PRESIDENT

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS

TRANSACTIONS

VOLUME 41

DETROIT MEETING
NEW YORK MEETING
1919

NEW YORK
PUBLISHED BY THE SOCIETY
29 West 39th Street
1920

CONTENTS OF VOLUME 41[1]

DETROIT AND NEW YORK MEETINGS

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

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

To the Members:

THE accompanying forty-first volume of Transactions records the activities of the Society during the past year and includes a number of papers on industrial problems which have been brought to the fore by the unsettled conditions following the great war.

The volume comprises forty-six papers, addresses and discussions on engineering, industrial and economic subjects, three reports to the Society — one being the report of the Council, another the Manual on American Standard Pipe Threads and the third the report of the Joint Conference Committee of the four Founder Societies.

One group of papers deals with the subject of research in private, industrial and educational institutions, while another is concerned with personnel relations. Two papers, with a voluminous discussion, consider the burning of powdered-fuel and give the results of tests of powdered-fuel plants. Other papers deal with power plants, aeronautics, measuring devices, the mechanism of fractures, internal-combustion engines, machine shop practice, drainage pumps and locomotives.

The preparation of the volume has been carefully supervised by the Publication Committee and special pains have been taken in the selection, arrangement and indexing of the material to make it of the greatest possible reference value.

Calvin W. Rice, Secretary
29 West 39th Street
New York

September 1920.

OFFICERS
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

FORMING THE STATUTORY COUNCIL

1919

PRESIDENT

Mortimer E. Cooley.................................Ann Arbor, Mich.

VICE-PRESIDENTS

Terms expire December 1919

John Hunter.................................New York, N. Y.
Spencer Miller.................................New York, N. Y.
Max Toltz.................................St. Paul, Minn.

Terms expire December 1920

Fred R. Low.................................New York, N. Y.
Henry B. Sargent.................................New Haven, Conn.
John A. Stevens.................................Lowell, Mass.

MANAGERS

Terms expire December 1919

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

Terms expire December 1920

Fred N. Bushnell.................................Boston, Mass.
Fred A. Geier.................................Cincinnati, Ohio
D. Robert Yarnall.................................Philadelphia, Pa.

Terms expire December 1921

Charles L. Newcomb.................................Holyoke, Mass.
Charles Russ Richards.................................Urbana, Ill.
Frank O. Wells.................................Greenfield, Mass.

PAST-PRESIDENTS

Members of the Council for 1919

James Hartness.................................Springfield, Vt.
John A. Brashear.................................Pittsburgh, Pa.
D. S. Jacobus.................................New York, N. Y.
Ira N. Hollis.................................Worcester, Mass.
Charles T. Main.................................Boston, Mass.

CHAIRMAN OF FINANCE COMMITTEE

W. E. Symons.................................New York, N. Y.

TREASURER

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

SECRETARY

CALVIN W. RICE.......................................New York, N. Y.

EXECUTIVE COMMITTEE OF THE COUNCIL

MORTIMER E. COOLEY, *Chairman*
IRA N. HOLLIS
JOHN HUNTER
D. S. JACOBUS
CHARLES T. MAIN
HENRY B. SARGENT

STANDING COMMITTEES OF ADMINISTRATION

FINANCE

Supervises all financial affairs, makes all contracts, orders all expenditures, administers the general offices

W. E. SYMONS (3), *Chairman and Representative on Council*
ALFRED E. FORSTALL (1)
GEORGE M. FORREST (2)
F. E. LAW (4)
ALEX DOW (5)

MEETINGS AND PROGRAM [1]

Conducts Annual and Semi-Annual Meetings, arranging professional and entertainment programs

D. S. KIMBALL (2), *Chairman and Representative on Council*
WALTER RAUTENSTRAUCH (1)
A. L. DE LEEUW (3)
W. G. STARKWEATHER (4)
R. V. WRIGHT (5)

Assisted by sub-committees which examine papers for special sessions

PUBLICATION AND PAPERS

Supervises publication of The Journal (monthly), Transactions (annually), Year Book (annually), Condensed Catalogues (annually), and other publications

GEORGE A. ORROK (5), *Chairman and Representative on Council*
JOSEPH W. ROE (1)
H. H. ESSELSTYN (2)
GEORGE J. FORAN (3)
RALPH E. FLANDERS (4)

MEMBERSHIP

Receives and scrutinizes all applications for admission or transfer, and reports names to Council

S. D. COLLETT (1), *Chairman and Representative on Council*
FREDERICK A. WALDRON (2)
R. F. JACOBUS (3)
W. S. TIMMIS (4)
HOSEA WEBSTER (5)

CONSTITUTION AND BY-LAWS

Reports on all matters relating to Constitution, By-Laws and Rules referred to it by Council

JESSE M. SMITH (3), *Chairman and Representative on Council*
GEORGE M. BASFORD (1)
IRA H. WOOLSON (2)
E. S. CARMAN (4)
JAMES E. SAGUE (5)

[1] Sub-Committees are published in 1919 Year Book.

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

LOCAL SECTIONS

Supervises conduct of affairs of all Local Sections throughout the country

D. Robert Yarnall (1), *Chairman and Representative on Council*
W. H. Kenerson (2)
C. Russ Richards (3)
Louis C. Marburg (4)
S. B. Ely (5)

STANDING COMMITTEES

HOUSE

Maintenance of rooms of Society, furnishings, paintings, memorabilia, etc.

Orrie P. Cummings (1), *Chairman*
Maxwell P. Upson (2)
H. O. Pond (3)
J. D. Maguire (4)
F. A. Muschenheim (5)

LIBRARY

Represents Society on Library Board of United Engineering Society

Walter M. McFarland (1)
A. M. Hunt (2)
W. N. Best (3)
Jesse M. Smith (4)
The Secretary

PUBLIC RELATIONS

Considers matters referred by Council bearing on relations of engineer to community

F. H. Newell (1)
Fred R. Low (2)
James E. Sague (3)
J. Waldo Smith (4)

RESEARCH [1]

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

Arthur M. Greene, Jr. (1), *Chairman*
Carl C. Thomas (2)
Albert Kingsbury (3)
Walter Rautenstrauch (4)
Reginald J. S. Pigott (5)

STANDARDIZATION COMMITTEE

Standardizes method of making and arriving at standards themselves
Endeavors to bring about a unification of the standards of the Society

Henry Hess (3), *Chairman*
Lionel S. Marks (1)
Clarence F. Hirshfeld (4)
Walter S. Twining (5)
William F. Kiesel (2)

SOCIETY REPRESENTATIVES [2]

Appointees by the Council on committees, etc., of other bodies

[1] Sub-Committees are published in 1919 Year Book.
[2] Partial List of Society Representatives. Complete list published in 1919 Year Book; also special committees and Local Sections' Committees.
Note:—Numbers in parentheses indicate the number of years the member has yet to serve

TRANSACTIONS

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOLUME 41 — 1919

THIS volume contains an account of the activities of the Society for 1919, and in it will be found papers and addresses given at the Spring Meeting, held in Detroit, and the Annual Meeting, held in New York, with the discussions thereon.

In selecting the material published, the intention has been to include all papers having permanent reference value. For this reason condensed accounts are given of much discussion on papers covering the less technical subjects and reference is made to issues of MECHANICAL ENGINEERING in which the subject matter may be found in a more complete form.

MORTIMER ELWYN COOLEY

MORTIMER COOLEY, born March 28, 1855, was the fifth member of a family of eight children, all reared on a farm in Canandaigua Township, Ontario County, N.Y. He is of the ninth generation of Cooleys in this country. Benjamin, the first of the name, came from England and settled in Springfield, Mass., in 1642, where for many years he was a selectman. Benjamin was an ensign in the Hampshire Regiment commanded by Major John Pynchon in the King Philip War. He was a weaver by trade and lived not far from Mill River. The Barney and Berry skate factory, overlooking the Connecticut River, is located on the rear end of the old Cooley homestead. Later, a dozen or more Cooleys owned homes in Longmeadow, four or five miles south of Springfield on the other side of Mill River. As the tribe multiplied some of them moved to Granville, Mass., a beautiful and picturesque locality at the south end of the Green of naming her with appropriate naval ceremonies. She was duly christened Alliance in the ship's honor and if the tales of rear admirals of today who were cadets in those days are to be believed the occasion was a notable one in the annals of the Navy.

Cadet Engineer Cooley was forthwith detached and ordered home for a few weeks, then to the Bureau of Steam Engineering at the Navy Department. In June 1881 he was examined and promoted to Assistant Engineer and in August was ordered to the University of Michigan to teach steam engineering and iron shipbuilding. At the end of three years on request of the Regents his detail was continued a fourth year. Being then detached and ordered to the Pacific Station, the Regents conferred on Assistant Engineer Cooley the honorary degree of Mechanical Engineer and invited him to resign and accept the chair of Mechanical Engineering. This he did, his resignation taking effect December 31, 1885. It was with a great deal of regret that he resigned as he was in love with the Service. There was at the time no prospect for any great increase in the naval force, and it seemed to him the opportunity for real work afforded him at the University ought not to be declined.

Professor Cooley has given his entire life since he was twenty-six years of age to university work — thirty-eight years up to now. He has been Dean for fifteen years, having been appointed in February 1904. The Michigan Agricultural College conferred on him the degree of LL.D. in 1907, and the University of Nebraska the degree of Eng.D. in 1911. When he came to the University there were but sixty or seventy engineering students out of a total of about 1300 in the University, and the entire technical work in engineering was done in seven rooms at the south end of the main university building. The first engineering laboratory was built the winter after he came. It was a two-story brick veneer building 24 x 36 ft. costing $1500 and the equipment $1000. In it Professor Cooley himself taught forging, pattern making, and machine shop practice. It was styled by his colleagues "the scientific blacksmith shop." It was the beginning of an effort, now altogether general, to give to engineering students while in college some practical knowledge of the materials and processes used in the execution of engineering projects.

But Professor Cooley could not wean himself altogether from the naval service. He was from 1895 to 1911 the Chief Engineer officer of the Michigan State Naval Brigade and is now a retired officer in the Brigade. In 1898 he returned to the Navy as Chief Engineer during the Spanish War. He was attached to the U.S.S. *Yosemite* and later to the League Island Navy Yard, his period of service being altogether about ten months. His honorable discharge was handed him by the commandant of the navy yard with words of commendation for his efficient work.

