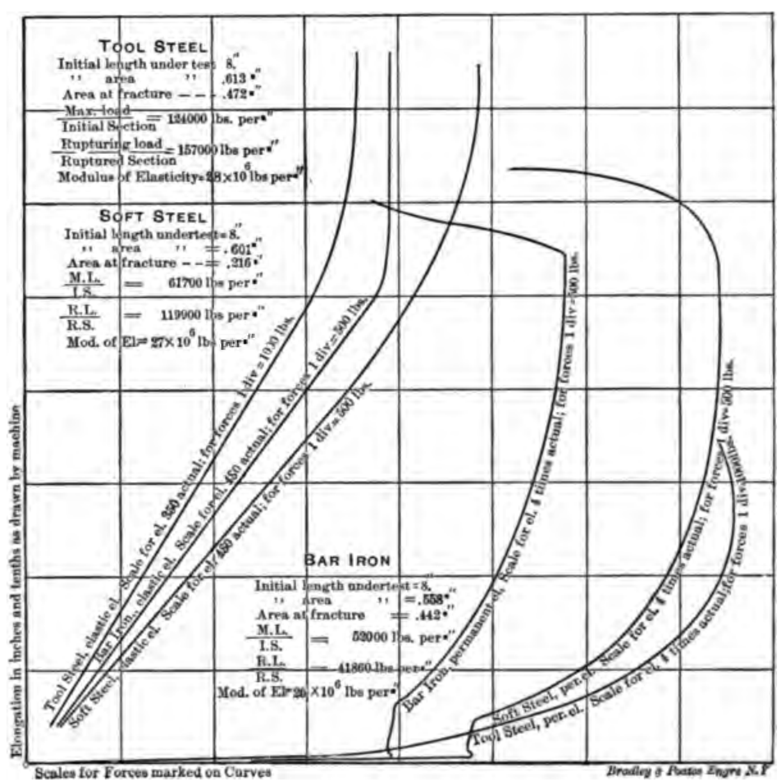

FIG. 173.*

was, as will be seen from the figures in the diagram, much smaller than that given by the soft steel, although the fall of strength was about equally great. The character of the fracture was similar to that usually obtained for material of the kind experimented on.

* The squares in this diagram correspond to square inches in the original diagram, which were again decimally subdivided.

In determining the exact magnification given by the sensitive system of levers, the most convenient arrangement is to introduce a micrometer screw in the rod *ef*, by means of which the length of the rod can be changed by a known amount and the corresponding rise or fall of the pen observed. Some care has of course to be taken to insure accuracy in the use of a standardizing device of this kind, but the necessary precautions will readily suggest themselves to any one having experience in such measurements.

DISCUSSION.

Mr. Gus C. Henning.—In my paper on the "Elastic Curve and Treatment of Structural Steels" presented at the San Francisco meeting* I have shown numerous curves of the elastic line of two grades of steel, the data for which had been determined by a reasonably exact testing machine and an improved form of double electric contact micrometer, made by Brown & Sharpe.

During the examination of these curves it became evident that autographic recording apparatus drawing a diagram on a very highly magnified scale would have been most desirable, as many apparent errors and discrepancies could then have been avoided. At the time when the tests in question were made, a number of devices were in use for drawing diagrams on a small scale; they were familiar to the writer, but were not considered of any practical value for the purposes and objects in view, and hence actual measurements were resorted to, which were afterwards worked out and laboriously plotted. Only those who have done similar work can appreciate the labor this necessitated and the ease with which errors would pass unnoticed. There are many diagramming apparatus beginning with Thurston's in 1875, such as those of Unwin, Kennedy, Martens, Pohlmeyer, Mohr and Federhaff, Wicksteeds, Olsen, Abbott, and several others, all of more or less value and accuracy, but none of them give a greatly enlarged curve within the elastic limit, and are therefore of little use in investigating the elastic properties of materials within that limit, as all variations in this curve for any class of material such as wrought iron, low steel, medium and tool steels, would be covered by a single line as drawn by an ordinary pen or pencil.

* *Transactions*, Vol. XIII., p. 572, No. 496.

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* *Transactions*, Vol. XIII., p. 572, No. 496.

In order to explain or illustrate this more fully, I have drawn several curves on Fig. 199, on several scales.

Curve *abcd* is one that could be drawn on a sheet wrapped about a drum 6 inches in diameter, with elongations magnified 10 times and loads drawn 8,000 lbs. to the inch. Curve $ab_1c_1d_1$ is the same drawn on a sheet wrapped about a drum 12 inches in diameter, scales twice the above. Curves, 693, 695, and 1,003, are three curves drawn to actual figures of tests 693, 695, 1,003, and on scales the same as used for all the curves shown in any paper read before this meeting, viz.: elongations magnified 1,000 times, and loads of 2,000 lbs. to the inch.

The parts of the two first mentioned curves (*abcd* and $ab_1c_1d_1$) lying between a and c, and a and c_1, are correctly drawn from tests 693, 695, 1,003, and appear as perfectly straight lines, because the scale on which they are drawn is so small. To show this, I have laid down on the large curves of tests 693, 695, 1,003, a zone or belt of hatching, which represents an ordinary line as drawn on the smaller curves magnified to the scale of the curves or 100 times. Had I drawn curve 693, with its origin or starting point to the left of the other two, as shown by the dotted line, instead of to the right as shown, then a zone or belt 1 inch wide, on the full-size original drawing, instead of 2 inches, from which the engraving is reduced, would have included all of these curves. All of the curves discussed in my paper would actually fall within this belt or zone 2 inches wide, and I think this proves conclusively that the ordinary diagrams are quite unsatisfactory to serve as a guide to, or indication of, the elastic properties of materials.

Having carefully studied tests of materials for several years, I had long ago concluded that small diagrams were of little practical value, and also that apparatus for such purpose would have to be constructed on different principles, in order to give satisfactory results.

Being familiar with most of the existing forms of such apparatus from personal observation and examination, I desire to offer a few remarks about the apparatus described by Prof. Gray in the paper before us, and regret exceedingly that I have not been able to examine the machine in operation, and can merely judge of its work by a careful examination and study of the curves presented in the paper as results thereof. The figures showing the apparatus are hardly more than schematic, and may be very misleading; they are certainly not sufficiently detailed or accurate to give satisfactory knowledge of the apparatus itself, and whether it promises to give duplicate results, or even whether it is not easily deranged during a test.

The machine has the appearance of great complexity and multiplicity, and can be used on few testing machines other than those like the type named; but this is not a very grave objection, for it carries within itself a device for calibration, a micrometer, which is undoubtedly a great improvement on most other machines, and is I think quite new.

The multiplicity of parts is very apt to lead to indefinite or varying errors, which cannot be detected or corrected instantly or even easily, and it seems to me that changing centers by shifting the links *n*, or pivots *E*, *F*, *G*, would each time introduce a new variation, because it is not likely that these links or pivots can all of them be made mathematically correct with relation to each other and other parts of the machine. Moreover, it will be observed that the moving parts are of considerable mass, and although designed to have the greatest stiffness, their inertia must of necessity be the cause of errors in the diagrams.

As I have before explained, these curves—even those in which the elongations have been magnified 500 times—are on too small a scale to be used for a study of the elastic line, and I shall recur to this further on. From all of them it is not easy to locate or determine the true elastic limit, although the yield point is sharply defined, except in the case of tool steel, in which case the inertia of the apparatus has undoubtedly obliterated it, as frequent determinations by actual measurements by many observers always clearly defined this point, although not as markedly as in softer materials.

In Fig. 173 are given six curves on varying scales; and assuming that the large curves represent the same material as would give the smaller curves, we find considerable discrepancy between them.

In the first place, the curves of bar iron and soft steel are not correct immediately beyond the yield point, as the machine has not responded to the material with sufficient promptness; the gradually rising line immediately following the drop at yield point should in both cases be a curve convex to the right edge of the diagram, merging into the long sweeping curve beyond,

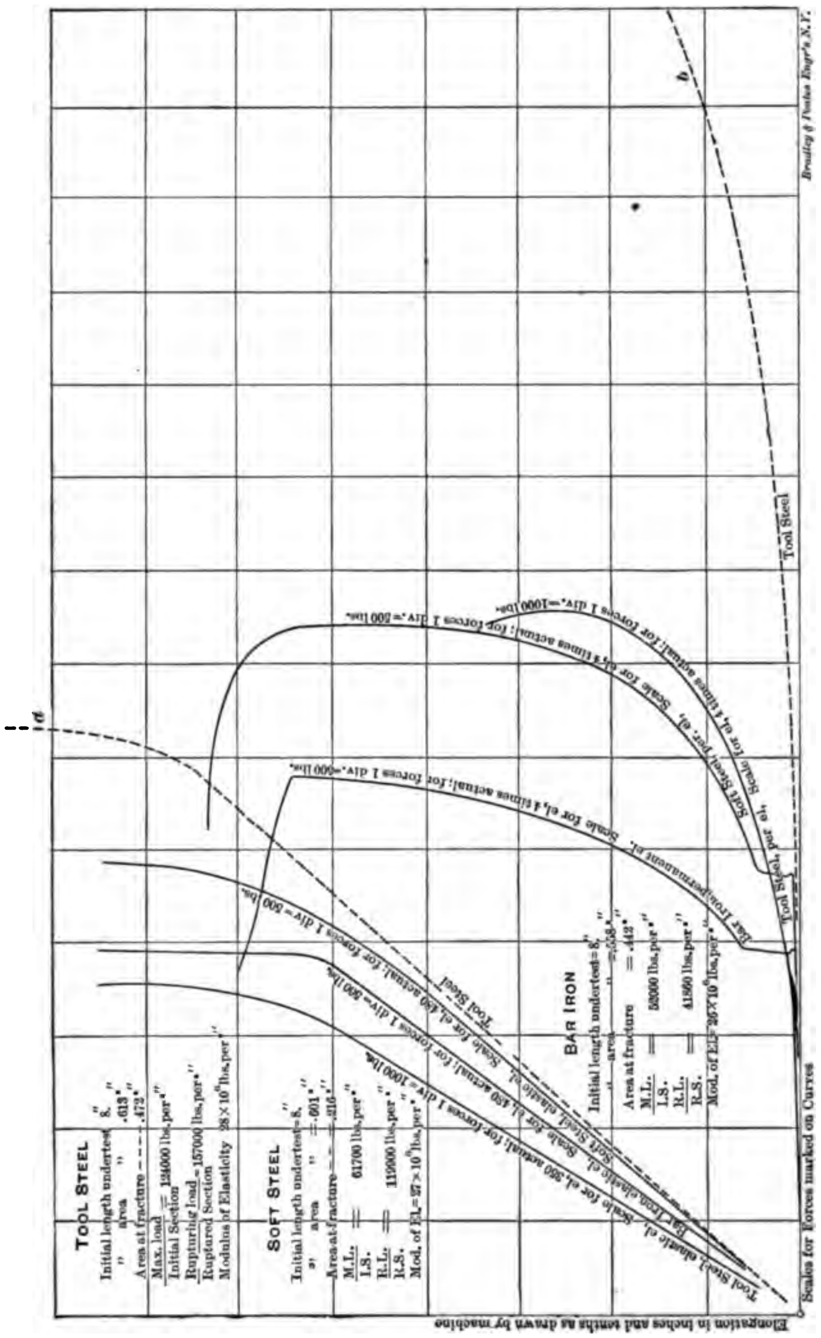
TOOL STEEL
SOFT STEEL
BAR IRON
Tool Steel
Scales for Forces marked on Curves
Elongation in inches and tenths as drawn by machine

Fig. 200.

concave to the same edge. As these curves were not all drawn to the same scales, I have re-drawn them so as to be comparable, and by means of these, as shown by dotted lines *a* and *b*, Fig. 200, and the legends on the original diagram, I find considerable discrepancies for which there is no explanation in the paper.

The modulus of elasticity given in the legend on the plate, Fig. 173, is given for tool steel at 28,000,000, for bar iron at 26,000,000, and for soft iron at 27,000,000. Now these results are certainly not in agreement with those obtained by many observers from innumerable tests, and the large curves on the Fig. 173, when drawn to the same scale as at *a*, Fig. 200, do not give these results as the angles between the curves of bar iron, tool steel, and soft steel clearly show. The correct modulus is obtained by finding the tangent of the angle made by the straight part of the curves with the horizontal multiplied by a constant.

The modulus for soft steel is probably given too low, while for bar iron it may be correct; it would be interesting to know how those values were obtained; that for tool steel is undoubtedly too low. Again, assuming that the origin of the small curves is at the lower left hand corner of the square, it becomes evident that the elastic limit, as shown by the sudden break in the curves, is too low as compared with the maximum load. From the diagrams, taking the tenacity as 48,100 lbs. given in the case of bar iron, and as 66,660 for soft steel, we obtain directly by measurements, elastic limits of 31,800 lbs. given and 42,000 lbs. given, both of which are undoubtedly too high for the material described. If on the other hand the curves have their origin at the points where the lines actually begin on the figure in the third square, then the elastic limits would be considerably too low in both cases, viz.: about 21,600 lbs. and 31,250. Hence, under each assumption, the curves given in the paper would lead to error or misunderstanding.

There is, however, another point in the small curves, which we have to discuss,—namely, that part which represents the breaking period of the test, during which local elongation and contraction take place. By measuring the amount of elongation during these periods in the two cases of bar iron and soft steel, we find that, assuming the total elongation of soft steel represented by the curve to be 26.6%, the local elongation is only 4.16%; while in the case of the bar iron, assuming the total elongation as 18.7%, we find a local elongation of only 1.5%. Both of these values, 4.16% for soft steel, and 1.5% for bar iron, are too small by several per cent., and the question which presents itself is, What is the cause of these differences and discrepancies? Unfortunately the exact method of determining the equation or factor of error has not been given; and from the design shown, it must be doubted whether any good and reliable method can be devised for obtaining it. The question of back-lash, play, and elasticity of different members of the apparatus, as well as temperature changes, play such an important part in the prompt and correct action of this apparatus, that I am inclined to believe that other means shall have to be obtained for producing the results for which this apparatus was designed, and which are so much desired.

Prof. Thos. Gray.—I have to thank Mr. Henning for drawing attention to some inconsistencies between the diagrams and the legends attached to them. I am sorry that, owing to the apparatus in the form described in the paper being incomplete when the description was written, the part containing the diagram was prepared hurriedly from the first tests made with it, and, even then, so late that I was unable to see a proof before the meeting. I find that the sections of the soft steel and iron specimens, the diameters of which are correctly stated in the text, had been interchanged in copying the diagrams, and therefore all the results involving these were wrongly stated in the first form of the paper. Besides this error, I find that the ratios of the levers were not what I had intended and assumed them to be. When corrections are made, as is done in the revised paper, for these errors, the moduli are more nearly in accordance with ordinary experience. The tests show the specimen to have been somewhat abnormal, especially with regard to the relation of the load at the yield point to the maximum load. This is not a fault in the diagram, but a fault in the material, and is a peculiarity which I have frequently noticed in ordinary commercial bar iron not made to specification. The peculiarity at the yield point of the tool-steel specimen is real, and has been frequently obtained both with this and other forms of apparatus for certain kinds of tool steel. Mr. Henning remarks that the diagram just beyond the yield point should be convex upwards. This is no doubt frequently, indeed almost always, the case when the load is continuously increased, but it is not so if the yield is allowed to take place before any addition of load is made. This latter method was adopted in these experiments, and I prefer in most cases to so make the test. The lever which draws the diagram of permanent elongation is very rigid, and is absolutely controlled by the specimen. The latter remark also bears upon the shape of the top part of the curve just before breaking. The tendency is in fact to make this part too steep on account of the inertiæ of the levers of the machine itself. As a matter of fact, however, the character and extent of the local contraction is somewhat variable.

The arrangement of the levers and the knife edge bearings are such as to insure the same kind of certainty as to repetition of the same constants as we have in any ordinary balance. No trouble, in fact, arises from this cause. As to derangement during the test, the principal difficulty with a sensitive attachment is at the time of rupture. At this time the sensitive part of this recording apparatus is out of connection with the specimen, and hence it has simply to bear the shock, due to its own inertiæ, when the machine recoils. With the present form of apparatus this shock is taken up without the slightest trouble.

CCCCXCIX.*

TWO-CYLINDER VERSUS MULTI-CYLINDER ENGINES.

BY SAMUEL M. GREEN, HOLYOKE, MASS., AND GEORGE I. ROCKWOOD, WORCESTER, MASS.
(Members of the Society.)

IN a recent issue of a technical journal, † the theory was advanced by Mr. Rockwood that more than two cylinders in a compound "multi-cylinder" engine were unnecessary to secure the highest theoretical economy in the use of steam. This proposition was severely criticised and declared to be inconsistent with the modern philosophy of the steam-engine. It may, therefore, be interesting to the members of the Society to give their attention to an account of a series of tests of a triple-expansion engine, so constructed as to permit "cutting-out of the circuit" the intermediate cylinder, and running the high-pressure and low-pressure cylinders as a two-cylinder compound, using the same conditions of initial steam-pressure and load.

ENGINE.

The engine is a triple-expansion, condensing engine, designed by George I. Rockwood for the Merrick Thread Company, Holyoke, Mass., and built by the Wheelock Engine Company, Worcester, Mass. The high-pressure and intermediate cylinders are tandem on one frame, the low-pressure cylinder occupying the right-hand position to an observer standing at the cylinder and looking toward the shaft.

The relative proportions of the cylinders are somewhat novel. As the objects of the designer were to secure an engine of symmetrical appearance, of uniform turning moment at each crank, and of highest attainable steam efficiency, and also to make it possible to run the low-pressure side with high-pressure steam, in case of accident to the high-pressure side, the tandem cylinders

* Presented at the San Francisco Meeting, May 1892, of the American Society of Mechanical Engineers, and forming part of Volume XIII. of the *Transactions.*

† *Railroad and Engineering Journal,* December, 1891.

were made of shorter stroke than that of the low-pressure cylinder. The high-pressure cylinder was put next to the frame. The exhaust steam from the high-pressure cylinder passes directly into a receiver of the tubular re-heater variety, and thence directly into the intermediate cylinder. Another similar receiver lies between the intermediate and low-pressure cylinders. These two receivers are so connected that the exhaust from the high-pressure cylinder may pass through both into the low-pressure cylinder without going through the intermediate cylinder, the steam and exhaust pipes of which are provided with valves. The first and second cylinders are jacketed on heads and barrels; the heads only of the low-pressure cylinder are jacketed, and all receiver and cylinder jackets contain steam at full boiler pressure. The cylinder jackets consist of cored spaces. The jacket-drips all collect into one pipe 1¼″ in diameter, which discharges into a reservoir, whence it is returned through a steam loop to the boiler, and in no instance are the jackets connected with the cylinder steam-chests.

The valve-gear of the high-pressure cylinder is of a new type, designed to operate gridiron valves under heavy pressures. The valve gears of the intermediate and low-pressure cylinders are, in all respects, such as have been used heretofore on engines built by the Wheelock Engine Company. The governor operates only upon the cut-off mechanism of the high-pressure cylinder, the releasing gears of the other two cylinders having independent hand adjustments. In case of accident to the high-pressure side of the engine, however, means are provided for connecting the governor with the cut-off mechanism on the low-pressure cylinder.

The engine is located at some distance from the boiler (a Manning upright of 175 rated horse-power), the supply pipe being 325 feet in length. A separator, placed about 15 feet from the engine, collects the entrained and condensed water, which is also returned through a steam loop to the boiler. The condenser is of the jet type, supplied from the canal with injection water, which is removed by a direct-connected air pump.

DIMENSIONS OF ENGINE.

	H.P.	I.	L.P.
Diameter of cylinder	12″	16″	24$\frac{13}{32}$″
" " piston rod	2 and 2$\frac{3}{4}$″	2″	3$\frac{1}{4}$″
Stroke of piston	36″	36″	48″
Clearance in percentage of piston displacement.	2%	4%	3%
Inside diameter steam pipe	5″	6″	9″
" " exhaust pipe	6″	7″	10″
Area of steam port	13″	21″	38′
" " exhaust port	16.5″	25″	60″

METHOD OF TESTING.

It was considered unnecessary to make coal measurements, as they have no bearing on the results.

The feed-water was measured in the following manner:

One large tank was employed as a reservoir, from which the feed-pump drew its supply. Above this tank, on a platform, were placed a pair of scales and a small tank which held about 400 lbs. of water. (Just before the trials the scales were sealed by the Sealer of Weights and Measures.) To the beam of these scales was attached a long pointer. They were accurately balanced with the tank empty, and the position of the pointer was noted and marked. A scale weight of 400 lbs. capacity was then placed on the beam, and water was run into the tank until the pointer resumed its balanced position, thus giving just 400 lbs. of water in the tank. A small valve was provided in the side of the weighing tank, so that any water which might run in, in excess of the 400 lbs., could be readily withdrawn. A counter was also attached to the tank, so that every filling would be automatically registered independently of the attendant's registration. In this way an accurate count was kept of all the water pumped into the boilers. The boiler feed-pump was connected only with the reservoir and feed-pipe to the boiler used during the tests. Steam for this pump was taken from other boilers.

During the period of testing, the water of condensation from the jackets was not allowed to return to the boilers, but was drained through pipes connected with the lowest points in each of the jackets, each pipe leading down to a separate reservoir provided with a gauge glass. The discharge pipe, $\frac{1}{2}$ inch in diameter, from each reservoir, was connected with a surface condenser and discharged into a weighing tank. An accurate record was kept of all water drawn from each jacket during each test. A revolution counter indicated accurately the number of revolutions of the engine. Six Tabor indicators were kindly loaned for these tests by the Ashcroft Mfg. Co., of New York. The instruments were all in the best condition and were sent directly from the factory. The manner of attaching the indicators and the pantographs may be seen from the photographs which accompany this paper (Figs. 185 and 186). The springs used in the indicators on the high-pressure and intermediate cylinders were tested under steam pressure with a steam-gauge which had

FIG. 185.

itself just been tested with a mercury column. The springs used in the indicators on the low-pressure cylinder were compared with the mercury column employed instead of a vacuum-gauge. The steam-gauges were also tested with a test-gauge.

For determining the quality of the steam after passing through the separator a Peabody throttling calorimeter was connected with a perforated ½-inch pipe, screwed several inches into the elbow of the steam supply pipe at its point of juncture with the high-pressure cylinder, the connections and calorimeter being thoroughly covered with hair felt.

The following description of the tests will illustrate the manner in which each trial was conducted:

At one o'clock P.M., the engine having been running for fifteen

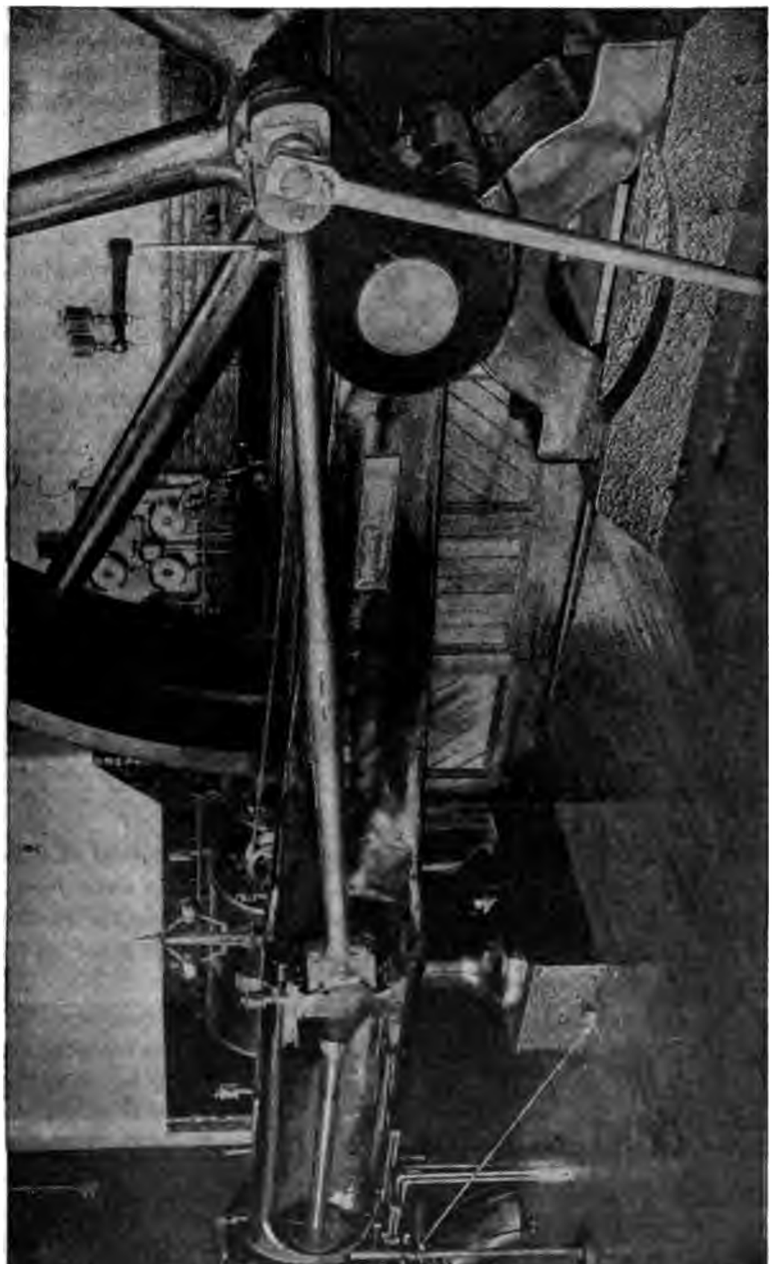

FIG. 186.

minutes, electric bells were sounded in the engine and boiler rooms, the heights of the water in the boiler and in the lower tank were measured, the reading of the scale counter was noted, the heights of water in the various jacket reservoirs were taken, and the test began.

During the trials, simultaneous indicator diagrams lasting ½ minute were taken every half-hour, which was considered often enough in view of the exceedingly steady load on the engine; and pressures and temperatures were carefully noted each time. Every hour the water in the boiler and tank was brought to the heights observed at the time of starting the test, and observations were made for a check on the final result. Just before the time of closing, the boiler pump was stopped, the water in the

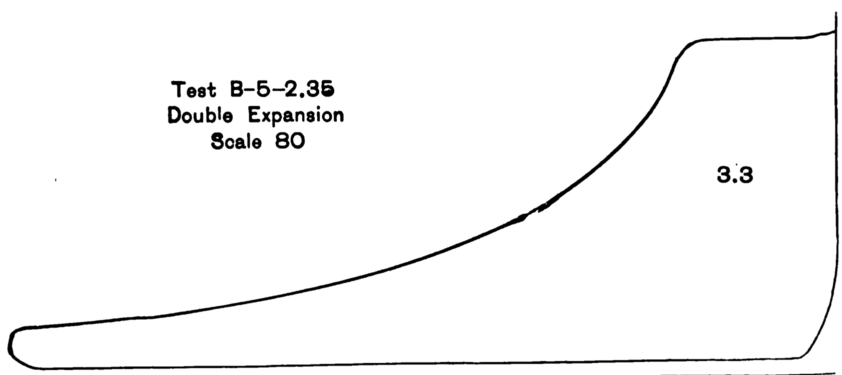

Fig. 175.

boiler was allowed to fall below the point of starting, and at precisely six o'clock the bells were sounded, the engine shut down, and steam was shut off from the jackets. The heights of the water in boiler and tank were brought to the same level as at starting.

Three preliminary tests of the engine were thus run, in order to accustom the attendants to their duties. In all the tests made, the reading of the thermometer in the calorimeter was practically constant, showing a uniform degree of moisture in the steam amounting to 2.64%.

On Wednesday, April 6, two five-hour formal trials of the engine, run as a two-cylinder compound, were held. During Thursday, a holiday, the change was made to a triple-compound, and on Friday two five-hour trials were again made. The sample diagrams appended (Figs. 175 to 184) show the valve-setting

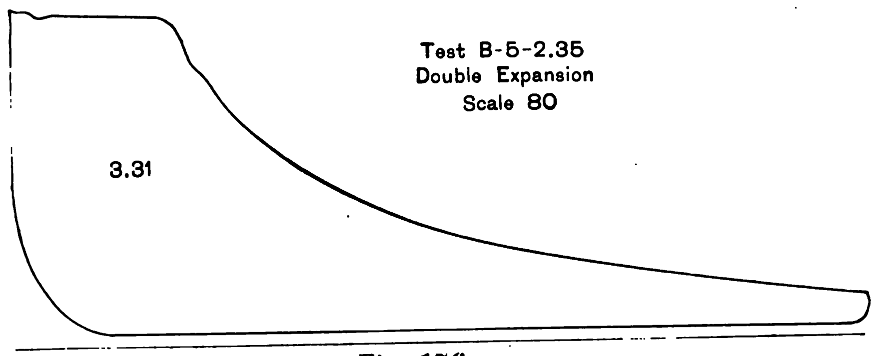

Fig. 176.