While on blockade duty off San Juan, P.R., the *Yosemite* engaged in a five-hour battle with the Spanish forts, gunboats and torpedo boats following the interception of the *Antonio Lopez*, a Spanish cruiser loaded with munitions, putting into the harbor. During the blockade a serious fire broke out in the coal bunkers of the *Yosemite* which for a time threatened serious consequences. The fire was deep down and could not be reached. Chief Engineer Cooley, recalling the method of sinking piles on western rivers by means of a water pipe attached to the pile, had a hose and nozzle triced to a long slice bar with which, under fire pressure from the pumps, the fires were successfully quenched. The slice bar could be shoved down into the coal like a knife into soft butter.

Following his return to the University in 1899 Professor Cooley was invited by the Citizens' Committee of Detroit, of which Governor Pingree was Chairman, to appraise the power plants, rolling stock and stores and supplies of the Detroit street railways, which the city was contemplating purchasing. It was a hurry job and was done in a hurry. The appointment was made on Friday, the staff was organized Saturday and the report submitted the following Saturday covering $2,000,000 of property. The following year, 1900, at the request of Governor Pingree, the Board of State Tax Commissioners, and the Board of State Auditors, Professor Cooley undertook to appraise the specific tax-paying properties of the State of Michigan which included the steam railroads, the telegraphs, the telephones, the plank roads and the river improvements. This was late in August. The field work was completed in ninety days and the results submitted at the end of December in time for the incoming legislature. The work involved the inspection of 10,000 miles of track, thirty-odd thousands of the freight cars, all the passenger and special equipments, all the locomotives, telegraph, and telephone lines, in short everything involved in the different kinds of properties. Some 150 men were employed. The total of the appraisal was about $240,000,000 and the cost less than $75,000.

As a result of this work the legislature enacted laws placing the railroads on an ad valorem tax basis which increased their taxes threefold and more. When the assessment was made under the new law in 1903 the railroads brought suit to enjoin their collection of taxes. This made necessary another appraisal as of the date of the assessment in which the value found for the railroads was $240,000,000 an increase of $40,000,000 due largely to using 1903 prices for labor and materials instead of the average from 1890 to 1900. The case was carried to the U.S. Supreme Court and being finally decided in favor of the state, brought into the state treasury twelve or fifteen millions in back taxes.

Michigan's pioneer work in valuation of large public utility properties was soon followed by other states. First among them was Wisconsin in a valuation of her steam railroads. Substantially the same methods were employed as in Michigan. Professor Cooley was consulting engineer.

In the twenty years which have elapsed since that first appraisal in Detroit, Professor Cooley has had charge of many hundreds of appraisals in various states and municipalities, in most of them employed by the public. In all of them he has stood consistently for correct results regardless of employer, "hewing to the line letting the chips fall where they may." In the aggregate the value of property appraised under his direction lies somewhere between one and one-quarter and one and one-half billion dollars.

Nor has Professor Cooley neglected opportunities to serve in other capacities. He was for a time chairman of the Board of Fire Commissioners, and President of the Common Council in Ann Arbor in 1890-1891. He served on the Board of Awards for the World's Fair in Chicago in 1893, and for the Pan-American Exposition in Buffalo. He has for twenty-five years served as mechanical expert in patent causes, and testified many times on mechanical matters before juries and commissions. He was for five years (1907–12) chairman of the Block Signal and Train Control Board of the Interstate Commerce Commission.

Professor Cooley is a Fellow of the American Association for the Advancement of Science, member, since 1884, of The American Society of Mechanical Engineers, American Society of Civil Engineers, American Institute of Consulting Engineers, Franklin Institute, Society for the Promotion of Engineering Education, Society of Naval Engineers, Michigan Engineering Society, Detroit Engineering Society, Sigma Phi, Tau Beta Pi, Sigma Xi, the Army and Navy Clubs in Washington and in New York, the Detroit Club and the Yondotega Club in Detroit.

Professor and Mrs. Cooley have four children, three daughters and one son. All are married and seven grand children now keep them from growing too old. The son is a Commander in the United States Navy.

Thus has Mortimer Cooley established his record of unusual accomplishment and honest devotion to his democratic ideals. And he is still "doing the chores."

ANNUAL REPORT OF THE COUNCIL

In his annual address[1] President Cooley has reviewed the organization of the Council and Committees and much of the work which has passed through these channels during the year 1919. This annual report of the Council therefore supplements the President's address by giving the more statistical record of actions taken by the Council as the body administrating the Society's affairs.

COMMITTEES

Under a revision of the Constitution C45, which was put into effect this year, changes in the grouping of the standing and special committees of the Society have been made and further changes are pending, all leading towards an ultimate grouping of all committees into administrative, professional and non-professional, with each group having two classes, standing and special.

The work of adjusting the membership of the committees during the year, to have as equable as possible distribution of their membership among qualified men all over the country, has been a matter of serious consideration by the President and has finally led to the appointment of a special committee of the Council, the Committee on Committees, to report on all committee activities. Continuous membership on certain types of committees will be discouraged in the future, but a sufficient number of experienced and active members of such committees will be continued to secure continuity of policy. It is believed that the adoption of these policies will greatly stimulate the committee work.

COMMITTEES — NUMBER AND SUBJECTS

A review of the pages at the beginning of the volume will show the large number of committees having in charge the many activities of the Society.

[1] Transactions, 1919, No. 1710.

COMMITTEES OF ADMINISTRATION AND STANDING COMMITTEES

These committees have made annual reports, as required by the Constitution, which in complete form are on file in the records of the Society and are summarized in the following:

Finance. The year 1919 has been a particularly difficult year in the financing of the Society's activities, due to increased costs of all work. The expenditures of the Society per member for the fiscal year October 1, 1918 to October 1, 1919 were $32.43, as against $25.65 the preceding year. These expenditures were covered by current income.

Invested Funds and Trust Funds. The present standing of the invested funds of the Society is shown in the following table:

BALANCE SHEET

As of September 30, 1919

Assets

Society's one quarter interest in Engineering Society's Building, Land and Real Estate Equipment (25 to 33 West 39 Street)		$486,792.79
Library Books	$13,000.00	
Furniture and Fixtures	5,000.00	
		18,000.00
Stores, including plates and finished publications		29,307.11
Engineering Index		10,000.00
Trust Fund Investment:		
New York City 3½%, 1954 (par $45,000.00)	$39,696.81	
St. L., Peoria, & N. W. 1st 5%, 1948 (par $10,000)	10,613.89	
United New Jersey Canal Co., (par $1,000)	970.00	
Cash in Banks representing Trust Funds	7,518.17	
		58,798.87
Liquid Assets:		
Liberty Bonds		55,000.00
United Engineering Society		10,000.00
Accounts Receivable:		
Members Dues	15,008.10	
Initiation Fees	14,712.50	
Sales of Publications, Advertising, etc	58,112.53	
		87,833.13
Advance Payments		4,573.98
Cash: In banks for general purposes	5,535.87	
Petty Cash Fund	1,500.00	
		7,035.87
		$767,341.75

Liabilities

Trust Funds:		
Life Membership Fund	46,102.81	
Library Development Fund	4,902.71	
Week's Legacy Fund	1,957.00	
Melville Fund	1,127.36	
Hunt Memorial Fund	208.99	
Juniors' and Students' Prize Fund	2,000.00	
C. T. Main Award Fund	2,500.00	
		58,798.87
Dues paid in advance		2,241.06
Initiation Fees uncollected		14,712.50
Replacement Fund		1,163.18
Accounts Payable		5,244.12
Unappropriated Revenue	30,392.46	
Unexpended Appropriations-Excess	6,527.48	
		23,864.98
Capital Investment	$514,792.79	
Surplus and Reserve	146,524.25	
		$661,317.04
		$767,341.75

Meetings and Program. Under the Spring and Annual Meetings of the Society is recorded the result of the work of the Meetings and Program Committee, but between the lines must be apparent the tremendous amount of preparation required to complete the programs of these meetings.

Notable sessions of the year were those on Industrial Relations, and Research. The committee's annual report points out that the committee has long been under pressure from progressive members of the Society to broaden the scope of the general meetings and papers presented.

Standing sub-committees of the Meetings and Program Committee are being gradually reorganized into Special Committees and now into Professional Sections reporting to the Council.

Publication and Papers. This committee has had a busy year in plans for the enlargement of the monthly publication, MECHANICAL ENGINEERING, and progress has been made in the face of many discouragements in adverse conditions in the printing trade, and increased costs of publication and materials.

Under this committee's jurisdiction, there have also been issued the annual TRANSACTIONS, now in its fortieth volume, the CONDENSED CATALOGUES, ninth annual edition, THE ENGINEERING INDEX and the YEAR BOOK.

CONDENSED CATALOGUES increased nearly fifty per cent in size over the preceding year, indicating progressive recognition of its usefulness.

The Engineering Index Annual, taken over from *Industrial Management* in 1919, has been practically doubled in the number of pages, and also in the number of copies issued.

The Year Book has been published in abbreviated form due to conditions in the printing trade, but this coming year it is hoped to return the book to its former scope.

Membership. The Society's membership has increased during the past year by 2182, and is now 11,882. The report of the Membership Committee includes recommendations for 11 reinstatements, and 63 promotions to higher grade. The committee has held twenty-nine meetings and had under consideration nearly three thousand applications.

The full table of activities for the year shows the following:

Recommended for membership	2182	
Deferred indefinitely	52	
Deferred	33	
Denied promotion	22	
In course of procedure	635	
Reinstatements	11	
Reconsiderations granted higher grades	63	
		2998

Honorary Membership has been conferred by the Council upon Charles Alphonse Clement de la Poix de Freminville, of Paris, France, for services rendered his country and the engineering world in the practical development of transportation, and Auguste C. Rateau, also of Paris, France, pioneer investigator in the field of the steam turbine and turbo-compressor.

In Memoriam. The report of a special committee on War Service and Members' Memorial, Major Fred J. Miller, chairman, was received at the Annual Meeting, and the Council records the resolutions taken by rising vote at this meeting in memory of those members who made the supreme sacrifice in the Great European War.

> "Resolved: That the Society hereby expresses its greatest appreciation and pride in the service of its members who gave their lives that freedom might be preserved among the nations of the earth. . . ."