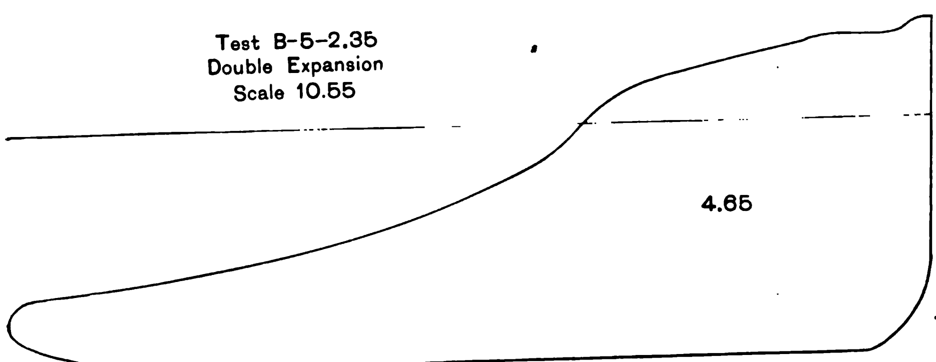

Fig. 177.

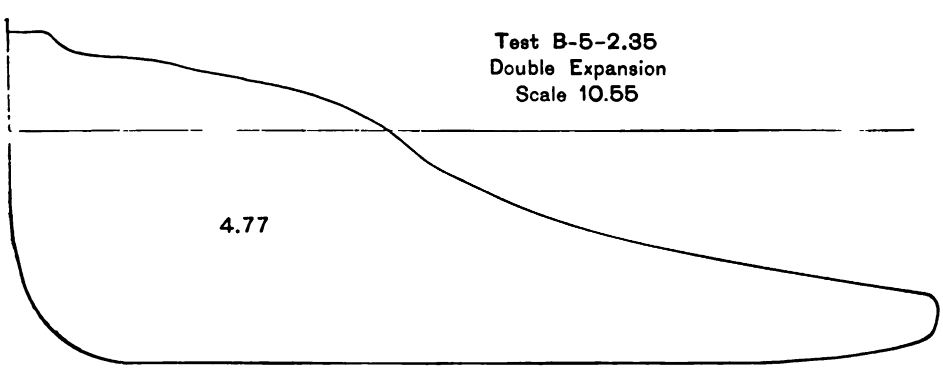

Fig. 178.

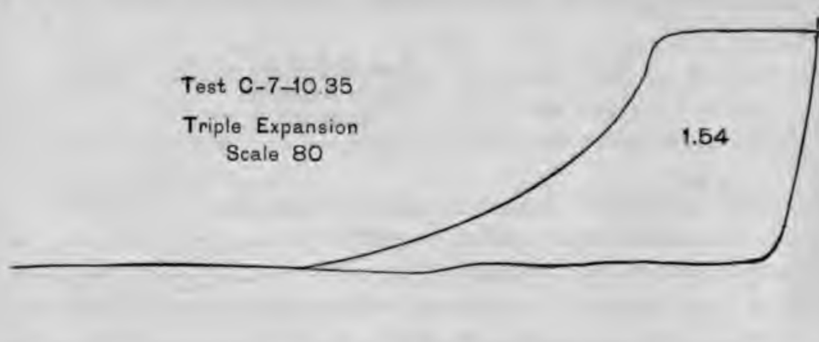

Fig. 179.

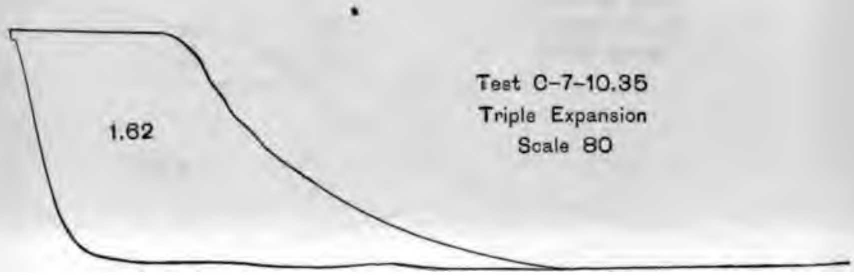

Fig. 180.

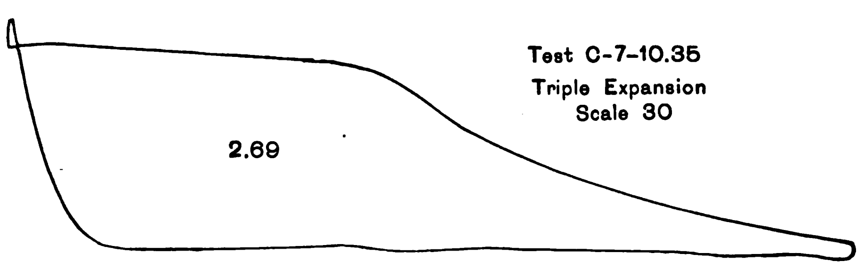

Fig. 181.

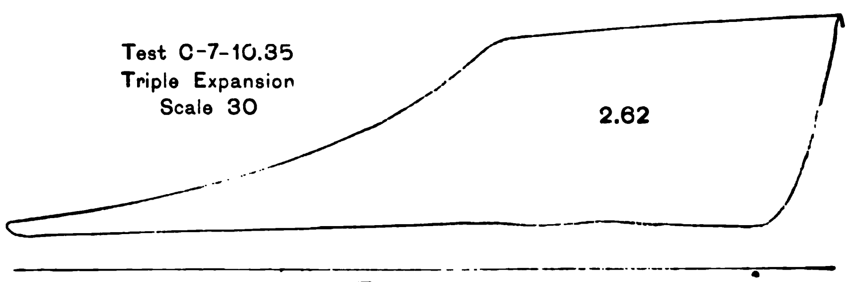

Fig. 182.

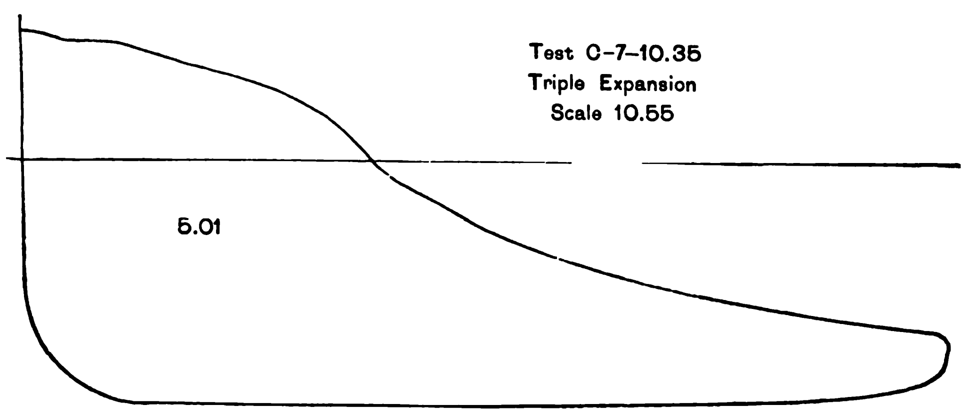

Fig. 183.

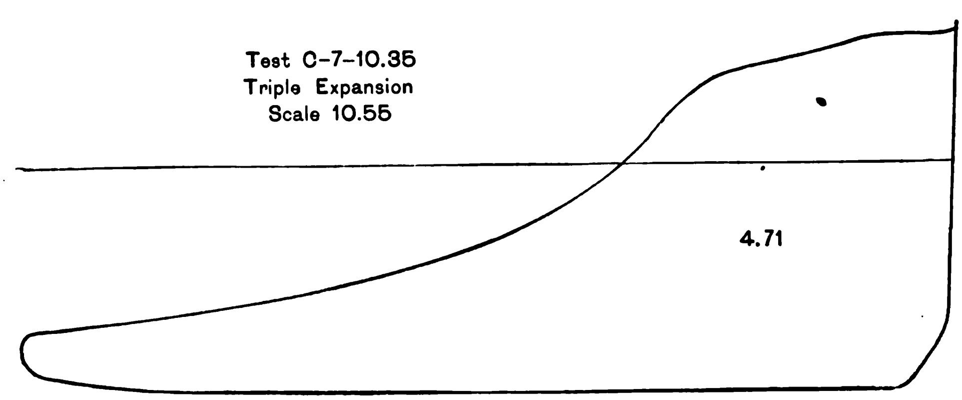

Fig. 184.

and degree of expansion. The results are seen in Table No. 2. They are practically identical, and would seem to support Mr. Rockwood's theory that the receiver may be so constructed as to take the place of the intermediate cylinder or cylinders of the multi-cylinder engine. As these tests were held so shortly before the spring meeting of the Society, the time allowed in which to prepare this paper was much too limited to admit of the exhaustive treatment which the importance of the subject demands. It is hoped that at the next meeting the results of further trials, together with their proper analyses, may be presented for further consideration. But the results of these tests, it is believed, show an economical performance surpassing the best records hitherto published in this country, and clearly indicate that more than two cylinders are unnecessary to secure the highest attainable economy in the use of steam.

TABLE II.

GENERAL RESULTS OF FOUR TESTS OF A TRIPLE-COMPOUND ENGINE, RUN BOTH AS A TRIPLE AND AS A DOUBLE COMPOUND.

Test.	Engine.	Duration, Hours.	R. P. M.	Average Steam-Pipe Pressure.	Average Indicated Horse-Power.	Water per I. H. P. per Hour.	Dry Steam per I. H. P. per Hour.	Weight of Water used in Jackets, per Hour.
						Lbs.	Lbs.	Lbs.
A....	$\frac{12 \times 36}{24\frac{31}{32} \times 48}$	5 (7-12)	79.2	142.	187.11	13.41	13.06	330.3
B ...	$\frac{12 \times 36}{24\frac{31}{32} \times 48}$	5 (1-6)	79.3	142.	180.71	13.11	12.76	330.3
C....	$\frac{18 \times 36}{24\frac{31}{32} \times 48}$	5 (7-12)	79.0	142.	199.08	13.01	12.67	416.0
D ...	$\frac{18 \times 36}{24\frac{31}{32} \times 48}$	5 (1-6)	79.0	143.	178.16	13.25	12.90	388.8

DISCUSSION.

Mr. John H. Cooper.—The experiments of Messrs. Green and Rockwood for ascertaining the fuel economy of two cylinders as against many in one steam engine will prove interesting to the locomotive engineer, whose designs must carry engines, boiler, fuel, water, and attendants wherever they go.

All the working mechanisms being inaccessible while in motion, and most of them subjected to blows, strains, and twistings to which stationary engines are not exposed—for all these reasons the number of cylinders should be reduced to the least, which, of course, require fewest joints and working parts ; and in everything appertaining to the machinery of transmission, the problem must work out to that which is least in wear, repair, and renewal.

It is evident that two cylinders will fill this requirement in a cheaper way than any greater number, and with the economic result which has been reached, as stated above, the road to greatest commercial success in locomotive engineering is made short and easy.

Not intending to be too previous of present statement in a matter which with us is as yet in its experimental stage, a cursory review of the applications of two-cylinder compounds to locomotives, and a statement of some of the results thus far obtained, may advantageously be presented here.

The compound steam engine properly begins with Hornblower's conception and first construction, in the year 1776, the patent for which was granted in 1781, according to English records. In this engine, as both pistons were connected to one end of a pumping beam, they worked in the same direction at the same time.

In 1838, Mr. E. A. Cowper employed a compound engine, the two cylinders of which were connected by a "receiver," and the pistons were united by linkages to two cranks at right angles to each other. This separation of the pistons' motions gained an important and indispensable advantage for marine and locomotive engines over the original type of piston connections, while the lengthening of the "receiver" to a pipe connecting distant cylinders presented no serious objection, but perhaps an advantage, as proven subsequently in locomotive practice—its course being through the highly heated smoke-box.

At that time neither of these plans was adapted to, and perhaps was not thought of for, railway engines ; the steam used was low in pressure and was condensed in both, which would unfit them for train service upon railways.

The direct application of compound steam cylinders to locomotives was made in 1844 by Mr. Thomas Craddock, and was clearly shown by him in his book, published in England, in 1848 ; but he used a high and a low pressure cylinder combined on each side, adhering too closely to Hornblower's design, by which he made a four-cylinder locomotive.

Each pair of cylinders, however, had but one slide valve and one set of gear, and each piston rod was connected to the one crank pin by its own cross-head and connecting rod—an admirable design (which has been repeated since), embodying correct mechanical principles, together with simplicity itself, as to valves and valve gear.

In 1850 Mr. John Nicholson (a "driver") devised and run upon the road perhaps the first two-cylinder compound locomotive, which was further improved and patented by Mr. James Samuel, superintendent of the Eastern Counties Railroad, England, to which he gave the name of "continuous expansion," because its valve arrangements prevented a lapse of pressure on the pistons, which distinguished it from *intermittent expansion*, as employed in the usual two-cylinder type of compound engine.

At this period of locomotive engineering, the belief prevailed that no locomotive could keep up steam without violent blast, and, therefore, to secure this effect part of the exhaust steam from the high-pressure cylinder went to the chimney at once, and part went to the low-pressure cylinder.

This gave an economy of 20% over simple running, where full compound running would have given, as it now gives, 25 to 35%, with ample steam-making capacity, less noise, and less discharge of sparks and ashes.

As less steam is wanted for compound running, and all that is made, in any event, escapes through the chimney, and as economy of fuel goes with uniformity of draft and combustion, which later practice proves, there seems no room for objections to this innovation on the blast problem, and so the Nicholson engine has not been repeated.

These experiments and publications no doubt served a good purpose in calling attention to a new application of an old principle.

The reason why the expansion of steam in two vessels proved more economical than the same amount of expansion in one, was one of the less known facts about steam, and without condensation the problem seemed not worthy of a thought in that day.

It is passing strange how near these early inventors came to making grand successes. They failed rather in the doing of and in the insisting upon the overmuch and in the ungainly in appearance in mechanisms.