COMMITTEE OF ADMINISTRATION AND STANDING COMMITTEES

Local Sections. The Local Sections of the Society now number thirty-six. In the following cities where local section meetings are regularly held, coöperation with existing local engineering organizations has been developed: Atlanta, Baltimore, Birmingham, Boston, Bridgeport, Buffalo, Chicago, Cincinnati, Cleveland, Denver, Detroit, Erie, Hartford, Houston, Indianapolis, Los Angeles, Meriden, Milwaukee, Minneapolis, New Haven, New Orleans, New York, Philadelphia, Portland (Oregon), Richmond, Rochester, Saint Louis, Saint Paul, San Francisco, Schenectady, Toronto, Troy, Tulsa (Okla.) Washington, Waterbury.

The Secretary made three trips to the local sections during the year, and visited practically every section. These visits, together with the visits of the Committee on Local Sections, resulted in the formation of new sections at Cleveland, Colorado, Eastern New York, Houston, Tulsa (Mid-Continent Section) Rochester and Washington, D. C.

In the Cleveland Section there has been installed a plan of joint membership in our Society and in the Cleveland Engineering Society.

The Local Sections Committee also hopes to carry out into future meetings its coöperation in the professional programs of meetings under the jurisdiction of the Meetings and Program Committee. This was done at Indianapolis this year.

Increase of Membership Committee work has this year been placed in the hands of the Local Sections Committee. Great benefit is derived from the local information concerning applicants for membership. The inclusion of increase of membership work is expected to be especially successful.

Constitution and By-Laws. This committee acts as an advisory committee, reporting on all matters referred to it by the Council relating to the Constitution, and By-Laws and Rules, and harmonizes with the Constitution any new policies of the Society.

Suggested changes in the rules are being drafted by this committee as the result of the recommendations of the special committee on Aims and Organization.

During the past year a notable change in policy has been effected in providing that the "Nominating Committee shall be elected annually by the voting membership of the Society." By-laws are now in effect which detail the manner of election.

Library. This Committee acts as the representative of the Society on the Joint Library Board through which the five libraries of the Founder Societies and United Engineering Society are administered as one.

Increased use has been made of the Library Service Bureau.

Recataloguing the joint libraries has been started; this is much needed in order to make more quickly available the great source of information in these individual collections.

House. The House Committee has given much time to the rearrangement and redecorating of the rooms of the Society. Part of the necessity for this was for suitable installation of the Memorial tablet to Frederick Remsen Hutton, prepared under the direction of a special committee, Past-Presidents Ambrose Swasey and Jesse M. Smith and Wm. H. Wiley, *Treasurer.* The foyer hall has a small committee room as a result of the remodeling and a more harmonious scheme of arrangement and decoration has resulted.

Research. This year the Research Committee, led by Arthur M. Greene, Jr., as chairman, has been able to carry out some of the ambitions which the committee has had for several years.

A reading of the complete report of this committee brings out many interesting features of the work.

Sub-Committee Reports have been presented in the following fields — Bearing Metals, Lubrication and Fluid Meters. A Heat Transfer Committee has been organized and started work which will take several years. The Research Committee had charge of one session of the Spring Meeting in Detroit.

On one of the trips of the Secretary to Local Sections, the chairman of the committee accompanied the Secretary and as a result sectional research committees have been established in several centers. Manufacturers having research laboratories have expressed their cordial interest and desire to coöperate with the committee; also educational institutions having engineering experiment stations.

Through the Research Section of Mechanical Engineering the Research Committee has contributed data covering, research results, research progress and problems, research equipment, personnel and bibliographies. The Council, when asked to appoint representatives on the Engineering Division of the National Research Council, appointed men from the Research Committee of the Society in order that the work of this committee might tie in with the work of the national organization under the United States Government. Professor Greene, *Chairman*, was this year's appointee.

Standardization. This Committee has been most active, in coöperation with the American Engineering Standards Committee, which latter has within its organization a membership representative of the Civil, Mining, Electrical, Mechanical Engineers and A. S. T. M. The work of this joint organization is reported later.

PROFESSIONAL AND SPECIAL COMMITTEES

A scrutiny of the list of committees will show the great number of professional and special committees and the vast field they cover. All have contributed splendid work in their various lines. Since, however, the purpose of this report is to outline only final and completed results, all the committees may not receive special mention. These committees included some war measure committees and with the coming of peace have been "mustered out," or in some instances their research work continued through other channels.

Professional Committees. Most important in the life of a professional society is the work of its technical committees. The Mechanical Engineers may be well proud of the splendid work being done by its professional committees at great expense of time and personal sacrifice on the part of the many members, each chosen to membership in a committee because of his special experience in the field covered. The completed work of these professional committees is permanently recorded in TRANSACTIONS and becomes in many cases Standard Practice. Notable progress has been made by the Boiler Code Committee and the Power Test Codes Committee, the latter with its nineteen individual committees.

Some further mention is made of these committees in the Joint Activities under Standardization.

Aims and Organization Committee. This committee appointed in 1918 has completed its report which has been separately printed and widely discussed through all the committees, local centers and at two of the general meetings of the Society. Similar committees of the other Founder Societies have, through specially designated representatives of their main committees, combined in a Joint Conference Committee, to secure the benefit of "united action of the engineering and allied technical professions in matters of common interest to them."

Drawing their conclusions regarding the final action from the two general discussions by the members of the recommendations of the report and from the sympathetic attitude of the Council regarding the recommendations, the Standing Committees of the Society have not been slow to sense the spirit behind this movement and have had the policies recommended under discussion for many months with the result that the final action turning the recommendations over to them, finds them ready to proceed.

Awards and Relations with Colleges. In his address the President has covered quite completely a review of the plans of the Council to inaugurate closer and more helpful policies with the student-engineer and the young engineer as represented in the Junior and Student Member; President Cooley also covered quite thoroughly the question of awards and prizes for contributions to mechanical engineering, either of a practical nature or in literature. The Charles T. Main Award was established this year, from a fund of $2500 partly contributed by Past-President Main.

SPRING AND ANNUAL MEETINGS

In the conduct of the two general meetings this year the Meetings and Program Committee has been assisted by the Committee on Local Sections and the local committees and by the Research Committee.

The Spring Meeting was held in Detroit, and a full account of the meeting has been published in the July issue of MECHANICAL ENGINEERING. The keynote session on Industrial Relations was so successful that the committee was asked to continue this session at the Annual Meeting. Another strong session of the Spring Meeting was one on Research, under the auspices of the Research Committee.

The Annual Meeting showed the best attendance of any meeting of the Society, the total number registered being 2116. There were sessions on Appraisal and Valuation, Gas Power, Industrial Relations, Machine Design, Power Machinery, Textiles, and Machine Shop. The first of these was a joint session with the American Society of Refrigerating Engineers, and was followed by a request to have a session on this subject at the Spring Meeting in St. Louis.

REPRESENTATION

Honorary Vice-Presidents were appointed to represent the Society at various functions throughout the year, such as conferring Honorary Degrees on M. Eugene Schneider by the Stevens Institute of Technology, the Annual Meeting of the Engineering Institute of Canada, the National Rivers and Harbors Congress, the Society for the Promotion of Engineering Education, the Promotion of Vocational Education, and the James Watt Centenary in Bingham, England.

PROFESSIONAL SECTIONS

An amendment to the Constitution is before the Society to create a Standing Committee on Professional Sections. Pending action on this amendment the Council has approved the policy of inaugurating professional sections and has authorized a special committee to report on various aspects of such an organization, — as local work, finances, meetings, etc. The new policy is in line with recommendations of the Committee on Aims and Organization and is intended to provide for those groups of men in the Society who are interested in a particular subject, and so sub-divide the membership professionally in the same way that the Local Sections groups provide a geopraphical sub-division.

STUDENT BRANCHES

There are 49 Student Branches. One of the features of the Annual Meeting on December 3, 1919 was the holding of a student branch conference and Dr. Hollis, Chairman of the Committee on Relations with Colleges presided. The plans for the Student Branches have been fully approved by the Council as outlined in the report of the Committee on Awards and Relations with Colleges submitted at the Spring Meeting in Detroit, and published in the October issue of MECHANICAL ENGINEERING.

JOINT ACTIVITIES

Certain joint activities of the Founder Societies, — Civil, Mining, Mechanical, Electrical engineering societies — are combined under United Engineering Society, the holding corporation for the property of the Engineering Societies Building and the administrative organization for the funds of Engineering Foundation, Engineering Societies Library, and Engineering Council.

Engineering Foundation. The Engineering Foundation, founded in November 1914 by a gift from Mr. Ambrose Swasey, Past-President and Honorary Member of this Society, for the furtherance of research in sciences and engineering or for the advancement in other manner of the profession of engineering and the good of mankind, has found its principal activity in coöperation with the National Research Council. In October, Engineering Foundation issued a progress report and copies of this pamphlet are on file in the Secretary's office.

Research Council. During the past year the National Research Council has been reorganized on a "peace basis." The American Society of Mechanical Engineers, three representatives on the Engi-

The American Engineering Standards Committee was formed in 1918 with a desire to bring into existence an organization similar to the British Engineering Standards Association and coöperate in the organization and control of engineering and industrial standards in the United States.

Anglo-American Standardization has been brought into active thought of the day, the Secretary of the British Association visiting the United States, and in one of the statements covering his mission said "It seems to me with so much talk of fierce trade competition between America and England, everything ought to be done to turn that harmful idea into Anglo-American coöperation rather than trade rivalry. The world is surely large enough for the great Anglo-Saxon producing countries to find ample scope for their products without in any way endangering their bonds of friendship, drawn so closely by their great defense of Right."

Permanent Franco-American Engineering Committee. It will be recalled that Charles T. Main as President in 1918 formed one of a delegation of representatives from the Founder Societies to the French Engineering Congress in Paris. Out of this has grown what is known as the Permanent Franco-American Engineering Committee, which is a joint committee with the other Founder Societies to assist the Government of France and all organizations of France desiring information.

No. 1687

MEETINGS JANUARY-JUNE

MEETINGS OF SECTIONS

NINETY-FOUR meetings were held by the twenty-four organized Sections of the Society and the Providence Engineering Society, an affiliated body, during the first six months of the year. Seventeen of these meetings were arranged jointly with one or more local technical organizations or branches of other engineering societies. A number of the papers of more general interents were published in MECHANICAL ENGINEERING during 1919.

ATLANTA

February 1: A dinner in honor of Secretary Calvin W. Rice at the Druid Hills Golf Club. In the afternoon, Secretary Rice addressed the students at George School of Technology.

March 21: Joint meeting with Birmington and New Orleans Sections. In the morning an inspection trip to the Fulton Bag and Cotton Mills. In the evening an address by N. C. Harrison on Powdered Fuel.

April 24: Address on Materials for High-Pressure Steam-Pipe Work, Wm. J. Neville.

BALTIMORE

January 27: Address by Secretary Calvin W. Rice.

April 29: Papers on Oil Engines, by Leon Wygodsky, and on Boiler Explosions, by R. E. Munro.

BIRMINGHAM

January 23: Addresses by J. R. McWane and Oscar Wells on labor and finance. Abstract of Past-President Main's President's Address, Broader Opportunities for the Engineer, read by J. J. Greggan.

February 3: Addresses by R. W. McWane on Efficiency Work in the Emergency Fleet Corporation, by Col. T. O. Smith on Finance, and by Secretary Calvin W. Rice.

April 13: Address on Steel Specifications, O. U. Cook.

May 23: Informal banquet.

BOSTON

January 31: Address by Past-President Main on experiences in France. Abstract published in MECHANICAL ENGINEERING.

March 13: Address by H. W. Rowley on Final Disposition of City Wastes.

April 2: Tenth Annual Engineers' Dinner. Addresses by George H. Moses on A League of Nations, by Prof. George Fillmore Swain on Reflections Suggested from a Recent Trip to France and by Richard H. Rice.

May 21: Joint meeting with Boston Society of Civil Engineers. Report on the Chicago Public Service Committee, by Mr. Metcalf.

June 27: Outing at Villa Napoli, Nantasket, Mass., with A.I.E.E., Boston Society of Civil Engineers, Engineering Society of Western Mass., Providence Engineering Society and Worcester Section.

BUFFALO

January 29: Address by Nathan L. Lieberman, on Horsepower Requirements of Aeroplanes and Power Consumption through Parasite Resistance.

April 2: With the Engineering Society of Buffalo; illustrated address by E. S. Collins on Industrial Applications of Electric Furnaces. The meeting was preceded by a dinner.

CHICAGO

January 13: Informal get-together meeting and dinner in honor of Secretary Calvin W. Rice.

January 27: Address on Sugar Manufacturing, by M. J. Kermer.

April 21: With Western Society of Engineers. Papers on the Triplex Process of Making Steel, by Robert J. Young, shown by motion pictures, and on Fatigue of Metals, by Herbert F. Moore, illustrated with motion pictures and lantern slides.

May 29: Ladies' night. Paper on Stress and Strain, by William S. Sadler.

September 30: Address by Mr. Whitten, Chairman, Zoning Committee, Cleveland, O., on City Zoning as It Pertains to the Requirements for Residential and Manufacturing Districts.

CINCINNATI

February 1: Address by Mayor John Galvin on Financial Problems of Cincinnati and by Bert L. Baldwin on Conditions of Light-Railways Service in France.

February 17: Informal address by Secretary Calvin W. Rice.

March 20: Joint meeting with the Engineers' Club. Address by Chas. H. Fox on The Evolution of the Fire Engine.

March 25: Business meeting.

May 15. Illustrated address by Ernest F. DuBrul on Trade with South America.

CLEVELAND

February 4: All-day convention. Addresses in the morning by H. E. Simmons on Rubber and Its Manufacture and by G. W. Shem on Electric Travelling Crane Development (illustrated). The luncheon was followed by an address by J. R. McQuigg on Experiences of Engineers in France. An inspection trip to the National Acme Manufacturing Company's plant occupied the afternoon, after which a dinner was held. C. A. Otis and Colonel G. M. Barnes gave an account of how the big guns were developed.

June 10: Joint meeting with Cleveland Engineering Society. Morning address, Open-Hearth Charging Machine, by S. T. Wellman and I. D. Thomas. Luncheon aboard steamship *City of Buffalo*, followed by a trip to the American Ship Building Plant, Lorain, and a talk by J. C. Workman.

CONNECTICUT

Bridgeport Branch

June 26: With local members of A. S. C. E., A. I. E. E., S. A. E. and A. C. S. Paper on Mechanical Apparatus in the Treatment of the Wounded, by H. W. G. Thompson; paper on Investigation of Case Carburizing and Case Hardening, by Mr. Boeghold.

Hartford Branch

May 12: With American Chemical Society. Inspection trip to Hartford Rubber Works and to Laboratories of Henry Souther Engineering Co. Illustrated lecture on Rubber, by Theodore Whittlesey.

June 5: Address by Hiram Percy Maxim on Sound.

Meriden Branch

February 28: With Meriden Manufacturers' Association. Address by John C. Spence on The Training Department After the War. Addresses by Charles T. Clayton and H. C. Miles.

April 28: Address by Joseph F. Keller on The Machine Making of Dies.

May 20: Address by George H. Thacher on The Hand Stoker, What It Is and What It Does.

June 6: Informal meeting and dinner.

New Haven Branch

January 8: Address by Douglas K. Warner on The Friction of Ball Bearings.

January 31: With Yale Mechanical Engineers' Club. Address by G. Douglas Wardrop on War Aviation in Retrospect; Commercial Aviation in Prospect.

March 7: With Yale Mechanical Engineers' Club. Address by D. C. Buell on Long-Range Navy Guns with Railway Mount.

March 19: Address by President M. E. Cooley.

May 27: Address by Harry Gordon Hayes on Labor Problems.

Waterbury Branch

January 6: Address by R. A. Cairns on Waterbury's Water Supply.

April 8: Solution of Factory Waste Problems. Professor Newlands and others spoke.

DETROIT

January 11: Informal dinner to President M. E. Cooley and Secretary Calvin W. Rice. Address by President Cooley on An Unoccupied Rung in the Engineer's Ladder of Fame, and by Secretary Rice on Broader Opportunities for the Engineer.

April 4: Joint meeting with Detroit Engineering Society. Address by William B. Stout on Commercialization of Air Craft.

ERIE

February 19: Informal address by Secretary Calvin W. Rice.

INDIANAPOLIS

January 15: Informal reception to Secretary Calvin W. Rice.

LOS ANGELES

March 13: Addresses by Secretary Calvin W. Rice and George G. Anderson, Chairman of the Local Society of the A. S. C. E. Address by Professor Ford on Behavior of Steels under Test.

April 16: Address by F. Hornberger on The Manufacture of Gas from Crude Petroleum.

MID-CONTINENT

February 5: Meeting of organization resulting in petition to Council for the establishment of the Mid-Continent Section with headquarters at Tusla, Oklahoma. Secretary Calvin W. Rice addressed the meeting.

May 23: All-day meeting; business meeting in the morning. Inspection of airplanes in the afternoon. Address by Dean J. H. Felgar on What Should Be the Content of a Course of Instruction Designed to Fit a Man to Become a Petroleum Engineer; (*a*) From the Production Standpoint; (*b*) From the Refining Standpoint. Other addresses by George Tayman on Volumetric and Mechanical Efficiency of Gas Compressors with Varying Combinations of Pressure and Vacuum, and by C. E. Pearce on Graphic Methods and Charts for Design of Steam Boilers and Other High-Pressure Vessels. Following an informal dinner addresses were given by O. J. Berand on Appraisement and Valuation of Oil and Casing-Head Properties; by P. F. Walker on Industrial and Manufacturing Possibilities in the Mid-Continent Section; by Paul Bateman on Tank-Car Maintenance; and by W. S. Smith on Effects of Compressed Air or Gas on Petroleum Oil Production.

MILWAUKEE

January 15: Address by Henry L. Dale on Engineering Experiences at the Front.

February 13: Address by Secretary Calvin W. Rice.

April 16: Joint meeting with the Engineers' Society of Milwaukee, Milwaukee sections of all National Engineering Societies and the Aero Club of Wisconsin. Illustrated address by George R. Lawrence on Flying, Today and Tomorrow.

May 21: With Engineers' Society of Milwaukee. Address by Arnold Pfau on a Trip to Japan.

MINNESOTA

February 10: Dinner to Secretary Calvin W. Rice by the Minneapolis Steel & Machinery Co. Addresses by Secretary Rice, John R. Allen, Max Tolz and James L. Record.

March 4: Address by W. H. Adams on The Manufacture of Beet Sugar.

April 7: Regular monthly meeting.

NEW ORLEANS

April 14: Address by W. B. Gregory on Pumping Machinery. Used by the American Army in France.

NEW YORK

January 14: Address by L. C. Marburg on Aims and Organization; by Edwin J. Prindle on The Patent Situation in the United States; and by W. W. Macon, H. L. Aldrich and A. J. Baldwin on their experiences abroad on a trip of inspection of the battlefields of Europe.

February 10: Joint meeting with Founder Societies. Delegates to Joint Engineering Congress spoke on the work of Congress and conditions in France.

February 24: Address by Peter P. Dean on the Application of Electrical Control of Gate Valves, buffet supper served, followed by an address on the Application to Industry of the Personnel Work in the U. S. Army, by Lt.-Col. J. J. Swan. At the close of this address, motion pictures of animated technical drawings for commercial and scientific use, showing electrical starting and lighting systems, the Burroughs adding machine, etc., were shown.

March 26: Engineers' Symposium under the general auspices of the Local Sections of the American Institute of Mining and Metallurgical Engineers, American Society of Mechanical Engineers, and the Society of Automotive Engineers, and in which the members of the American Institute of Electrical Engineers, American Society of Civil Engineers, American Chemical Society, American Electrochemical Society, American Institute of Chemical Engineers, Amercan Society of Heating and Ventilating Engineers, American Society of Refrigerating Engineers, Brooklyn Engineers' Club, Illuminating Engineering Society, Institute of Radio Engineers, Municipal Engineers of the City of New York, Société de Chimie Industrielle, Society of Chemical Industry, and the Society of Naval Architects and Marine Engineers were invited to participate.

The general title of the meeting was The Engineer as a Citizen. Gano Dunn, President of the J. G. White Engineering Corporation, presided, and the following addresses were delivered. The Civic Responsibility of the Engineer, by Philip N. Moore; The Relation of the Engineer to Legislation, by Calvert Townley of the Westinghouse Elec. & Mfg. Company; The Relation of the Engineer to Administration, by Nelson P. Lewis of the Public Service Commission; The Relation of the Engineer to Public Opinion, by Spencer Miller of the Lidgewood Mfg. Company, and The Relation of the Engineer to Production and Distribution, by Comfort A. Adams, President of the A. I. E. E.

April 9: Joint meeting with the Metropolitan Section of the Society of Automotive Engineers in a Symposium on the Heavy Oil Engine.

May 28: Addresses by John M. Goodwin on The Five-Color System of Camouflage and by Frederick Meron on The Layout and Equipment of Factories (illustrated).