From this period down to the present time the patent records teem with contrivances for compounding locomotives, many of which have been completed and put upon the road. Of these the best-proportioned ones are saving in round numbers about the same amounts of fuel, allowing for kind of service. The figures of some of these are given in this article.

For advocates of simplicity in engine and locomotive construction some good names may be quoted. Watt says: "The supreme excellence of machinery is its simplicity."

Mr. J. T. A. Mallet, in his English patent, gives preference to two cylinders, "both for simplicity of construction, and to avoid an excessive amount of cooling surface."

The economy of short steam passages, by which "much waste of steam is avoided," and the exposure of "less surface for atmospheric resistance and cooling—a very important consideration at high speeds"—are points claimed by Mr. Joseph Lewis in his United States patent of 1889.

Mr. Thomas Urquhart says: "What is really wanted is to mature a thoroughly efficient *two-cylinder compound with running parts as light as possible and well-balanced.*"

The greatest advantage sought for by compounding is, of course, the saving of fuel; of this, reliable quotations are here given from results of experience on the road:

	Per cent.
Nicholson & Samuel's	20
Fowler's traction engines	30
Foden's traction engines	29
Burrell's traction engines	30
Worsdell's locomotives	18 to 25
Urquhart's (petroleum fuel)	18 to 28
Borodine	15 to 32
Von Borries	14 to 21
Worsdell & Von Borries (120 engines)	22

These results are sufficient to establish the fact of commercial advantage to be derived from the compounding of locomotives, and all of them deduced from the two-cylinder type; not saying, however, but that three or four cylinders have done and will do as well for saving fuel. The point aimed at here is: If two cylinders will secure an economy commercially equal to that obtained by the use of a greater number, then two cylinders are enough.

For the necessity of superior simplicity there needs be no special pleading. Resulting from all this it appears that the best type of compound engine for locomotive service is the one which has two cylinders, one on each side, and these will fully meet most of the cases in practice.

With the ordinary link and valve motion to each cylinder there will be no increase of complexity for the compound beyond that of the present popular high-pressure, twin-cylinder type of simple locomotive, which far outnumbers all others.

With proper proportions of cylinder capacity and points of cut-off for each, for the service required, and with the addition of one or more valves, hand worked from the cab or automatic combination, or both, as already in use, for permitting steam directly from the boiler to enter the low-pressure cylinder at will, at starting, or during any length of time on the road, and for simultaneously equalizing the total pressure of the working steam on the two pistons while running as a simple engine, the compound locomotive in these respects may be called practically perfect.

Prof. D. S. Jacobus.—The results given in the paper are in my judgment misleading, because the conditions under which the engine was run are more favorable for the compound than for the triple engine. It will be noticed that the expansion line of the indicator cards of the high-pressure cylinder, when the engine is run as a triple expansion, meets the line of back pressure when the piston has travelled about 5/8 of its stroke. In such a case, there ordinarily is a loop formed at the lower end of the expansion line, but there is none in the present case, so that the cards are either in error, or there has been a peculiar variation of pressure in the receiver combined with the lifting of the exhaust valves, thereby admitting exhaust steam, which has caused a constant pressure to be preserved in the cylinder during the latter portion of the stroke. In either case the conditions are unfavorable to the economy of the engine, for if there is a loop which has been omitted in tracing the card, there is a loss on account of the negative work; whereas, if the exhaust steam entered the cylinder during the last 3/8 of the stroke, there is an additional loss due to cylinder condensation. Other criticisms may be made on the general method of testing, such as the variation of the height of water in the boiler, with the steam flowing at its full velocity to the engine, and when the latter is shut down so that no steam is passing from the boiler.

These were the conditions at starting and ending the test, and any variation produced thereby in the apparent height of water in the boiler affects the final result.

Up to the present time the most economical results which have been reported for a compound engine are those obtained by Professor Denton, and afterwards verified by Mr. Kent and the speaker, for the Pawtucket pumping engine. The results obtained by Messrs. Green and Rockwood, however, show a still higher economy than that obtained for the Pawtucket engine.

In order that we may discuss more intelligently the results of Messrs. Green and Rockwood, the following data have been collected :

TABLE A.

COMPOUND ENGINE.

Class of work.	Authority.	No. of expansions.	Boiler pressure in lbs. per square inch.	Vacuum in inches of mercury.	Dimensions of cylinders in inches.			Revolutions per minute.	Total horse-power.	Steam per hour per H. P.
					Bore H. P.	Bore L. P.	Stroke.			
Pawtucket pumping	Denton.	16	127	27.9	15	30¼	30	49	146	13.7
Ferry-boat Bremen, double compound.	Denton & Jacobus.	10	98	26.4	20	36	28	115	778	18.1

TABLE B.

TRIPLE EXPANSION.

Class of work.	Authority.	No. of expansions.	Boiler pressure in lbs. per square inch.	Vacuum in inches of mercury.	Dimensions of cylinders in inches.			Stroke.	Revolutions per minute.	Total horse-power.	Steam per hour per H. P.
					Bore H. P.	Bore Int.	Bore L. P.				
Iron mill.	Schröter.	22	145	28.0	11.1	18.0	27.6	39.4	70	200	12.6
Electric station.	Henthorn.	16	128	26.5	14	25	33	48	99	516	12.9
Ferry-boat Bergen.	Denton.	9	114	27.0	18¼	27	42	24	145	665	18.3

The number of expansions in the tests by Messrs. Green and Rockwood is about 25, and the boiler pressure about 142 lbs., so that the conditions as a compound engine are more favorable for the economy of steam than those under which the Pawtucket engine was worked.

The figure of 12.9 lbs. of steam per hour per horse-power, is therefore not impossible for the compound engine, notwithstanding the fact that it is 6% lower than the result for the Pawtucket engine. In the triple engine, however, the figure 13.1 lbs. of steam per hour per horse-power is probably too high and is certainly unreliable, and therefore should not be used in making comparisons.

When both the compound and triple engines are worked to their best advantage, the results given in Tables A and B show that there is a gain of about 1 lb. of steam per hour per horse-power in favor of the triple engine. If not worked to the best advantage, however, the economy of the triple may fall down to or below the economy given by the compound engine. To illustrate this, line 2, Table A, and line 3, Table B, have been presented. These are the records of tests made on ferry-boat engines, one of which is a triple expansion and the other a double compound. The ratio of expansion is about 10, and economy the same in each. This is probably due to the fact that for a small ratio of expansion the loss by wire-drawing through the valves of the intermediate cylinder offsets the gain due to working the steam in each cylinder between a smaller range of temperature than would be obtained with two cylinders.

This exception does not modify the error in conclusion of Messrs. Green and Rockwood, for they make the broad statement that more than two cylinders are unnecessary to secure the highest attainable economy in the use of steam, which statement cannot be substantiated by means of the experimental data now at hand.

Prof. H. B. Gale.—In regard to the point which has just been raised about a drop at the end of the expansion in the high-pressure diagram of a compound engine, I think it is very easy to prove beyond a question that a certain amount of drop is beneficial. Of course, there might be a question as to how much drop there should be, but it has been stated once or twice before this Society that any drop at the end of the expansion in the high-pressure cylinder is a disadvantage. I think that certainly there is an error there, as can be shown by referring to Fig. 198. There is, of course, no doubt that the larger the cylinder of an engine is, the more cylinder condensation there will be; and it is also true that in any engine cylinder it takes a certain amount of pressure to overcome the friction of the moving parts, and that when your expansion is carried down so near the back-pressure line that the difference is just enough to overcome the friction, then enlargement of the cylinder to admit of further expansion must be a loss. Beyond that point the steam is not accomplishing enough to overcome the friction of the engine, and the energy of the fly-wheel is being absorbed in overcoming useless resistance, so that, even if there were no condensation at all in the cylinder, there would be a point, as A, about $1\frac{1}{2}$ lbs., more or

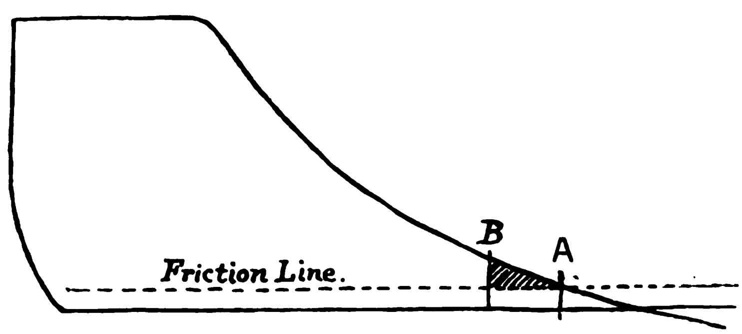

FIG. 198.

less, above the back-pressure line, where further expansion would cease to be beneficial.

Now, the point where it would actually be best to limit the stroke of the piston would be back of that necessarily, because the larger the cylinder is made the more condensation you will get; and in this part of the card the only available pressure to do any actual good is the pressure above that required to overcome the friction. In the latter part of the stroke, the effective pressure, above the friction pressure, is very small, and the useful work done is very small, but the condensation will be increased practically in proportion to the additional cylinder surface. Now, if you cut the cylinder off at the point B, the amount of useful work which you have lost over what you would have if you cut it off at the point A is indicated by the small shaded area. On the other hand, if you make the cylinder that much shorter, you save a proportional amount of condensation. Now, it is very evident that you reach a point somewhere here where the loss by condensation in increasing the size of the cylinder will more than balance the gain of work; so that the most advantageous point to stop the expansion will be back of the point where the pressure equals the pressure required to overcome friction. From my own experience and observation I think that it is generally advisable to have this pressure at the end of the expansion from 6 to 10 lbs. above the back pressure; and that to expand any further is generally a loss. Of course, the thing can be looked at in this way: If you stop the expansion there, you make the cylinder so much smaller; and if you carry the expansion further, you make the cylinder larger, increasing the condensation and doing very little more work. That would apply, of course, to a single-cylinder engine, compound, or triple expansion. The same reasoning applies to any one cylinder of any of those engines.

*Mr. E. T. Stut.**—I have had a little experience to test compound in comparison with triple-expansion engines. Those tests were not made on any fine scale, nor have I had time yet to make as good a test as I wanted. We have a set of triple engines on one of our cable roads. The horse-power varies considerably; sometimes they may make about 20 horse-power, and a minute or two after make probably 500. The prediction was that the compound probably might do better than the triple in this particular case, and we have run them for quite a time compound—that is, using the second and large cylinder. The cylinders are 14 inches initial, 20 inches the second, and the third cylinder is 30 inches in diameter by 54 inches stroke. The result was very good to run them compound with the second and large cylinder, but after running a good while we found that, running triple, the result was considerably better than compound. We also have run the first and large cylinders compound, but we have found that they run a little more economically with the second and large cylinders, and also run more regularly with the second and large cylinders than they do by running with the first and large cylinders. But running them triple expansion, they run most economically and most regularly. The boiler pressure is from 140 to 155 lbs. triple, and running it

* Of San Francisco; by invitation.

compound we run from 135 to 140 and 145 lbs. So, taking this experience as a basis, we came to the conclusion that to run it triple was a great deal better. It runs more steadily also. But we have another thing in connection with these engines, and that is, controlling the cut-off of the steam in all three engines; this we find very important. The cut-off can be adjusted on the second as well as on the third cylinder independent of each other, but the governors keep perfect control of the cut-off for each cylinder, so that the steam is checked to a certain extent in the receivers. Now, all those things we have to consider in comparing triple to compound engines. Also the valve motion is adjustable. Each valve has an independent eccentric; the compression, lead, and exhaust can be regulated independent of each other. Taking advantage of all these things combined make the triple-expansion more efficient than the compound.

Looking at these cards as Professor Jacobus pointed them out, it seems to me that the cut-off is entirely too short in each cylinder. We all know that the condensation in each cylinder has a great deal to do with economy; that is to say, if you cut off very early, the internal condensation becomes so much that we lose by that what we might gain on expansion, and I have come to the conclusion that the cut-off was too early; you cannot form a definite conclusion in the matter from what is stated in the paper. So, taking the tests as published here, I think they are not very reliable to form a definite conclusion. Perhaps the gentleman did his best in making the tests, but I don't think we can take that as a basis. Neither would I take this particular case I mention as a basis.

The Secretary.—I should like to ask how those cylinders are jacketed; by live steam—all three?

Mr. Stut.—All by steam—each cylinder as well as on the receivers—and we find that the jacketing practically saves considerable also.

The Secretary.—Are the heads jacketed?

Mr. Stut.—All the back-heads are jacketed. They all have live steam directly. I tried also on another cable road, on compound engines, the plan of having the steam run from the first cylinder into the receiver, and then the expansion cylinder one after another. I had it connected also in such a way that the steam would run in at high pressure direct also, and I found that running it direct was much the best. So I shut the valves for running the steam from one to the other. It was a question in my mind, until three or four years ago, which was the best; but we found by practical experience in these cases that running the steam in by direct high pressure is the best for jacketing.

Prof. G. I. Alden.—I may be allowed to say, in regard to the cards to which Professor Jacobus has referred, that the explanation which is given by Mr. Rockwood is this: as the cut-off in the high-pressure cylinder was rather early, the valve between the high-pressure cylinder and the receiver lifted as soon as the pressure in the cylinder dropped to that in the receiver, so that there was no loop, and, as far as the indicators are concerned, the test with the triple is just as reliable as the test with the compound, because the same apparatus was used. The lifting of the valve accounts for the straight line on the card. This is Mr. Rockwood's explanation, and, I think, is the correct one. While it is clear that the triple engine was not run under the very best conditions for high efficiency, I presume the test is just as correct and accurate in the triple engine as the compound. There are several very important questions which are raised by the suggestion of Mr. Rockwood and the article which he published, and his theory would indicate that you might have a high-pressure cylinder taking very high-pressure steam, and expand the steam as little as you please in the high-pressure cylinder; then drop it to the pressure in the receiver and use the low-pressure cylinder under the same condition in the triple engine as near as possible as in the compound—that is, have the receiver pressure the same; and, as there are heaters in the receiver, very likely you could have the steam dry in each case. Now, the question arises, Would a drop from the high-pressure cylinder into the receiver be detrimental? In a paper read last night,* it was said that any drop of that sort was detrimental, and that you could not recover what you lost by any subsequent manipulations. This theory suggests that you might have that drop and expand the steam in the receiver and still get the same result from the expansion. I give now, not my own ideas about it, but Mr. Rockwood's ideas, in order that we may have a chance to think more about the whole matter, and I hope that some discussion may be had before the record is published relative to this point. I hope I may be able to add something myself

* J. M. Rites, "Steam Distribution in the Form of Single-acting Compound Engine." No. 495 page 557.

which I had not time to prepare before coming here, as I did not receive the paper before leaving home. One question which arises is this: If the drop from the high-pressure cylinder to the receiver is detrimental, just *how much* do you lose theoretically by the drop? And also the question, Is it all right to expand the steam in the receiver? One point which Mr. Rockwood makes which must not be lost sight of is, that you may so design the receiver as to make the compound engine equally efficient with the triple engine. He would put emphasis upon that point, so that other tests where the receiver was quite small might not be as favorable to his theory as a test with a large receiver like the one on his engine. At this meeting we have two papers, one presented last night, saying that the receiver is practically a clearance space, and that it is detrimental and should be made as small as possible. Mr. Rockwood says that to carry out his theory he must have the receiver large. I emphasize those points in order to bring them to the attention of members who may wish to discuss these important questions, which the paper of Messrs. Green and Rockwood suggests.