June 10: General discussion of report of Committee on Aims and Organization.

ONTARIO

May 16: Discussion on the Metric System: Pro: E. F. Burton and W. Percy Dobson; Con: Chester B. Hamilton and Ernest V. Pannell.

PHILADELPHIA

January 28: Address by William B. Dickson on Relations between Employer and Employee.

February 19: Out-of-town meeting at Wilmington, Delaware. Address by F. A. Wardenburg on Power Development of the Old Hickory Plant.

February 25: Address by E. B. Morden on The Work of the Construction Division of the Army from Coast to Coast.

March 25: Address by Joseph A. Steinmetz on the question

What Are We to Do With Our Returned Aviators and Their Battle Planes?

May 27: Discussion of report of Committee on Aims and Organization.

PROVIDENCE ENGINEERING SOCIETY

January 7: Address by E. L. Wooley on Work Accomplished at Providence for the Emergency Destroyer Program.

February 12: Annual banquet. Among the speakers were P. H. W. Ross, President of the National Marine League of the U.S.A., Leonard W. Cronkite of Boston, Special Agent for the U.S. Department of Labor, Captain Delport of the French Army, representing the French High Commission, Lieutenant J. A. H. Muirhead, Engineer, officer of the British Army, and Alfred D. Flinn, Secretary of the Engineering Council.

As an added feature moving-picture films of the assembling and operation of the 14-inch naval guns at the front were shown by Ensign C. S. McCrae, U.S.N.R.F., of the Naval Bureau of Ordnance.

May 6: Address by Nicholas Stahl on Central Station Growth.

May 13: Address by Mark Whitehead on The Potter and Johnston Automatic Lathe and Its Tooling.

June 7: Inspection trip to the concrete steamship being outfitted by the Lord Construction Company.

June 17: Annual Meeting.

SAN FRANCISCO

February 13: Address by F. A. Anderson on Electric Arc Welding (illustrated).

March 19: Inspection trip to concrete ship being built by the Shipping Board at Alameda.

April 16: Illustrated addresses by Commander Reed, U.S.N., describing a destroyer launched in fifteen days, and by A. P. Allen on The Modern Destroyer and the Part It Played in Winning the World War.

ST. LOUIS

January 24: Address by J. M. Olvin on The Manufacture of Army Cartridges.

January 29: With the Associated Engineering Societies of St. Louis. Address by L. C. Nordmeyer on Refrigeration and Eggs in China.

February 6: Dinner to Secretary Calvin W. Rice.

February 28: Informal dinner. Address by C. B. Lord on Women in War Industries.

March 21: Address by A. S. Langsdorf on Industry, Research and the Engineer.

April 25: Address by Dwight T. Farnum on what the industrial engineer is trying to do in industry.

May 23: Address by Dr. Edward J. Swift on The Human Element in Industry.

June 4: Illustrated address by Wallace C. Capen on Certain New Developments in Rear-Axle Construction.

WASHINGTON

April 30: Addresses by S. W. Stratton on Standardization of Screw Threads; Colonel E. C. Peck on Gage Work of the Ordnance Department for the U.S. Army; H. L. Van Keuren on the Certification of Gages at the Bureau of Standards and C. G. Peters on The Use of Interference Methods in Calibrating Length Standards.

June 6: General discussion on the relation of the mechanical engineer to his work, to the community and to other engineers.

WORCESTER

February 19: Address by M. Eskil Berg on Recent Development of Propelling Machinery for War and Merchant Vessels.

March 6: Address by A. E. Kennelly on Field Ordnance and Field Ordnance Appliances.

May 28: Discussion on Fuel Conservation by J. F. Tinsley.

THE SPRING MEETING

Detroit, Mich., June 16 to 19

The Spring Meeting of the Society was held in Detroit, Michigan, June 16 to 19, with headquarters at the Hotel Statler. The outstanding features of the meeting were the large attendance; 1180 registered, of which 638 were members; extended discussion on aims and organization, sessions on industrial relations, research and pulverized fuel and many entertainments.

The discussion[1] of the Report of the Aims and Organization

[1] See Mechanical Engineering, July 1919, for summary of discussion.

Committee began on Monday afternoon and was continued on Tuesday and Wednesday. The largely attended Research Session, lasting all day, showed the awakening interest of engineers in conditions resulting from the war and an appreciation by them of the need for research work in the development of American industries. A session on Industrial Relations, in which the human side of industrial management was developed, evoked so much enthusiasm that a resolution was passed at its conclusion calling for a continuation of the discussion at the Annual Meeting. The discussion of the papers on pulverized fuel on Thursday morning also drew a large attendance.

In accordance with the policy of the Meetings and Programs Committee to secure papers characteristic of the engineering work done in the part of the country where the meeting is held, a Sections Session was arranged for Wednesday morning with papers contributed by the Society's sections of the Mid-West.

On Monday evening the Local Committee arranged an informal reception in the ballroom of the hotel, with a brief address of welcome by Mayor James Couzens, to which President Cooley replied. The reception was followed by dancing. On Tuesday evening a concert was given by the Burroughs Band at the Arena Gardens. On Wednesday, the Committee arranged a sail on Detroit River and Lake St. Clair Flats. This trip, which lasted throughout the afternoon and evening, gave an opportunity of meeting members and friends from all sections of the country. Both the Council and Sections Committee held meetings during the trip.

Many ladies were in attendance at the meeting. Automobile trips were provided, there was a drive around Belle Isle, luncheon at the Detroit Boat Club, tea at Red Run Golf Club, and opportunity was offered to inspect many points of interest, including the United States General Hospital and Priscilla Inn.

The success of the meeting was due to the faithfulness and untiring efforts of the various local committees. A list of the chairmen of these committees follows. General Committee, H. H. Esselstyn; Reception, James Couzens; Printing and Publication, John C. McCabe; Transportation and Information, W. E. Cann; Entertainment, M. W. Taber; Ladies' Entertainment, Mrs. F. G. Ray. The Detroit Local Section committee, E. C. Fisher, chairman, contributed two excellent papers to the meeting.

PROGRAM

Monday Morning, June 16

Registration of members and guests at headquarters. Meetings of Council and Society's Committees.

Monday Afternoon

BUSINESS MEETING

Report of tellers of amendments to constitution, reports of committees, discussion of aims and organization of the Society.

Monday Evening

Reception and dance.

Tuesday Morning, June 17

RESEARCH SESSION

The Present Condition of Research in the United States, Arthur M. Greene, Jr.

The Organization and Conduct of an Industrial Laboratory, A. D. Little and H. E. Howe.

Address by Acting Chairman of Engineering Division of National Research Council, G. H. Clevenger.

Research Work on Malleable Iron, Enrique Touceda.

Reports of Sub-Committees on Flow Meters, Bearing Metals and Lubrication.

Tuesday Afternoon

INDUSTRIAL RELATIONS SESSION

Industrial Personnel Relations, Arthur H. Young.

The Status of Industrial Relations, L. P. Alford.

Tuesday Evening

Musical entertainment and dancing.

Wednesday Morning, June 18

SIMULTANEOUS PROFESSIONAL SESSIONS

SECTIONS SESSION

Central-Station Heating in Detroit, J. H. Walker.

Production of Liberty Motor Parts at the Ford Plant, W. F. Verner.

Fire Engines and the Essentials of Fire Fighting, C. H. Fox.

An Electrical Device for Measuring the Flow of Fluids in Pipes, J. M. Spitzglass.

GAS POWER SESSION

Crude Oil Motors vs. Steam Engines in Marine Practice, J. W. Morton.

A Suggested Formula for Rating Kerosene Engines, D. L. Arnold

Standards for Carburetor Performance, O. C. Berry.

Wednesday Afternoon and Evening

Steamboat trip through St. Clair Flats.

Thursday Morning, June 19

SIMULTANEOUS PROFESSIONAL SESSIONS

FUEL SESSION

Pulverized Coal as a Fuel, N. C. Harrison.

Pulverized Coal for Stationary Boilers, Fred'k A. Scheffler and H. G. Barnhurst.

Economy of Certain Arizona Steam-Electric Power Plants Using Oil Fuel, C. R. Weymouth.

GENERAL SESSION

Elements of a General Theory of Airplane-Wing Design, Walter C. Durfee.

Fans for Driving Electric Generators on Airplanes, Capt. G. Francis Gray, Lieut. John W. Reed and P. N. Elderkin.

Mechanical Lifts, Past and Present, and a New Method for Their Balancing, Lieut. J. F. Robbins.

The Design of Riveted Butt Points, Alphonse A. Adler.

Economical Section of Water Conduit for Power Development, Cary T. Hutchinson.

Thursday Afternoon

Excursions to Ford Motor Company and Packard Motor Company.

No. 1688

THE PRESENT CONDITION OF RESEARCH IN THE UNITED STATES

By Arthur M. Greene, Jr., Troy, N. Y.
Member of the Society

This paper, by the Chairman of the Research Committee of the Society, deals with the conditions under which research is now being carried on in the United States. The author first discusses research in its relation to the technical school and gives a list of the universities having mechanical engineering laboratories. Engineering experiment stations are next considered, lists being given of the stations and of those which publish research bulletins. Coöperative research and the research activities of the Government are next presented and finally the author considers commercial and industrial research work, giving in connection therewith lists of the private research laboratories in the country and of manufacturing companies having their own research facilities.

HISTORY records the development of science from fortuitous observations and from systematic, careful and exhaustive research. The work of the Greeks as shown by Lucretius on the constitution of matter could only have been made after a careful and exhaustive study of the laws of nature. The work of Hippocrates certainly indicates a previous study of anatomy and the action of certain drugs. The studies of Galileo and Newton, of Kepler and Herschel, of Watt and Stephenson, tell of careful thought applied to the interpretation of facts which led to the establishment of laws. The story of Berzelius and his kitchen laboratory illustrates the spirit of adventure into the unknown which brought to us our early quantitative knowledge of the elements. The accidental observations of Galvani, Newton, Bell and Crookes by future development and study led to results of incalculable value to science, while the theoretical studies of Kepler, Maxwell and Hertz predicted results which in giving confirmation to theory gave also confirmation to the correctness of experimental observations.