* To get some basis for an answer to the question, How much is the theoretical loss due to drop in a given case? the following problem is proposed.

Given:

Initial pressure in high-pressure cylinder............	180 lbs. absolute
Pressure at end of expansion in high-pressure cylinder.	60 " "
Initial pressure in low-pressure cylinder.......... ...	24 " "
Drop...............	36 " "
Steam used per stroke.............................	1 lb.

Volume of low-pressure cylinder equals volume of one pound of steam which has expanded adiabatically from dry steam at 180 lbs. pressure to 6 lbs. pressure.

It is required to find the work done in this case as compared with the work which would be done by one pound of steam at the same initial pressure, and expanding adiabatically without "drop" to the same final volume. This problem is solved by finding, first, the total heat of one pound of steam after it has expanded adiabatically and without "drop" from the given initial conditions to the volume of the low-pressure cylinder; and, second, the same quantity when the steam has experienced the given "drop" and expanded adiabatically in each cylinder

* Added since the meeting.

for running the steam from one to the other. It was a question in my mind, until three or four years ago, which was the best; but we found by practical experience in these cases that running the steam in by direct high pressure is the best for jacketing.

Prof. G. I. Alden.—I may be allowed to say, in regard to the cards to which Professor Jacobus has referred, that the explanation which is given by Mr. Rockwood is this: as the cut-off in the high-pressure cylinder was rather early, the valve between the high-pressure cylinder and the receiver lifted as soon as the pressure in the cylinder dropped to that in the receiver, so that there was no loop, and, as far as the indicators are concerned, the test with the triple is just as reliable as the test with the compound, because the same apparatus was used. The lifting of the valve accounts for the straight line on the card. This is Mr. Rockwood's explanation, and, I think, is the correct one. While it is clear that the triple engine was not run under the very best conditions for high efficiency, I presume the test is just as correct and accurate in the triple engine as the compound. There are several very important questions which are raised by the suggestion of Mr. Rockwood and the article which he published, and his theory would indicate that you might have a high-pressure cylinder taking very high-pressure steam, and expand the steam as little as you please in the high-pressure cylinder; then drop it to the pressure in the receiver and use the low-pressure cylinder under the same condition in the triple engine as near as possible as in the compound—that is, have the receiver pressure the same; and, as there are heaters in the receiver, very likely you could have the steam dry in each case. Now, the question arises, Would a drop from the high-pressure cylinder into the receiver be detrimental? In a paper read last night,* it was said that any drop of that sort was detrimental, and that you could not recover what you lost by any subsequent manipulations. This theory suggests that you might have that drop and expand the steam in the receiver and still get the same result from the expansion. I give now, not my own ideas about it, but Mr. Rockwood's ideas, in order that we may have a chance to think more about the whole matter, and I hope that some discussion may be had before the record is published relative to this point. I hope I may be able to add something myself

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* Added since the meeting.

to the same volume as in the first case. If there is any difference in the two quantities of heat thus computed, such differences must be the heat equivalent of the excess of work performed in one case over that in the other.

The question also arises whether such excess is the same when the steam is assumed dry at cut-off, and when, as is generally true in practice, the steam is wet at cut-off in the high-pressure cylinder.

The problem has been solved for three different qualities of steam at cut-off, viz.: dry steam, steam containing 10% of moisture, and steam containing 20% of moisture. The results are given in the following table:

Initial pressure.	Quality at cut-off.	Quality at 60 lbs.	Quality after drop in No. 2.	Total heat at final volume.	Heat converted into work during forward stroke.	Percentage of work lost by drop.
No. 1. 180	100 %	93.63 %	......	969.13	330.05	....
No. 2. 180	100 %	93.2 %	93.5 %	1053.09	227.99	29.04
No. 1. 180	90 %	86.7 %	......	896.77	269.93	....
No. 2. 180	90 %	86.7 %	89.1 %	938.58	248.12	14.42
No. 1. 180	80 %	76.3 %	......	882.43	260.90	....
No. 2. 180	80 %	76.3 %	79.0 %	861.70	231.52	11.36

No. 1 refers to expansion without drop; No. 2 has 36 lbs. drop, or from 60 to 24 lbs. absolute. The quality is given in per cent. of dry steam, the pressure in pounds absolute, and the heat in British thermal units.

From the table it appears that in the problem considered there is a considerable loss of work on forward stroke due to drop; also that both the percentage and the total amount of such loss is less when the steam is wet at cut-off than when it is dry. Assuming dry steam in the steam chest, the steam at cut-off in the actual engine will be wet, owing to initial condensation. In addition to the drying of the steam due to drop will be added the drying due to the heat given up by the hot surfaces during expansion and drop. How this will affect the loss due to drop as computed for adiabatic expansion, it may be difficult to state accurately. The only offset to this theoretical loss due to drop is the advantage due to the comparatively slight drying of the steam which drop accomplishes. Dry steam in the low-pressure cylinder at release is necessary to the minimum loss by escape of heat during exhaust. Without heating-coils in the receiver it is impossible to get dry steam admitted to the low-pressure cylinder, since drop does not dry out as much moisture as adiabatic expansion generates in the steam before the drop occurs.

The above solution can be applied to any other given or assumed case.

*Mr. George I. Rockwood.**—The writer was compelled to be absent from the meeting of the Society at which this paper was read, otherwise he might have explained satisfactorily certain points which the brevity of the paper left somewhat obscure.

Prof. H. B. Gale illustrated clearly the relation of "drop" to the *mechanical* efficiency of the compound engine.

Mr. John H. Cooper, in referring to the history and possibilities of the compound locomotive, adduces these tests as tending to show that the bi-compound locomotive is as economical of steam as one with any greater number of cylinders could be. The writer would point out that relatively to the volume of high-pressure cylinder, the low-pressure cylinder and the intermediate passage-way from one cylinder to the other of the compound locomotive are generally made so small that the greatest advantage of compounding is lost.

Prof. D. S. Jacobus concludes that the results given in the paper are misleading, as in his opinion the conditions under which the engine was run were more favorable for the bi-compound than the tri-compound. It is not, however, made clear by the critic why increasing the number of expansions should have a more unfavorable effect upon the tri-compound than upon the bi-compound. According to his statement, it is better when the boiler pressure is 142 lbs., and the number of expansions as great as 25, to run a bi-compound engine rather than a tri-compound, as the loss by cylinder condensation will in this way be reduced. This is certainly discordant with the common theory of the peculiar advantage of the multiple-cylinder over the two-cylinder engine, the central idea of which theory is that increasing the number of cylinders decreases the internal losses. As a matter of fact, the number of expansions shown by the cards illustrating the tri-compound test, instead of being 25 is 29; while the number of expansions in the case of the bi-compound is nearly 31. Speaking of the fact that

* Author's closure.

the expansion line of the high-pressure cylinder diagram meets the back-pressure line at a point about $\frac{5}{8}$ of the stroke, and then runs out on that line, nearly coincident with it; the statement is made that either there was a loop formed on the actual card by the expansion line running under the back-pressure line, which has been overlooked in tracing the card, or that the indicators were inaccurate and the tests therefore unreliable. As Prof. G. I. Alden pointed out, and as the authors stated, the apparatus for taking the diagrams was the same for both tests, and the indicators were found to be practically without error. (Professor Alden will permit the writer, at this point, to correct a part of the explanation of the excessive back-pressure in high-pressure cylinder which in the report of the discussion Professor Alden attributed to him.) The exhaust valves are long, narrow, multiported slide valves, somewhat similar to those used in the Strong locomotive. When no steam is present in the cylinder, these valves are held to their seats by gravity alone, and serve, of course, as relief-valves when the back-pressure is any greater than the forward-pressure.

The normal load which this engine was designed to carry is 200 H.P. For an hour or so in the evening, however, the engine has also to drive 100 H.P. of dynamos for lighting the mill, so that the maximum horse-power of the engine must be well above 300. To enable the engine to drive this load the high-pressure cylinder was made larger than otherwise would have been necessary, as the valve gear is actuated by a single eccentric, and the latest cut-off which can occur is at a point less than $\frac{5}{8}$ of the stroke. Necessarily the effect of making this cylinder of greater volume than that of the intermediate cylinder up to the point of cut-off is to discharge a greater weight of steam into the large intermediate receiver than the intermediate cylinder can take out of it without taking it out at a greater pressure than the normal terminal pressure in the high-pressure cylinder. This is obviously the reason why the receiver pressure is so high as to open the exhaust valves of the high-pressure cylinder at $\frac{5}{8}$ of the stroke, and why there was no loop formed in the indicator card. Now the ratio of expansion of the steam in the high-pressure cylinder of the tri-compound is much less than the ratio obtained by dividing the length of the admission line by the whole length of the card; it is evidently found by dividing the admission line by the length of the card, minus the straight line extension. So instead of there being five or more expansions in the cylinder, there are really less than three. And it is difficult to understand how the cylinder condensation can be affected by this constant pressure and temperature phenomenon, either favorably or unfavorably.

It is further suggested that other criticisms might be passed on the general method of testing. The writer can naturally deal only with the single criticism offered, that having reference to the manner of stopping the tests by allowing the water line in boiler to fall a little below the level at time of starting, and when the engine was shut down, raising the water line exactly to this level. This criticism can hardly be taken seriously, as the water level was observed not to fluctuate when the engine was shut down, and at any rate the amount of water thus pumped into the boiler was an insignificant amount in any one of the tests reported, and was against the economical performance of the engine. In reference to the paper it will be seen that the authors made no such broad statement in conclusion as is attributed to them. The words advisedly used in concluding the report are as follows: "The results, *it is believed*, clearly indicate that more than two cylinders are unnecessary to secure the highest attainable economy in the use of steam." The authors fail to see anything in the arguments proving or tending to prove that "the figure 13.1 is probably too high, and is certainly unreliable." On the contrary, indeed, in the opinion of engineers qualified by very long and wide experience to judge, the data here reported are no less reliable and accurate than any similar data which have ever been collected. Many of the criticisms therefore seem groundless, and doubtless would have been withheld on a more intimate acquaintance with the engine.

Professor Alden has shown exact apprehension of the real question at issue in his clear statement of the theoretical loss due to "drop." There still remain other theoretical considerations to receive due weight in arriving at a knowledge of the absolute effect of "drop." But it was not the intention of the writer to dwell upon them at this time. The authors contemplate holding another series of tests to discover the curve of efficiency of this engine with varying loads.

An engine of some 900 H.P., designed in accordance with the writer's theory, will be available for trial at an early date.

D.

MEMORIAL NOTICES OF MEMBERS DECEASED DURING THE YEAR.

ALFRED C. HOBBS.

[NOTE.—The special interest which attaches itself to the relation of Mr. Hobbs to the standing of American mechanical experts among foreign nations, as the result of his labors at the Exposition in London in 1851, made a more extended notice fitting and desirable than the brief notice usual in the volume of *Transactions*. Such a notice was therefore prepared by his friend and admirer, Mr. W. F. Durfee, and will be found at page 263 of this volume.—*Secretary*.]

GEORGE F. REYNOLDS.

George F. Reynolds was born at Somerville, Mass., July 17, 1864. He graduated at the Fort Dodge, Iowa, high-school in June, 1881, and entered the Northwestern University, at Evanston, Ill., in September of the same year. Later he entered the Massachusetts Institute of Technology, from which he graduated in June, 1885, having done five years of thorough college work before his twenty-first birthday.

In July, 1885, he entered the service of M. C. Bullock, of Chicago, where he was employed until July 1, 1889, at which date he started for the gold fields of the South African Republic, and was there engaged in the management of mine prospecting machinery until the time of his death, which occurred on January 9, 1891.

He entered the Society in May, 1889, as junior member.

GEORGE A. PORTER.

George A. Porter was born in 1845; was a graduate of Hamilton College, Clinton, N. Y., 1866. He was the founder of the Porter Manufacturing Company, of Syracuse. In 1888 he moved to Chicago, and there became the founder of the firm of Porter, Jackson & Co. and the Porter Boiler Manufacturing Company, the latter being a corporation formed in 1890. By his energetic efforts he built up a business favorably known throughout the country tributary to Chicago.

His last illness was a short one, about eight days, and he passed away October 5, 1892.

He became an associate member of the Society in the year 1884.

GEORGE H. BLELOCH.

George H. Bleloch, a member of the American Society of Mechanical Engineers,* died very suddenly at his residence in Springfield, Mass., on the morning of November 24, 1891, of neuralgia of the heart. For several years he had been a great sufferer from neuralgia, and had been compelled to spend the winters at the Hot Springs of Arkansas. He made a voyage to Europe in the summer of 1890 for the benefit of his health, which for a time seemed to improve, but for some months previous to his death his heart had been affected. Notwithstanding this threatening phase of his malady, he felt unusually well the day before his death, and was at his office at the National Needle Company, attending to business; but in the evening he had a severe chill, followed by a night of great suffering, culminating in death in the early hours of morning.

Mr. Bleloch was born in Rochester, N. Y., December 28, 1835. His early years were spent on the large farm of his father.

He received his scholastic education in the public schools of Rochester. Being naturally inclined towards mechanical pursuits, he served an apprenticeship at the machinist's trade, but soon after, with his brother, organized the firm of Bleloch & Co., for the purpose of prosecuting the book and stationery business, and at one time this firm had stores in New York, New Orleans, Memphis (Tenn.), Little Rock (Ark.), and Pine Bluffs (Ark).

About twenty years ago Mr. Bleloch became interested in a self-threading sewing-machine needle, and attempted to develop its manufacture at Hyde Park, Mass., but was induced to remove

* George H. Bleloch was elected a member of this Society May 26, 1886. The writer made his acquaintance, and was associated intimately with him for a number of months, as a member of Group 21 of the Board of Judges of the Centennial Exposition of 1876. This group took cognizance of all machine tools for wood, iron, and stone working. The acquaintance then begun has always endured, and he will ever be regarded as among the most honorable and agreeable of the *personnel* in the picture of that grand illustration of the world's progress which hangs on the walls of my memory.—W. F. Durfee, West New Brighton, N. Y.

to Springfield, Mass., where he commenced the manufacture of all varieties of sewing-machine needles in a modest way in leased premises. But the business grew rapidly, and led to the organization of the National Needle Company, which soon erected a factory of its own, to which large additions have since been made, and now the business gives employment to over two hundred persons. In 1888 Mr. Bleloch became a large owner and a director in the Excelsior Needle Company, of Torrington, Conn., but still retained his position as treasurer and general manager of the National Needle Company. He was prominent in the organization of the Sewing-machine Supply Company, of Boston, and became its President. As a fitting recognition of Mr. Bleloch's great mechanical talents, he was elected a member of the Board of Judges of the Centennial Exhibition of 1876, and was assigned duty with Group 21, which was especially charged with reporting on machine tools for wood, iron, and stone working. This was one of the most important of the groups into which the Board of Judges was divided, and Mr. Bleloch contributed in no small degree to the labor of their examinations and reports, and to the remarkable harmony and good-fellowship which characterized their meetings throughout the five months of their continuance. Six years later he served as one of the vice-presidents of the Cotton Exposition at Atlanta, Ga.

In politics Mr. Bleloch was a Democrat, and stood high in the confidence of his party. In 1883 he was nominated for mayor of Springfield on the Democratic and a Citizens' ticket, but failed to secure an election. He was a delegate to the National Conventions of 1884 and 1888, but was unable to attend the latter on account of illness. He was a member of the Democratic State Central Committee of Massachusetts in 1885 and 1886, and took an active and efficient part in its executive work. His advice was often sought, and his opinions had especial weight in local matters, during the administration of President Cleveland.

Mr. Bleloch was an active member of the school board of the city of Springfield for nine years, and the manual-training system in operation in the schools of that city received his cordial approval and zealous support.

The universal esteem in which he was held in the city of his home, and the high appreciation by his fellow-citizens of the great value of his judgment and executive ability in public affairs, was most appropriately illustrated by the unanimous renomination as a member of the school board which he received from Democrats and Republicans, and at his death he was about entering upon his fourth term of three years as a member of that body. As soon as his death was made known to his associates, a special meeting of the school board was called, and appropriate resolutions were unanimously passed, one of which stated that "his unselfish and untiring labors in behalf of our public schools have always been intelligent, earnest, painstaking and efficient."

Mr. Bleloch came of a sturdy Scotch ancestry. He was of stalwart and commanding personality, possessing a dignity of carriage and deportment that attracted attention and compelled respect. He was a notable example of a strictly upright and conscientious business man, who earned an enviable reputation for integrity of purpose, high character, and those essentials of business success—a clear comprehension of all business questions, supplemented by capacity for, and great industry in, the work he had in hand.

HIRAM P. MINOTT.

Hiram P. Minott served his apprenticeship with the Lowell Machine Shop and the Putnam Machine Co., of Fitchburg, Mass. He was with the latter for nineteen years, nine of which were spent in the small lathe department. He acted as superintendent for C. L. Rice & Co., of Chicago, and the Union Iron Works, of Clinton, Iowa, but returning East, became superintendent of the Whitestone Iron Works, where he remained five years, building marine work. At the time of his death he had been proprietor of the Industrial Machine Works for some years, at Columbus, Ohio.

He joined the Society at its Chicago meeting, in May, 1886, and died December 8, 1891.

WINFIELD H. E. GRANT.

Mr. Grant was born November 10, 1861, at East Boston, Mass. He received his education in his native city, and entered business as a specialist in ventilation, heating, and drying appliances.

From January, 1887, until the time of his death, in February, 1892, he had been consulting engineer and manager for the

Boston Blower Co., being in charge of their factory. He moved to Denver shortly after he connected himself with the Society, in June, 1891, after its Providence meeting, and he was a resident of Colorado when he died.

EDWARD M. REED.

Edward M. Reed was born in Lancaster County, Penn., November 17, 1821. His ancestors for two generations at least were residents of the Keystone State. His father, in his early life, followed the profession of an architect and builder.

Mr. Reed's early educational advantages were those afforded by the common schools of his native place.

At sixteen years of age he was apprenticed to a machinist of Lancaster, working in the machine shop owned by Boone & Cockley, and after serving his apprenticeship of four years he was made general foreman of the shop. From this position he wênt into the machine shops of the Baltimore and Ohio R. R., and in 1843 he was appointed a locomotive engineer on the same road.

In 1845 he was appointed by the Philadelphia and Reading R. R. Co. as master mechanic of their Port Richmond shops at Philadelphia. In the same year he resigned, and accepted a position in Cuba on the Havana and Guines Railway, as superintendent and master mechanic.

Three years later, in 1848, he left the West Indies and came to Connecticut, commencing his duties there with the Hartford and New Haven R. R. Co., first as locomotive engineer, but serving in this capacity for only a few months, as within the same year he was appointed master mechanic of the same company, which position he retained until September, 1853.

From the beginning of his railroad experience he took much interest in civil engineering, making it a special study, but had no opportunity to obtain any practical knowledge in this branch of work until the building of the second track of the Hartford and New Haven R. R., between 1850 and 1853, in which he assisted to a considerable extent in addition to performing his duties as master mechanic.

By close attention, and being quick in acquiring the knowledge of the construction and general operation of the road, he was promoted in 1853 to the office of superintendent and engineer. This official position he retained until the consolidation of the Hartford and New Haven and the New York and New Haven railroads in 1872, when he was appointed general superintendent of the consolidated company—the New York, New Haven and Hartford R.R. Co. This position he held until 1874, when he was elected director and vice-president of the company, still continuing as general superintendent.

In 1886 his duties had so increased that, by reason of advancing age and failing health, he found it necessary to resign the office of general superintendent, retaining, however, the vice-presidency of the company until his death, February 13, 1892.

"Forty-nine years of active railroad service," can be said of very few, and it is especially worthy of note, in Mr. Reed's case, that for forty-four years he was continuously with practically one railroad company. His long and successful service speaks for itself and shows plainly the high esteem in which he was held by the company.

During his railroad experience his special forte and delight seemed to be matters pertaining to civil engineering.

Some of the most important structures planned and built by him or done under his supervision are the railroad bridge over the Connecticut River at Warehouse Point, Conn.; the bridge, including draw span, over the Housatonic River at Naugatuck Junction, Conn.; the drawbridge over Pequonnock River, Bridgeport, Conn.; three spans of the bridge, including draw span, over Mianus River, Cos Cob, Conn.; the drawbridge across the bay at Pelham, N. Y.; the stone arch bridge near Berlin, Conn., over the Mattabessett River, built in eight spans of 23 feet each; the stone arch bridge near Hayden's Station, Conn., over the Farmington River, built in seven spans of 55 feet each; the elevated structure at Hartford, Conn., and the tunnel a short distance east of that station, through which run both the tracks of the New York, New Haven and Hartford R.R., and the New York and New England R.R.; the passenger station at New Haven; also the construction of the Suffield and New Britain branches.

He was an apt student, a close observer, and his quick insight was perceptible to all with whom he came into contact.

Yale University, of New Haven, recognized his services and scientific attainments by honoring him in 1885 with the degree of Master of Arts.

THOMAS WILBRAHAM.

Thomas Wilbraham was born June 14, 1827, in Clay Cross, Derbyshire, England. He left his native land with his parents in 1842, and was wrecked on Fire Island, and, while their lives were saved, all they owned was lost, leaving the young man without a cent in his pocket when he landed in this country. He was bound out as an apprentice in the machine shop of James Brooks, at Frankford, Pa., serving also with John F. Starr and Merrick & Towne, and as engineer and machinist with the P. and R. R.R. He was one of a party of machinists selected to accompany the late Mordecai Reed on his trip to Cuba, where he served, among other things, as a locomotive engineer. Upon his return he was caught by the gold fever, and went to California in 1849, remaining there for several years. He was afterward employed by Commodore Cornelius Vanderbilt, and ran the first steamboat on Nicaragua Lake as chief engineer. Having saved a little money, he returned to Philadelphia and engaged in the iron business at the site on which the business remained up to the time of his death. The firm name was first Wilbraham & Whittington, but on the withdrawal of the partner, his two brothers were admitted to the firm, and it became Wilbraham Bros.

He was one of the party visiting England in 1889, and joined the Society on his return at the New York meeting of that year.

He died very suddenly from heart failure, at his residence, March 1, 1892.

SIR JOHN COODE.

Sir John Coode was born in 1816 at Bodmin, Cornwall, England, and came of a distinguished family which has been resident in Cornwall for many centuries. He was educated at the grammar-school of his native town, and early in life chose the profession of civil engineer, in which he soon distinguished himself. He began his career under the eminent engineer Rendel, and for a short time was employed on the Great Western Railway. As early as 1847 he was appointed resident engineer at Portland Harbor and Breakwater, and on the death of Rendel, engineer-in-chief in 1856. He continued in charge until the completion of this great work in 1872, when he received the honor of knighthood. He was for many years consulted by the Board of Trade and other Government departments on matters connected with harbors, docks, rivers, and drainage. Many important works were carried out from his designs, including the great breakwater and docks at Cape Town, the breakwater at Colombo, the improvement of the River Bar in Ireland, the harbors of the Isle of Man, and similar works elsewhere. He was a member of the Royal Commission on Harbors of Refuge in 1858–59, and of the Royal Commission on Metropolitan Sewage Discharge, 1882-83. The harbor of refuge now in progress at Peterhead, and the new harbor works at Dover, are from the designs of himself and his firm. Sir John persistently advocated the construction of a national harbor at Filey, on the east coast, recognizing the great value of such a harbor as a basis of operations for the navy and as a refuge for all vessels in the hour of need. Among his other public services, it may be mentioned that he served as a member of the International Consultative Commission on the Suez Canal in 1884-85. He was president of the Institution of Civil Engineers in the years 1889 and 1890, and received, on behalf of the Institution, the party of about 250 American engineers who visited Great Britain in a body during the summer of 1889. He was one of the Royal Commissioners for the Colonial and Indian Exhibition of 1886, and acted as president of the engineering section of the International Congress on Hygiene, which held its sittings in London in August, 1891. Sir John Coode twice visited the Cape, the Australian continent, and New Zealand at the request of colonial governments, and many of the harbors constructed or in progress there are from his designs. In 1886 he was made K.C.M.G. In recognition of his talent as an engineer, and his warm identification with the interests of this Society resulting from the visit to England in 1889, he was made an honorary member of this Society in November of that year, and retained his interest to the end. He died March 3, 1892.

EDGAR M. BIXBY.

Edgar M. Bixby was born in Boston, October, 1847. He served three years' apprenticeship with the Whittier Machine Company, and as draughtsman and in charge of erecting boiler plants, and for one year as draughtsman in the Bureau of Equipment at Charlestown Navy Yard. He was for ten years with

Moore & Wyman, manufacturers of hydraulic machinery, and one year as manager of Boston office of the Deane Steam Pump Company. At the time of joining the Society, November, 1890, he was manager of the Boston office and superintendent of construction for the Springfield Foundry Company.

At the time of his death was general agent of the Eaton & Prince Company, elevators, Boston. He died of pneumonia, after one week's illness, on the 7th of April, 1892.

GEORGE M. COPELAND.

Mr. Copeland was for three years assistant engineer in the U. S. Navy, and a steamboat engineer for the same length of time. He acted for eight years as draughtsman and designer for Thurston, Gardner & Co., at Providence, and at the Allaire Works he superintended in part the erection of the machinery of the steamers *Pacific* and *Baltic*, of the Collins Line, and that of the steamer *Union*, for Spofford, Tileston & Co., of New York. During the war, 1861-65, he was engaged in superintending the alterations to purchased vessels for the Government use, and was for a year and a half engaged on boiler experiments for the U. S. Navy at the New York yard. For three years he was superintendent of the construction of the St. Louis Water Works machinery, and was for many years superintendent of construction for the U. S. Light-house Department, engaged in fog-signal machinery construction, and machinery for vessels. He was a charter member, joining the Society in 1880, and died very suddenly, April 11, 1892.

J. W. TYNAN.

Mr. Tynan was born in Old Point Comfort, Va., February 22, 1837. At the age of eighteen years Mr. Tynan joined the United States Navy and remained in the service of the government until the breaking out of the war between the States, when he resigned and offered his services to his native State and the Confederacy. He entered the service as an assistant engineer, and rose from one grade to another until he reached that of chief. He was second assistant on the ram *Virginia* (the *Merrimac*) during her famous battle with the *Monitor* in Hampton Roads, in 1862, and afterwards served in other positions in the navy.

Subsequently he entered business with a foundry and machine-shop on general work, and earned a high repute for honest workmanship. He was a skillful mechanic, uniting a thorough understanding of the theory of mechanical engineering with the practice of a trained workman. He connected himself with the society at the Cincinnati meeting in May, 1890, and his death was due to consumption, June 24, 1892.

JAMES MAHONY.

Mr. Mahony was born in the year 1828. When fourteen years of age he was apprenticed to Henry R. Dunham, who had large works on West Street, in New York City.

When his apprenticeship was completed he was made foreman of the blacksmith shop. When the gold excitement was at its height in California, Mr. Mahony started for California.

Mr. Dunham sent a portable shop, with tools, and Mr. Mahony and three young men, to establish a works in San Francisco ; but the young men were anxious to go to the mines, and the project of starting the works was indefinitely postponed. When Mr. Mahony had been in the mines a few months he started a repair-shop for shipwork, which was very successful, making a profit some days of $1,000.

Mr. Mahony was the engineer of the boat which was the first to go up the Sacramento River. He returned to New York about 1852; and was confined to the house for nearly a year with Panama fever. When he regained his health, he accepted the position of foreman boiler-maker at the shops of the Bay State Steamboat Co., at Fall River, Mass., remaining in Fall River some eight years; the works then were removed to Newport, R. I. Mr. Mahony personally superintended the erection of the buildings, derrick, etc.