2 Although all past ages have had men devoted to research, they are not numerous in any one period. If there is one thing, how-

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

ever, that marks the present epoch, it is the prevalence of this idea. In the past, research was carried on by the few, but today as a result of education and as a result of the success of research in many fields one can scarcely read a periodical of any branch of science without finding some article on this subject. For the last fifty years the commercial value of research has become more and more evident to industrial plants. This appreciation has been coincident with the growth of these plants and has made their development possible, while reciprocally the growth of the plants has made extensive research possible. This growth has been intimately connected in the same manner with the growth of our universities and the more extensive education of our people. In spite of the statement of many engineers that the young engineer or college man is of little value when he begins his practical work, these men have proved their worth in the development of the industries and of the present civilization.

3 For many years our colleges and universities have had their laboratories where some research has been done, and with the development of our industries following the Civil War the necessity of commercial laboratories for examining or controlling products was evident. Forty years ago saw many small chemical laboratories and a few testing machines scattered throughout the country. The search for knowledge by some of the investigators and the possibility of application of the results of their researches undoubtedly reacted, so that the laboratories of examination and control became laboratories for investigations in new and unknown fields.

4 Research work, quite general before the present war, became more extensive in overcoming the dastardly appliances of the Hun in devising new apparatus, products and manufacturing methods and in improving quality and production. The war has demonstrated, if demonstration was necessary, the value of research, and it is now the opinion of most of us that this stimulating viewpoint should not be lost and that the war-time interest should be continued. For this reason it is well to consider the present condition of research in the country.

5 Before discussing the matter of the physical condition of the various laboratories there are a few general considerations which should be mentioned. The cost of research in the past has been such that in many cases it could only be undertaken in an extensive way by large corporations. The necessity and value have been evident but the small plant has been unable to inaugurate that

which it has known to be of value. In other instances there have been investigations which have been of such a nature that there would be no commercial gain from their results, although of great use to mankind. The necessity of such work has been clearly seen and appreciated by some, and in this country institutions and foundations have been established by public-spirited citizens to carry on investigations or to give grants to those who are working on problems for the general good. The reference is to such organizations as the Rockefeller Institute for Medical Research, the Carnegie Institution of Washington, and the Sage Foundation.

6 A number of private commercial laboratories have been undertaking commissions for clients, but recently the plan worked out years ago by the Associated Factory Mutual Fire Insurance Companies of New England in their coöperative laboratory and the Insurance Engineering Experiment Station has been applied in the coöperative work of certain industries. These investigations have been undertaken by an industry as a whole or by a group of manufacturers, and the results have been distributed among the contributors to the expense funds, or, in certain cases, the results of the experiments have been freely given to the world.

7 The various states in our Union and the National Government have believed that they should make investigations for the farmer, and for over thirty years agricultural experiment stations have been carrying on research in relation to soils, crops and live stock and within the last fifteen years the appreciation of their duty to the manufacturers and the industries has been shown by the establishment of engineering experiment stations. Here the general problems of the manufacturers may be solved and the resources of the state developed.

RESEARCH IN TECHNICAL-SCHOOL LABORATORIES

8 To turn now to the present research activities, let us consider first our universities and technical schools. The general equipment of the university laboratories is planned to give training to the undergraduate in methods and to illustrate certain laws. The engineering laboratories are equipped so that research work is possible, but in many cases the schedule for instruction work is so heavy that little or no research work can be done. Nevertheless, under these adverse conditions some work of great value has been produced during the last thirty years in the technical schools by faculty members, graduate students and even by undergraduates,

and the proceedings of our engineering societies indicate the extent of this work. The lack of time for experimentation has been cared for in the engineering experiment stations by the employment of full- or part-time investigators. In this way work can be carried on continuously to a conclusion. At present the disorganization of these laboratories by war activities has caused most of the work to cease. According to numerous letters, however, a return to normal conditions is looked for within a year. The laboratories of chemistry, physics, biology and the other sciences have been doing much graduate research work. This has been of a theoretical nature rather than of the applied form of research more evident in our engineering laboratories. The small number of graduate students of engineering has partially accounted for the limited amount of research from the engineering schools.

9 The equipment of these technical schools is usually quite diversified and adapted for research work of a varied nature. The equipment has been planned in many cases for certain problems and in some instances special contributions have been made by some associated industries for equipment to make investigations of problems of that industry. Thus, at Johns Hopkins University the gas interests in and around Baltimore donated a fund for the equipment of a laboratory to study gas manufacture and its by-products. At the Carnegie Institute of Technology at Pittsburgh a laboratory for rolling-mill research and instruction is being established from funds which are contributed by a number of steel manufacturers.

10 One of the great needs of the present time as voiced by directors of a large Government laboratory and of a large commercial laboratory is the need for more research men. Research demands a man of clear vision, great imagination, tremendous resources, absolute honesty, good training and devotion to work. The love of the work will have to be the incentive as in many cases the monetary returns are small. If our colleges of engineering and science could by some means instill into more men the great desire for discovery through research, they would aid much in the contributions of this age to the future. Training is also necessary, and that should be done by men engaged in research.

11 The field of research is broader than ever. As a great philosopher expressed it, the relation between the Known and the Unknown is that of the surface of the sphere: the greater the sphere of knowledge becomes, the greater the surface of contact with the unknown.

ENGINEERING EXPERIMENT STATIONS

12 To aid the work of research for the industries and manufacturers by supporting men on whole or part time to carry out investigations, engineering experiment stations have been established in many state universities. These have been active and the development of such institutions is considered by some to be of such national importance that several bills have been introduced in Congress for Government aid in establishing them throughout the United States.

13 The Engineering Experiment Station of the University of Illinois, organized in 1903, usually comes to mind when discussing this question, although there are fourteen such stations at other state universities. The Engineering Experiment Station of the University of Illinois up to January, 1919, has issued 110 bulletins and 10 circulars on its researches. Twenty-eight of these deal with structural problems, 28 with problems relating to fuel, its mining, storing, combustion and analysis, 10 with problems of mechanics, strength of materials and machine design, 14 with heat problems and 11 with problems of electricity and electrochemistry. These papers are sent free of cost to interested parties in some cases, and in other cases a nominal charge is made.

COÖPERATIVE RESEARCH

14 While discussing the subject of the experiment station, the possibility of the coöperative research as shown by present conditions and the different methods of solving this problem should be mentioned. The problems of hot-air-furnace heating have been solved by empirical rules which have had little if any scientific foundation. An association of builders of hot-air furnaces has granted the Engineering Experiment Station of the University of Illinois certain funds of money to finance an investigation of these problems, the results of the investigation to be made public at its conclusion.

15 The problems of metal rolling are complex and have been studied in the past with difficulty because investigations must not interfere with production. The coöperative plan of equipping a full-size rolling mill at the Carnegie Institute of Technology equipped with special apparatus for varying conditions and making quantitative determination of different data exemplifies what is being done by another industry.

16 The paper by Prof. Enrique Touceda at this meeting illustrates how the manufacturers of malleable iron have formed an association to employ a private laboratory to become their research laboratory for the purpose of improving and making uniform a product which was irregular in its properties.

17 The Canners' Laboratory in Washington, D. C., gives another example of the coöperative method of an industry. In this laboratory the problems of the proper harvesting, handling, storing and canning of natural food products has been studied and the industry guided.

18 The Mellon Institute of Research of the University of Pittsburgh is unique and illustrates a development of research by funds contributed to a laboratory by individuals, corporations or industries for the solution of problems confronting them. The Institute was organized about thirteen years ago by Dr. Robert Kennedy Duncan, and the contribution of funds for the support of the research was continued by each contributor for one or more years. The money so received served to pay the salary of the man or men on a special piece of research work and to pay for very special apparatus. The Institute houses the research, furnishes ordinary supplies and apparatus, affords library and consultation facilities and directs the work. The investigations are made for the donor and the results belong to him. At present the work is under the charge of a director, acting through two assistant directors in charge of the fellows on individual and multiple fellowships. In the Institute a method of developing complete unit experimental plants to study processes for certain donors has been used. In this way commercial processes have been developed from laboratory research in a way not done in many other research laboratories.

19 At the University of Michigan the Detroit Edison Company has established a number of fellowships for research. This indicates another method of coöperative effort and the utilization of the equipment of our educational institutions.

20 A list of coöperative efforts in research must mention the work of the laboratories of the Factory Mutual Fire Insurance Companies. The work of this association has covered many years, some of its bulletins being issued over thirty years ago. Many papers and discussions were contributed to the early TRANSACTIONS of this Society from its staff.

EDUCATIONAL INSTITUTIONS HAVING MECHANICAL ENGINEERING LABORATORIES

21 The educational institutions with which the Research Committee corresponded and which are equipped with mechanical engineering laboratories are enumerated below.

Alabama

Alabama Polytechnic Institute, Auburn, Ala. Dean J. J. Wilmore.
University of Alabama, University, Ala. Prof. George Jacob Davis.

Arizona

University of Arizona, Tucson, Ariz. Prof. W. W. Henry.

Arkansas

University of Arkansas, Fayetteville, Ark. Prof. B. N. Wilson.

California

University of California, Berkeley, Cal. Prof. B. F. Raber.
Leland Stanford Junior University, Stanford Univ., Cal. Prof. W. F. Durand.

Colorado

University of Colorado, Boulder, Colo. Prof. J. A. Hunter.

Connecticut

Yale University (S.S.S.), New Haven, Conn. Prof. L. P. Breckenridge.

Delaware

Delaware State College, Newark, Del. Prof. M. van G. Smith.

Georgia

Georgia School of Technology, Atlanta, Ga. Prof. R. S. King.

Illinois

Armour Institute of Technology, Chicago, Ill. Prof. G. F. Gebhardt.
Lewis Institute, Chicago, Ill. Prof. A. W. Moseley.
Northwestern University, Evanston, Ill. Prof. H. S. Philbrick.
University of Illinois, Urbana, Ill. Dean C. R. Richards.

Indiana

Purdue University, Lafayette, Ind. Dean C. H. Benjamin.
Rose Polytechnic Institute, Terre Haute, Ind. Prof. F. C. Wagner.

Iowa

Iowa State College of Agriculture and Mechanic Arts, Ames, Iowa. Prof. M. P. Cleghorn.
State University of Iowa, Iowa City, Iowa. Prof. S. N. Woodward.

Kansas

University of Kansas, Lawrence, Kan. Dean P. F. Walker.
Kansas State Agricultural College, Manhattan, Kan. Dean A. A. Potter.

Kentucky

State University of Kentucky, Lexington, Ky. Dean F. P. Anderson.

Louisiana

Tulane University of Louisiana, New Orleans, La. Prof. W. B. Gregory.

Maine

University of Maine, Orono, Me. Prof. W. J. Sweetser.

Maryland

Johns Hopkins University, Baltimore, Md. Profs. C. C. Thomas and A. G. Christie.

Massachusetts

Massachusetts Institute of Technology, Cambridge, Mass. Prof. E. F. Miller.

Harvard University, Cambridge, Mass. Prof. L. S. Marks.

Tufts College, Tufts College, Mass. Dean G. C. Anthony.

Worcester Polytechnic Institute, Worcester, Mass. Prof. W. W. Bird.

Michigan

University of Michigan, Ann Arbor, Mich. Dean M. E. Cooley.

Michigan College of Mines, Houghton, Mich. Pres. E. F. McNair.

Michigan Agricultural College, East Lansing, Mich. Dean G. W. Bissell.

Minnesota

University of Minnesota, Minneapolis, Minn. Prof. J. J. Flather.

Missouri

University of Missouri, Columbia, Mo. Prof. H. W. Hibbard.

Washington University, St. Louis, Mo. Prof. E. L. Ohle.

Nebraska

University of Nebraska, Lincoln, Neb. Dean O. V. P. Stout.

New Jersey

Stevens Institute of Technology, Hoboken, N. J. Prof. F. L. Pryor.

Rutgers College, New Brunswick, N. J. Prof. R. C. H. Heck.

New Mexico

New Mexico College of Agriculture and Mechanic Arts, State College, N. Mex. Dean A. F. Barnes.

New York

Polytechnic Institute of Brooklyn, Brooklyn, N. Y. Prof. E. F. Church.

Cornell University, Ithaca, N. Y. Prof. H. Diederichs.

Columbia University, New York City. Dr. C. E. Lucke.

New York University, New York City. Director of Testing Lab. C. P. Bliss.

Clarkson College of Technology, Potsdam, N. Y. Prof. A. R. Powers.

Rensselaer Polytechnic Institute, Troy, N. Y. Prof. A. M. Greene, Jr.

North Carolina

North Carolina College of Agricultural and Mechanic Arts, W. Raleigh, N. C.

North Dakota

North Dakota Agricultural College, Agricultural College, N. D.

Ohio

University of Cincinnati, Cincinnati, Ohio. Prof. A. L. Jenkins.

Case School of Applied Science, Cleveland, Ohio.

Ohio State University, Columbus, Ohio. Prof. W. T. Magruder.

Oklahoma

University of Oklahoma, Norman, Okla.

Oklahoma Agricultural and Mechanical College, Stillwater, Okla.

Oregon

Oregon State Agricultural College, Corvallis, Ore.

Pennsylvania

Lafayette College, Easton, Pa. Prof. Donald B. Prentice.

Bucknell University, Lewisburg, Pa.

University of Pennsylvania, Philadelphia, Pa. Prof. R. H. Fernald.

Carnegie Institute of Technology, Pittsburgh, Pa. Prof. W. Trinks.

University of Pittsburgh, Pittsburgh, Pa.

Lehigh University, South Bethlehem, Pa. Prof. Arthur W. Klein.
Pennsylvania State College, State College, Pa. Prof. E. A. Fessenden.
Swarthmore College, Swarthmore, Pa. Prof. G. F. Blessing.

Rhode Island
Rhode Island State College, Kingston, R. I.
Brown University, Providence, R. I. Prof. W. H. Kenerson.

South Dakota
South Dakota State College of Agricultural and Mechanic Arts, Brookings, S. D.
University of South Dakota, Vermillion, S. D. Prof. M. W. Davidson.

South Carolina
Clemson Agricultural College, Clemson College, S. C. Prof. W. B. Earle.

Tennessee
University of Tennessee, Knoxville, Tenn.
Vanderbilt University, Nashville, Tenn. Prof. C. S. Brown.

Texas
University of Texas, Austin, Tex.
Agricultural and Mechanical College, College Station, Tex.

Utah
University of Utah, Salt Lake City, Utah.

Vermont
University of Vermont, Burlington, Vt. Prof. E. Robinson.

Virginia
Virginia Polytechnic Institute, Blacksburg, Va.
University of Virginia, Charlottesville, Va.

Washington
State College of Washington, Pullman, Wash.
University of Washington, Seattle, Wash.

West Virginia
West Virginia University, Morgantown, W. Va.

Wisconsin
University of Wisconsin, Madison, Wis.

Wyoming
University of Wyoming, Laramie, Wyoming.

UNIVERSITIES HAVING ENGINEERING EXPERIMENT STATIONS

University of Arizona, U. S. Bureau of Mines Station, Charles E. Van Barneveld, Supt., Tucson, Ariz.
University of Illinois, Charles R. Richards, Director, Urbana, Ill.
Iowa State College of Agriculture and Mechanical Arts, Dr. S. W. Beyer, Acting Director, Ames, Iowa.
Kansas State College of Agriculture, A. A. Potter, Director, Manhattan, Kan.
University of Kansas, Perley F. Walker, Director, Lawrence, Kan.
University of Minnesota, School of Mines, W. R. Appleby, Director, Minneapolis, Minn.
University of Missouri, E. J. McCaustland, Director, Columbia, Mo.
Missouri School of Mines, Mining Experiment Station, A. L. McRae, Rolla, Mo.
Pennsylvania State College, R. L. Sackett, Director, State College, Pa.

Purdue University, C. H. Benjamin, Director, Lafayette, Ind.
Agricultural and Mechanical College of Texas, J. C. Nagle, Director, College Station, Tex.
University of Utah, J. F. Merrill, Director, Salt Lake City, Utah.
University of Washington, C. F. Magnusson, Acting Director, Seattle, Wash.
University of Wisconsin, Address Director, Madison, Wis.

UNIVERSITIES ISSUING ENGINEERING RESEARCH BULLETINS

Rensselaer Polytechnic Institute, P. C. Ricketts, Director, Troy, N. Y.
University of California, Charles Derleth, Jr., Editor, Berkeley, Cal.
University of Minnesota, Address Director, Experimental Engineering Dept., Minnesota, Minn.

GOVERNMENT ACTIVITIES IN RESEARCH

22 The Bureau of Standards at Washington and Pittsburgh, the laboratory of the United States Bureau of Mines at Pittsburgh, the Food Laboratory and Forest Products Laboratory of the Department of Agriculture and the Naval Experiment Station at Annapolis, are a few of the Government activities interested in research.

23 At the Bureau of Standards research work is being done in physics, chemistry, metallurgy, manufacturing and engineering. There is hardly a branch of human endeavor which is not touched by this enormous research laboratory. In 1917–1918 there were over 1400 employees connected with the Bureau, and accounts aggregating more than $3,400,000 were handled. During this year the Bureau issued fifty-three publications and these may be obtained through correspondence.

24 The primary work of the Bureau is the definition and fixing of standards of measurements, standard constants, standards of quality, standards of performance and standards of practice and to do this they have divided the scientific and technical staff into a division of weights and measures, a division of heat and thermometry, an electrical division, an optical division, a chemical division, a materials division, an engineering research division, a metallurgical division, and a ceramic division. Each division is under a Chief of Division and under him there are numerous experts and assistants. The Bureau feels that its function is one of service to the nation and it endeavors to aid all who apply for information or guidance.

25 The work of the Gage Section of the Bureau of Standards during the recent war activities must be remembered as of the greatest importance. This department undertook to regulate the

gages used in the various manufacturing plants through its headquarters in Washington and its branches in the East and the Middle West. The Section has developed instruments for testing screw-thread gages for profile and pitch, instruments for end measurements, and in fact it is prepared to test any commercial gage or templet for accuracy. The Section has studied the salvaging of gages and is vitally interested in the problems of duplicate production.

26 The activities of the laboratories of the Bureau of Mines and the Department of Agriculture are devoted to their special fields of endeavor, and in each case scientists of training and experience are in charge of the research.

27 The U. S. Naval Experiment Station at Annapolis, Md., is used to study the apparatus and materials used by the U. S. Navy or certain Government bureaus. The work consists in making tests on these, and in addition researches regarding the general laws underlying the apparatus have been undertaken. The Station is well equipped with apparatus and an excellent staff. The work of the Station is for Government information, but frequent papers by members of the staff appear at times before various technical societies.

28 The Forest Products Laboratory of the United States Department of Agriculture at Madison, Wis., is devoted to problems relating to the applications of forest products.

29 The Watertown Arsenal is equipped for research in materials of engineering. The reports from this laboratory have been for years the source of many data on the strength of materials.

30 The Philadelphia Navy Yard is equipped for research in fuel oils, while the Washington Navy Yard is equipped for testing ship models, propellers, airplanes and air propellers. The wind tunnels and testing basin are of special merit.

31 The research activities of the American Society of Heating and Ventilating Engineers in connection with the Pittsburgh Laboratory of the United States Bureau of Mines is important and illustrates the activities of certain groups of scientists and engineers. This society plans to make researches regarding problems arising in its field of endeavor for the benefit of the profession and the public. In this project the expenditure of $20,000 per year for a number of years is proposed.

32 The dentists of the United States have a number of laboratories devoted to the solution of their problems. The Research Institute of the National Dental Association in Cleveland is one which illustrates the coöperative endeavor of allied scientists. At

one of their recent conventions the dentists have declared themselves in favor of giving the results of all research to the profession without compensation.

33 According to the public press an Institute for Drug Research is about to be established. It is to be supported by the profits of the Chemical Foundation which has been formed to take over Government-held German patents on chemicals, dyestuffs and drugs. In the plan as outlined this work would be done by chemists, pharmacologists and physiologists working in the "vast undiscovered field of drugs and chemicals for the welfare of mankind."

COMMERCIAL RESEARCH

34 The private research laboratories of the country are primarily devoted to investigations of materials for commercial purposes, to check products or raw materials or to improve the product. These are quite numerous and the list of laboratories given below shows in a partial way the private research resources of our country. Much of the work done by these laboratories is of the nature of inspection, but in many of them the commissions undertaken for clients have been of a true research nature, in finding the cause of defects, the methods of improving product and in some cases planning actual production methods or processes.

35 Many of these laboratories have been in existence for almost a half a century; others have been developed in the last decade from a local need for such institutions. The work of such a laboratory is to be described in a paper at this meeting of the Society and the varied nature of its activities will be seen.

36 In the correspondence of the Research Committee with the various private laboratories, the willingness to coöperate in the work of the Committee was coupled with the statement that most of their research work was for clients and, therefore, could not be made public. The Committee hopes that in some cases the persons for whom the work is done will contribute information after this has been properly protected.