His position during this time was master mechanic and general superintendent; he filled the position during seven administrations, covering a period of twenty-five years. In the fall of 1878 he tendered his resignation.

Moving to New York, with a view of introducing his patents, among which are the Mahony paddle-wheel, the Mahony patent berth, Mahony patent closet, Mahony grate bar, the Mahony water-tube boiler, the Mahony patent furnace, etc., and meeting with success in this line, he commenced to install steam plants, in some cases furnishing the machinery, in other cases designing and superintending the work.

Mr. Mahony's health first became impaired last fall, with serious attacks of indigestion and heart failure, which termi-

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Mr. Mahony's health first became impaired last fall, with serious attacks of indigestion and heart failure, which terminated on the morning of July 7, 1892. He had connected himself with the Society at its Pittsburgh meeting in 1884.

WILLIAM H. WORTHEN.

William H. Worthen was born at Charlestown, Mass., on November 26, 1842.

He attended the Harvard School as a boy, and in 1860 went as apprentice into Hattinger, Cook & Co.'s machine shop in Charlestown, to learn the trade. Here he served four years, pursuing at the same time scientific and mechanical studies.

In 1864 he entered the office of the well-known ship-builder, Donald McKay, as draughtsman; went later to Hartford, Conn., with Woodruff & Beach, and in 1867 into the engineering office of Norman W. Wheeler, in New York City.

With a desire to enlarge his experience and to become familiar with all branches of mechanical work, at the end of the latter year he secured an important position in the drawing-room of Harlan & Hollingsworth, of Wilmington, Del., where he remained for upwards of three years, leaving there in 1870 to become superintendent of the Morris Co. Machine and Iron Co., of Dover, N. J.

Returning to New York City in 1877, he engaged in general engineering work. It was at this time that he designed, constructed, and put into successful operation the creosoting works of Eppinger & Russell, the first for preserving lumber on a large scale erected in this country.

Early in 1880 he went as superintendent with Guild & Garrison, makers of pumping machinery at Brooklyn, N. Y., and at the time of his death was a member of that firm.

Mr. Worthen was the inventor of several improvements in pumping machinery and condensing apparatus, and had a well-established reputation as an engineer of sound judgment and varied talents. He was elected a member of the Mechanics' and Tradesmen's Society of New York in 1880, and in 1891 became a member of the Society of Mechanical Engineers at its Providence meeting.

He died suddenly on the evening of July 18, 1892.

GEORGE S. PERCIVAL.

George S. Percival was born in 1867 in New York City—graduated in 1888 in a course of civil engineering at the School of

Mines at Columbia College, and upon completing his work as under-graduate he pursued further study in the department of mechanical engineering, making a specialty of compound and multiple expansion engines. He later connected himself with the firm of Westinghouse, Church, Kerr & Co., and was with them at the time of his death. He joined the society as a junior member in 1889.

He was a member of the second company of the U. S. Naval Reserve, and his death was due to pneumonia, attributed to a cold consequent upon exposure during the manœuvres of the Reserves in 1892. He was ill for one week only, and died August 1, 1892.

JOHN A. PRICE.

John Amon Price was born January 20, 1842, at Irvington, N. J. His father moved to Pennsylvania when his son was eight years of age, and his early education was acquired solely by his own efforts, thus fitting himself to enter the Brown University at Providence, R. I. He left the college at the breaking out of the war, at the first call for volunteers, and remained with his regiment until it disbanded at its close. He then came to Scranton, with whose progress and advancement he was closely identified. He served for several years as teller in the Second National Bank, but left this position to found the Scranton Stove Works, and subsequently became interested in the lumber interest in that city. He was the patentee of over one hundred mechanical inventions connected with the stove, furnace and general foundry business, making a specialty of experimentation in solid and gaseous fuels for steam and domestic uses. He was president of the Scranton Board of Trade for many years, and vice-president of the National Board of Trade at the time of his death, devoting himself to the labor of the commission appointed by the State of Pennsylvania to consider the question of utilizing the waste culm scattered over the State. He was also Commissioner for the International Trade Congress connected with the World's Fair.

He connected himself with the society at the Cincinnati meeting, May, 1890, and had been very active at the previous convention at Scranton. He was a member of a number of scientific, economic, geographic and political societies, both in this country and in Europe. He was president of the Lackawanna Institute of Science and History, and an interested member of the American Institute of Mining Engineers. His public spirit was shown by his efforts to organize a movement to make a permanent census department in the Federal Government, which should be the basis of representation, and also in the movement to secure a statistical department for the purpose of a basis of value on which taxation might be more equally distributed. His death was due to peritonitis, the result of a cold contracted by taking a long, fatiguing ride, in the pursuit of his investigations on the utilization of culm. He died August 2, 1892. He held decided views on the abolition of the credit system and the equalization of taxation. He was a noted traveler, and was deeply interested in anthropology, particularly in its bearing upon the origin of nations and the condition of primitive races.

J. ARCHIE TAYLOR.

J. Archie Taylor was born at St. Louis, Mo., in 1857, graduated from the Polytechnic College, State of Pennsylvania, June, 1876, taking the highest honors, and was awarded the first prize for the best graduating thesis. During the winter of 1876–77, he was employed at Chester, Pa., by John Roach, where he remained until 1880, when he entered the service of The Pusey & Jones Co. of Wilmington, making a specialty of marine and general machinery. He was part of the time foreman of the night force.. In 1880 he entered the drawing room, where he remained until his death, August 10, 1892, which was due to consumption, after an illness of twenty months.

He became a member of this Society at the New York meeting in November, 1886.

WILLIAM P. TROWBRIDGE.

William Petit Trowbridge was born at Troy, Mich., in 1828. He graduated at the head of his class in the U. S. Military Academy at West Point, in 1848, acting during the last year as assistant professor in chemistry to supply the vacancy created by the exigency occasioned by the Mexican war; which called for all available graduates to go to the field. For two years after graduation he served in the Astronomical Observatory as assistant, and in 1850 was transferred at his own request to the Coast Survey, where he was assistant to Prof. Bache, then at its head. His work lay upon the coast of Maine and in Virginia; the triangulation in Maine in 1852 was put in his charge. The work in Virginia was upon the James River for the improvement of navigation. He suggested the improvement which in the Civil War was carried out, and was then and is now known as the "Dutch Gap Canal." He was then transferred to the Pacific Slope for work in tidal and magnetic observations. In 1854 he was made first lieutenant, which position he resigned in 1856 to become professor of mathematics in the University of Michigan, but remained here one year only to return to the Coast Survey as permanent scientific secretary to its chief. He remained here from 1857 to 1862. During this period his work involved the preparation for publication of the explorations made of the Gulf Stream, the set of permanent observations at Key West for the registrations of magnetic variations, and at the breaking out of the war, a preparation for the navy of minute descriptions of the southern harbors, inlets and rivers. He was also in charge of the survey made of Narragansett Bay to determine its suitability for a navy yard. The project was afterwards abandoned.

During the remainder of the war, a civilian under the War Department, he was in charge of the branch office in New York City, acting as agent in obtaining materials for use in the field, and in charge of the construction of the defences around New York Harbor. The principal work here were the forts at Willet's Point and on Governor's Island, and the repairs at Fort Schuyler. At the close of the war in 1865, he became vice-president of the Novelty Iron Works, under the late Horatio Allen, honorary member of this Society. He remained in this relation until 1870, and it was during this period that he became so well and favorably known as a mechanical engineer. It was during this period that those experiments were undertaken to determine the consumption of water per horse-power in engines at different points of cut-off, which have become historic as the Novelty Iron Works experiments. These were made by Chas. E. Emery, member of the A. S. M. E., and reported by him at the Scranton meeting of the Society, and recorded in Volume X. of the Transactions. In 1870, it was deemed advisable, by reason of the business changes incident upon the conditions which prevailed at the close of the war, to close out the Novelty Iron Works, and Prof. Trowbridge was then selected for the chair of Dynamic Engineering at Sheffield Scientific School, Yale College, where he remained until 1877, when he was summoned to the chair of Engineering at School of Mines at Columbia College, which position he was filling at the time of his death.

He presented several papers before the Society, which are recorded in Volumes III., VI., and VII., but his close confinement by his duties as head of the Engineering Department prevented his attendance at many of the meetings. He was vice-president of the Society in its early years, and had served upon its Finance Committee with great satisfaction at a time when the society passed through a crisis before its present success became assured.

He died August 12, from heart failure, at his home in New Haven, Conn.

He had been very active in the initial steps leading to the formation of the society, and was one of its charter members.

JOHN W. HANDREN.

John W. Handren was born June 25, 1832, at Quaco, New Brunswick, Canada. He served an apprenticeship of four years in St. John, N. B., with the firm of Williams & Robertson, commencing as blacksmith and passing through the machine-shop. His work here was on steamboat and other repair work. For two years he was in the Pembroke Rolling Mills, and afterwards for four years superintendent of a large repair shop and dry docks in Milwaukee. For two years he was in charge of repair work on the Cromwell line of steamers. At the breaking out of the war he was assigned to the U. S. S. *Huntsville* as acting assistant engineer during the years 1861 and 1862. Later he was put in charge of the furnace work connected with the building of the monitor *Dictator*.

In March, 1863, he entered the business, which he followed ever since, of repairing steamships and other marine engineering, operating dry docks at the Erie Basin under the firm name of Handren & Robins. The most notable work which passed through his hands during this period was the steam yacht *Sultana* and two barges for the American Steel Barge Co., beside the usual routine of repair. His death took place September 5, 1892, and was due to Bright's disease.

He connected himself with the Society as member at the Erie meeting, May, 1889.

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APPENDIX.

THE TRANSCONTINENTAL EXCURSION OF 1892 *OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.*

IT has been thought worth while to put on record some of the matters attaching to the journey across the continent undertaken to attend the San Francisco Convention of the American Society of Mechanical Engineers, not only because it was a thing most enjoyable in itself, but because it is a noteworthy event in the Society's history, in much the same way as the trip to Europe in 1889.

While the idea had doubtless been in the minds of several before that, the first impetus was given to the movement to meet in San Francisco by Mr. C. W. Hunt of New York, in conversation at the Providence meeting in June, 1891.

This was in advance of any invitation from the Pacific Slope, so that the question of the expediency of such a move could be considered by itself, and did receive exhaustive discussion at more than one meeting of the Council. The arguments for and against the plan were summarized in a brief sent to the members of the Council for consideration, and were as follows:

"To meet in San Francisco would be to give the Society a standing as a national organization, in advance of 1893, as no other choice would. Such a jaunt would have something of the same public influence as the European trip of 1889; its effect on the Pacific Coast would be most beneficial; there would be much of incidental pleasure; there is every reason to believe it would be a numerical success. Against the plan would be the question of the right or policy of the Council, as a representative body, to assign a meeting at a point where only a wealthy and leisured minority could attend it; where many usual participants in debate would be shut out; at a point remote from the center of the Society's population."

The good effect of the meeting upon the Pacific Coast itself, in stimulating a closer professional bond among the engineers, has been frequently referred to since the meeting, and many new members are to be expected in the near future. At this time, also, there were received letters of warm commendation of the plan of a San Francisco meeting from resident members in California, and the result of these circumstances was the issue of the following circular, accompanied by a postal card for reply:

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS,
No. 12 West Thirty-first Street, New York

Dear Sir:—Letters have been received from the Pacific Coast urging a proposition to hold the spring Convention of the Society in 1892 at San Francisco, California. The question of such a meeting has come before the Council, and has received their favorable consideration, and such invitation will be accepted if the membership at large should approve the selection.

The outlines of the plan are as follows:

(1) The pleasantest time for the transcontinental journey would be the early weeks of May.

(2) The plan in general (subject to modification) would be to travel by special Pullman train of sleepers and dining cars over the Pennsylvania lines from New York to Chicago, picking up members on the way; thence by the M. St. P. to Denver; D. R. G. to San Francisco, with stop at Pike's Peak, Manitou, and Salt Lake City. After the meeting, to journey north to Portland *via* the Yosemite, and return by the N. P. R. R. *via* the Yellowstone Park. The party would have its own exclusive cars over the entire trip, and be run on its own time-table.

(3) The time required would be about four weeks for the above trip, which, of course, can be extended if the party direct it.

(4) The cost for *the journey*, including railroad fares, sleepers, and dining car service, and transportation in cities traversed, will not exceed $300 each for the thirty days' absence, and may be reduced if a sufficient number accept. This does not include the hotel bill at San Francisco during the Convention, nor where any stops may be made in other cities.

The considerations which have influenced the Council to favorably entertain the movement are:

(1) To hold a Convention on the Pacific Slope is to emphasize the fact that the Society of Mechanical Engineers is a national organization, and there are also some advantages in accomplishing this before the Columbian Exposition of 1893.

(2) The Society has very few members west of the Mississippi; hence it will be of benefit to have its membership and position brought publicly before the communities which it visits, since new members are always added from the district where a meeting is held.

(3) San Francisco has much interest to offer to engineers, and the engineering of the Sierra railways is notable. The city is also prepared to make much of the visit of so representative a body.

(4) The journey, under the circumstances attending it as planned, will be most enjoyable to those who can participate in it, and, for the Society at large, would have much the same favorable result as the excursion to England in the summer of 1889, whose good effects are still manifest in many ways.

The Council only hesitate to accept an invitation involving so many attractions until they can be advised by the members whether enough would go to make such a Convention a representative and worthy one; and, secondly, whether the policy of such a successful meeting would be upheld by those busy members whose engagements would preclude their getting away for so long an absence. You are therefore requested to advise the Council by the inclosed card as soon as possible, that a decision may be promptly reached.

Very truly,

F. R. HUTTON, *Secretary.*

SEPTEMBER 14, 1891.

POSTAL CARD.

ERASE THE VIEWS YOU DO *not* WISH TO EXPRESS: *sign* THE CARD AND MAIL.

I. I cordially approve the plan of the San Francisco Convention; should the Council decide to hold it there I should hope to go, and my party would consist of ...... persons.

II. I approve of the policy of a San Francisco Convention, but cannot say whether I would go or not. Will advise later if the plan is decided upon.

III. I approve of the San Francisco plan, but could not attend the meeting.

IV. I do not approve of the plan of holding a Convention on the Pacific Slope. I would prefer that the meeting should be held in some such city as ____________

Name ____________

Address ____________

The postal-card replies grouped the members into four divisions, and at the November meeting of 1891, in New York, the opinion stood as follows:

I.	Approve of the plan and hope to go	161
II.	Approve the plan, but would say later if they would go	275
III.	Approve the plan, but cannot go	270
IV.	Do not approve of the plan and prefer some other city	74

It may not be without interest to note that the party was finally made up as follows:

	Postal Reply.	Took the Trip.
From Group I	161	39
" " II	275	23
" " III	270	1
" " IV	74	2
Others, guests and new members	—	10
		75

The matter of arranging details of transportation was left to a committee, consisting of the treasurer and the secretary of the Society, and it may not be amiss to mention how much of the success of the journey was due to the skillful energy which the chairman of this committee brought to the work of laying it out. It was planned at first to contract direct with the railway companies for the necessary cars and transportation, but this was found to be surrounded with difficulties, to say nothing of the burden on the committee. The party would have to be of just such a size as fully to fill the necessary cars, making the carload the unit; and the entire party must come home together, or any withdrawing would have to pay full price for their vacancy. On the other hand, by operating through a tourist agency making a specialty of work of this class, we could not only do as well or better in price, but could arrange for considerable latitude as to the return from the Pacific, and relieve the committee from all anxiety and detail.

Under the direction of the committee, Messrs. Raymond and Whitcomb prepared a series of itineraries, identical outward bound, but different on the return, which were grouped in a booklet form and sent to every member in January, 1892. The date chosen for the start was May 4, to strike the interval when it should be spring east of the mountains, but the dry season not yet begun on the western side. The time chosen proved to be most fortunate.

When the time of departure drew near, it was discouraging to find how many were compelled to forego their journey for good and sufficient reasons. Prominent among these reasons, besides sickness, was the call to take action upon matters relating to the securing of space for the exhibits to be made at Chicago in 1893, on the part of many interested and prominent manufacturers.

On assembling at the West Shore Railway terminus, it was found that only one train of seven cars was needed; four sleepers, baggage, dining, and smoking car. The dining car ("The Iturbide") was put in the middle, and the smoker ("Capitano") at the rear. In this smoker was the barber shop; and access could be had to the trunks in the baggage car in front as required.

The party was distributed as follows: *

* See also the rhymed list of the party in the sequel.

HEAD OF THE TRAIN.

Mr. & Mrs. S. T. Wellman.		Mrs. H. G. Hammett.
		Misses G. L. & F. S. Hammett.
Misses Trump.		Mr. & Mrs. C. N. Trump.
Mr. & Mrs. V. G. Hazard.	Car "SOPRIS."	Mr. Albert Stearns.
		Miss G. L. Stearns.
Mr. & Mrs. D. Ashworth.		Mr. W. S. Wellman.
Mr. & Mrs. Jas. McBride.		
		Mr. F. A. Scheffler.
Mr. & Mrs. E. H. Parks.		

Mr. & Mrs. Joel Sharp.		Mr. & Mrs. F. H. Daniels.
Mr. & Mrs. C. W. Hunt.		Prof. G. I. Alden.
Mr. & Mrs. M. P. Higgins.	Car	Mr. W. E. Schoenborn.
Mr. & Mrs. Geo. H. Smith.	"KEWANEE."	Mrs. M. E. Harrington.
Mr. & Mrs. H. S. Haskins.		Mr. Walter G. Cotton.
Mr. & Mrs. A. K. Mansfield.		Mr. John Knickerbacker.

DINING CAR, "THE ITURBIDE."

Mr. W. B. Cogswell.		
Miss Mabel Cogswell.		Mr. F. M. Power.
Dr. & Mrs. H. G. Torrey.		
Mrs. M. W. Bringhurst.		Mr. & Mrs. E. Andrews.
Mr. Wilfred Lewis.		
Mr. & Mrs. Alfred Betts.	Car	Mr. Fred. H. Laforge.
Mr. W. F. Monaghan, (usually in engine-cab.)	"CORINTHIA."	Mr. Gilbert S. Smith.
Mr. Ph. Roux.		Mr. Ferd. Martens.
Mr. F. J. Miller.		Mr. Chas. A. Cooke.
Mr. Thos. Hibbard.		Mr. D. S. Jacobus.
Mr. T. J. Borden.		Mrs. Winebrenner.
		Mr. & Mrs. D. G. Moore.

Mr. & Mrs. W. C. Williamson.		Mr. & Mrs. Jno. D. Williamson.
Mrs. R. W. Hunt.		Mr. R. W. Hunt.
Mr. Washington Jones.	Car	Mr. Jno. T. Boyd.
Mr. J. A. Rittenhouse.	"SYDENHAM."	Mr. Emerson W. Judd.
Mr. Chas. F. Chandler.		Mr. & Mrs. W. H. Wiley.
Misses S. K. and H. O. Wiley.		F. R. Hutton.

COMBINATION CAR "CAPITANO."

REAR OF THE TRAIN.

Messrs. Charles A. Cooke, Emerson W. Judd, and C. F. Chandler accompanied the party to conduct it safely, and Messrs. J. A. Rittenhouse and Carpenter were respectively in charge of the Pullman and dining-car service.

Leaving Weehawken, over the West Shore Railway, at 9.45, Wednesday, May 4, the party spent most of the first day in getting accustomed to their new surroundings. At Syracuse Prof. Sweet and Mr. E. N. Trump, and Mr. and Mrs. E. P. Bates, were at the station, the latter out of their regret at remaining behind, yet thinking of the pleasure of the favored ones, and bringing boxes of lovely flowers for the ladies and plants for our general car to bloom all the way to the coast. On Thursday morning, on the Chicago and Grand Trunk line, it was raining hard, and Michigan and Illinois were beginning to disappear under water; but it held up enough for the debarkation at Halstead Avenue, in Chicago, into carriages to convey the party to Jackson Park and the Exposition grounds. Here the visitors were welcomed by Mr. H. F. J. Porter, member of the Society, and assistant mechanical engineer to the Commission, and after examining the moving sidewalk model, were convened in the Woman's Building, before a desk and a large map of the grounds, where Mr. Porter spoke briefly in welcome and explanation, and was followed by Mr. R. Allison, in charge of the Department of Manufactures. The party then boarded a train of open gondola cars and were taken to the various buildings accessible in this way. Upon the completion of this hurried visit the carriages again received the party for a drive down the Drexel Boulevard, Michigan Avenue, and the Lake front, to the old works of Fraser & Chalmers. After a visit of interested inspection here, the railway station was reached, and the information authoritatively given that by reason of washouts on the Rock Island road the train would not leave Chicago before the next morning. Mr. Chalmers and his associates, hearing of this, insisted that the travelers should be their guests at a theater party, and not only furnished seats reserved, but carriage transportation to and from the railway station. The night was wet, but a large number went and enjoyed the outing, and the kind thought which prompted it.

That night was spent in the station yard, and the train did not start till nearly noon on Friday. It was a lovely day, but the train made slow progress over the newly made bed, and the track was in many places supported temporarily on blocking. After the train had passed, the road was again washed out, and no more trains passed for some time. But the floods in the Illinois river were very interesting to the travelers, and there was much photographic activity.*

The Mississippi was crossed in the late evening at Rock Island, Ill., and Davenport, Ia., and the Missouri river on Saturday morning, from Council Bluffs, Ia., to Omaha, Neb., where the Pacific Railway system began. The journey across Nebraska was made noticeable by the persistent derailment of the rear truck of the engine tender, and the loss of time thereby caused. Comment was made upon the fact, during the recurrent necessity for replacing the wheels, how carefully the members of the party refrained from annoying the crew with suggestions as to how to do it. Many men could have capably undertaken it, but instead they refrained from comment, and even from close crowding.

That evening, in the dining-car, was held a musicale, of which Mr. Louis Cassier was impressario. The programme was as follows, and it was surprising how satisfactory a thing of the sort can be under circumstances so apparently unfavorable :

GRAND ENTERTAINMENT.

SIXTY MILES AN HOUR.

SPECIAL PULLMAN TRAIN.

IN CAR CORINTHIA, EN ROUTE TO CALIFORNIA.

Saturday Evening, May 7, 1892, 8 P.M.

1. Realistic Story,	Señor Don Robt. W. Hunt.
2. A. S. M. E. Quartet,	Mme. Higgins, Mlle. Stearns, Mons. Hutton and Smith.
3. Pigs and Razors—A California Story of '49,	Señor F. M. Power.
4. Duet,	Señora Higgins, Señora Stearns.
5. Oration,	Mons. F. R. Hutton.
6. Solo—Mrs. Lofty and I.	Mme. H. G. Torrey.
7. Humorous Recitation.	Mons. D. Ashworth.
8. Guitar Solo—Spanish Dance,	Signor G. S. Smith.
9. Lecture—Little Specks of Gold,	Mons. Torrey.
10. Guitar Solo, with Song—Bangor,	Mlle. Stearns.
11. Lecture,	Prof. Wm. H. Wiley, D.D.
12. Lecture,	Dr. Schoenborn.

* He shall be nameless who called it kodak-tivity.

The energy of the same member of the party surprised the diners one day by a menu, whose sauces or procedure were named after members of the party, with a view to suggest some of their characteristics. The dining-car service, by the way, was most satisfactory all the way to Oakland.

On Sunday morning, twenty-four hours behind the schedule, the train reached Colorado Springs in a cold, raw air which had snow in it, and turned to rain later. Five miles farther the train stopped for the day at Manitou Springs. Carriages were waiting at the station and conveyed the majority of the party to the Garden of the Gods, the Grand Caverns, and the Springs. Returning to the train in the rain, the view of Pike's Peak was obscured for the day by cloud, and a glimpse only was had by a favored few toward nightfall. A few went to the baths, others by electric car to the town of Colorado Springs, and a few were able to bring back metallic mementos bearing the legend "Good for 12½ cents." The party was divided for supper at the Barker House and Cliff House, at Manitou, and reassembled in the evening at their train, which took them over the Denver and Rio Grande system through Pueblo to Cañon City, where they found themselves at breakfast time Monday morning, ready for the trip through the Royal Gorge of the Arkansas river. An exceedingly enjoyable feature of the trip was this adjustment of time-table, so that impressive scenery should be traversed by daylight. The train would be run into a siding to wait if the point of interest were reached too early. The train was stopped at the narrowest point where the hanging bridge is, to allow the party to alight and saturate themselves with the impressive grandeur of the scene. The beholder feels at once the fatuity and inadequacy of his adjectives as means of expression at such points as these. The rest of the morning was spent in progress to Salida, in full view of the magnificent Sangre de Cristo range of mountains, 14,000 feet high, snow-capped, sharply peaked and rugged against a background of sky of the most brilliant blue.

After luncheon, a trip by narrow gauge up to the top of Marshall Pass filled the afternoon. The summit is 10,852 feet above the sea, and the elevation was trying to a number of the party. Coming down, three adventurous spirits obtained permission from the superintendent to ride on the engine pilot, and only met the to-be-expected heads of cattle when far enough down the descent for the train to be kept under control. The grades in the pass are 211 feet to the mile in places, and the curves are as high as 24 degrees.

That evening, while waiting for the start after midnight, some indefatigables went to the concert in town, given by the Glee Club of the University of Colorado, from Boulder City. It occurred to two of the professors in the party to exchange with the performers an inter-collegiate type of courtesies, which resulted in a celebration in the combination car in the station yard, and a serenade to the ladies in the rear car after eleven. The boys won golden opinions from many.

Passing Leadville by night and the Eagle River Cañon, the train entered Grand River Cañon in the morning, reaching Glenwood Springs in time for a bath in the warm saline waters of the great pool of the sanitarium. This novel experience was much enjoyed after the previous continuous railroading, and but few of the party remained upon the grassy banks. Some of the ladies seemed specially to enter into the spirit of the frolic about the cold fountain.

Leaving Glenwood after luncheon, the train crossed between mountains and buttes to the junction with the Rio Grande Western, and passing through Castle Cañon and the Red Cañon (the latter in early evening, with the moon set in deep blue behind the red of the cliffs), the summit of Soldier Pass was reached at 7,465 feet of elevation, and the descent began to Salt Lake City, down the side of the Wahsatch Mountains. At the station Mr. Wm. J. Silver, member of the Society, met the party, and with his son, Frank, acted as escort in a visit by electric car to Temple and Tabernacle, to the co-operative store and other points of interest. A few were invited to meet Presidents Woodruff, Smith, and Cannon, at the offices of the corporation—an opportunity highly appreciated. After lunch at the train, an excursion was arranged to Garfield Beach on Salt Lake, but it was too early in the season for bathing. Before the train started a large box was left there, its wealth of flowers to be distributed among the ladies. This came from Fraser and Chalmers, at the instigation of Mr. Eckart.

Leaving Salt Lake City in the evening, Ogden was the next point, where another hour disappeared, and in the morning of Thursday the train was crossing Nevada plains. Here the Piute Indian appeared for the interest of all, and of the photographers in particular. The experience at Carlin, Nev., of one

Safely into Chicago we came,
A city heretofore known only by name,
But hereafter all that savors of fame
Between Delaware and "Frisco."

Glad are we all they have the Fair;
New Yorkers, surely, ought not to care;
For the West should certainly have its share
Of good things nearer "Frisco."

Our train was in charge of a Cooke of renown,
Whose handsome face scarce knew a frown;
And the fault be ours if we're not cooked brown
Ere we get way over to "Frisco."

Without wishing to speak in a thrasonical manner,
I would say *our* State should *carry the banner*,
For no other State has sent more *than her*
To the convention out at "Frisco."

Our Trumps are high, we Hazard all,
Which Bring(h)u(r)s(t) luck, though our Betts are small;
We pleasantly play Andrew or "call,"
While on our way to "Frisco."

As mechanics have their Cogswell to order,
We have with us a father and daughter;
A girl so sweet we are all glad he brought her
With him on his way to "Frisco."

We have Mrs. Winebrenner, a widow gay,
Who thinks life but a summer day,
And says all who want to be happy may,
Whether at home or at "Frisco."

Mr. C. W. Hunt, with his genial face,
Surely a line of this rhyme should grace,
For he certainly fills an important place
In our train, on its way to "Frisco."

His Hunt for a wife has surely been blest,
For the *dear* he has caught came not from the West;
Their married life is happy and blest,
We found out on our way to "Frisco."

There is Mr. Powers with camera in hand,
Who gathers pictures from every land;
We hope some may serve to recall the band
He journeyed with to "Frisco."

We have Cotton, which is neither spool nor batt,
But wears a mustache under his hat,
And seems to feel very happy that
He is on his way to "Frisco."

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