37 The work of these private laboratories covers all fields of investigations and new equipment is obtained in many cases for special investigations. In some cases a laboratory has been specializing in problems of a definite character and its equipment for this work is expensive and complete. The list given below represents the private laboratories known to the Research Committee at the present time.

PRIVATE LABORATORIES

Booth, Garrett & Blair, Philadelphia, Pa.
Cement plants, cement, building materials, chemistry.
Dayton Engineering Laboratories, Dayton, Ohio.
Spark plugs, auto and airplane engines, wireless apparatus.
Detroit Testing Laboratories, Detroit, Mich.
Dairy products, lubricants, soaps, road and building materials.
Electrical Testing Laboratories, New York.
Instruments, lamps, insulation, fuels, lubricants, paper.
Fitzgerald Laboratory, Inc., Niagara Falls, N. Y.
Electrochemistry.
James H. Herron, Cleveland, Ohio.
Metallurgy, chemistry, ceramics, inspection.
Robert W. Hunt & Co., Chicago, Ill.
Metallurgy, chemicals, materials, apparatus, inspection.
Institute of Industrial Research, Washington, D. C.
Institute of Fermentology, Chicago, Ill.
B. B. Lathbury, Philadelphia, Pa.
Cements, materials, inspection.
Leeds & Northup, Philadelphia, Pa.
Electric instruments, electrochemistry, heat treatment.
Lehigh Valley Testing Laboratory, Allentown, Pa.
Cements.
Lincoln Hanson and Abbott, Portland, Me.
Electrical apparatus.
Arthur D. Little Co., Inc., Cambridge, Mass.
Chemical analysis, processes, paper, foods, textiles, metallurgy.
New York Testing Laboratory, New York, N. Y.
Pittsburgh Testing Laboratory, Pittsburgh, Pa.
General testing, metallurgy, chemical, materials, inspection.
Rome Testing Laboratory, Rome, Ga.
Rubber Trade Laboratory, Newark, N. J.
S. P. Sadler, Philadelphia, Pa.
Chemical analysis, processes.
Henry S. Spackman Engineering Co., Philadelphia, Pa.
Cements, materials, chemicals.
Textile Trade Laboratory, Newark, N. J.
Enrique Touceda, Albany, N. Y.
Metallurgy, chemical analysis.
Underwriters Laboratories, Chicago, Ill.
Fire-protective apparatus, electrical apparatus, insulation, chemical analysis.
United States Conditioning and Testing Co., New York, N. Y.
Textiles.
John H. Yocum, Newark, N. Y.
Leather and oil trade.

high as 200,000 or currents of 12,000 amperes may be obtained. The building is piped with city water, river water, illuminating gas, high-pressure hydrogen, low-pressure hydrogen, oxygen, high-pressure steam, compressed air and vacuum suction and vacuum cleaning. Distilled water is supplied to any room by gravity, and liquid air may also be obtained. Various kinds of gas and electric vacuum and arc furnaces are installed in one part of the building. A furnace is installed for argon purification. Various crushers, grinders, rolls, punches and a 60-ton hydraulic press are installed.

49 The illuminating laboratory, which is distinct from the research laboratory, is devoted to special problems in studying the best lighting units or methods of utilizing these units. The consulting engineering department laboratory, devoted to high-tension phenomena, the testing laboratory for materials, the standardization laboratory for instruments and the development of new instruments represent activities at Schenectady which are devoted to research in their commercial routine duties. Many engineering departments are constantly making investigations which are of a research nature.

50 The work at the laboratories at Lynn and Pittsfield is largely applied to the production problems of these plants and at Harrison and Cleveland the problems of lamp production are studied.

51 The problems of the Westinghouse Company are of a similar nature to those of the General Electric Company.

52 The research laboratory of the Eastman Kodak Company represents the research activity of another manufacturing corporation. The staff of this laboratory consists of about fifty men, some fifteen of whom are specialists. The budget amounts to more than $100,000. The work of the laboratory is devoted to physics, chemistry, to plant problems and new development. In this laboratory full-size apparatus is used at times in making research. As shown by the equipment of many laboratories the study of complete processes in the laboratory on a commercial scale is one of the features of the times. The problems of this laboratory are organic, inorganic and colloidal chemistry, optics, color photography, film products and applications of general photography, chemical products and emulsions.

53 The laboratory of the National Lamp Association, now the Nela Park Laboratory at the National Lamp Works of the General Electric Company at Cleveland, Ohio, represents one of the best-known research institutions of this country. The research work of

this laboratory is devoted to the physics, physiology and psychology of light, the production, utilization and efficiency of luminous energy. In this laboratory with a staff of eight investigators of the highest ability directing the work, all kinds of illuminating problems are studied from every angle. The laboratory has a policy of sending out its experts to study local conditions, and in many cases investigators from other institutions come to this laboratory to carry on research. During the first eight years of its existence from 1908 the laboratory has produced 125 high-grade papers. These are abstracted by the authors and the abstracts published at intervals. In having abstracts made by the authors the important points of the researches are sure to be covered.

54 The laboratory of the Packard Motor Car Company is devoted to the study of materials used in their plant, the inspection and test of supplies and finished work as well as the development of new devices or processes.

55 The great extent of research facilities is disclosed by the following list of laboratories used by the Research Committee.

MANUFACTURING COMPANIES HAVING RESEARCH LABORATORIES

Air Reduction Company, New York City.
Aluminum Castings Company, Cleveland, Ohio.
Aluminum Company of America, New Kensington, Pa.
American Agricultural Chemical Company, New York City.
American Beet Sugar Company, New York City.
American Brass Company, Waterbury, Conn.
American Locomotive Company, Schenectady, N. Y.
American Optical Company, Southbridge, Mass.
American Rolling Mill Company, Middletown, Ohio.
American Sheet & Tin Plate Company, Pittsburgh, Pa.
American Smelting & Refining Company, New York City.
American Telephone & Telegraph Company, New York City.
Amoskeag Mills, Manchester, N. H.
Arlington Mills, Lawrence, Mass.
Armour & Company, Chicago, Ill.
Armstrong Cork and Insulation Co., Pittsburgh, Pa.
Atlantic Refining Company, Philadelphia, Pa.
Babcock & Wilcox Company, Bayonne, N. J.
Baldwin Locomotive Works, Philadelphia, Pa.
The Barrett Company, New York City.
Bausch & Lomb, Rochester, N. Y.
Berlin Mills Company, Berlin, N. H.
Bethlehem Steel Company, So. Bethlehem, Pa.
Browne & Sharpe Manufacturing Company, Providence, R. I.
Buick Automobile Company, Flint, Mich.

Carnegie Steel Company, Pittsburgh, Pa.
L. D. Caulk Dental Company, Milford, Del.
Chalmers Automobile Company, Detroit, Mich.
Champion Spark Plug Company, Flint, Mich.
Columbia Graphophone Company, Bridgeport, Conn.
Corning Glass Works, Corning, N. Y.
Wm. Cramp & Sons, Philadelphia, Pa.
Crucible Steel Company of America, Pittsburgh, Pa.
Curtiss Aeroplane & Motor Company, Buffalo, N. Y.
Curtiss Engineering Co., Mineola, N. Y.
De Laval Steam Turbine Company, Trenton, N. J.
E. I. Dupont de Nemours & Company, Wilmington, Del.
Eastman Kodak Company, Rochester, N. Y.
Thomas A. Edison Company, Inc., West Orange, N. J.
Ford Automobile Company, Detroit, Mich.
Fulton Bag and Cotton Mills, Atlanta, Ga.
General Bakelite Company, New York City.
General Chemical Company, New York City.
General Electric Company, Schenectady, N. Y., Pittsfield, Mass., Cleveland, Ohio, Harrison, N. J., Lynn, Mass.
General Motors Company, Detroit, Mich.
Grasselli Chemical Company, Cleveland, Ohio.
Hudson Automobile Company, Detroit, Mich.
Holcomb Steel Company, Syracuse, N. Y.
Ingersoll-Rand Company, New York City.
International Harvester Company, Chicago, Ill.
International Acheson Graphite Company, Niagara Falls, N. Y.
H. M. Johns-Manville Co., New York, N. Y.
Lackawanna Steel Company, Buffalo, N. Y.
Linde Air Products Company, New York City.
Ludlum Steel Company, Colonie, N. Y.
Midvale Steel Company, Philadelphia, Pa.
National Aniline & Chemical Works, Buffalo, N. Y.
National Carbon Company, Cleveland, Ohio.
National Cash Register Company, Dayton, Ohio.
National Lumber Manufacturers' Association, Chicago, Ill.
Nela Research Laboratory, Nela Park Laboratory, Cleveland, Ohio.
New Jersey Zinc Company, New York City.
New York Shipbuilding Corporation, Camden, N. J.
Newport News Shipbuilding Company, Newport News, Va.
Packard Automobile Company, Detroit, Mich.
Pennsylvania Railroad Company, Altoona, Pa.
Pennsylvania Salt Mfg. Company, Philadelphia, Pa.
Pierce-Arrow Automobile Company, Buffalo, N. Y.
Pittsburgh Plate Glass Company, Pittsburgh, Pa.
Pratt & Whitney Company, Hartford, Conn.
Precision Instrument Company, Detroit, Mich.
Pyroelectric Instrument Company, Trenton, N. J.
Remy Electric Company, Detroit, Mich.

Reo Motor Company, Lansing, Mich.
Sangamo Meter Company, Springfield, Ill.
Sears, Roebuck Company, Chicago, Ill.
Solvay Process Company, Syracuse, N. Y.
Standard Oil Company, New York City.
Studebaker Corporation, South Bend, Ind.
B. F. Sturtevant Company, Hyde Park, Mass.
Taylor Instrument Company, Rochester, N. Y.
Titanium Alloys Manufacturing Company, Niagara Falls, N. Y.
Tidewater Oil Company, Bayonne, N. J.
Union Switch & Signal Company, Swissvale, Pa.
United Gas Improvement Company, Philadelphia, Pa.
United States Industrial Alcohol Company, South Baltimore, Md.
United States Smelting Company, New York City.
United States Steel Corporation, New York City.
Victor Phonograph Company, Camden, N. J.
Welsbach Company, Gloucester, N. J.
Western Electric Company, New York City.
Westinghouse Airbrake Company, Wilmerding, Pa.
Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa.
S. S. White Dental Manufacturing Company, Philadelphia, Pa.

56 I cannot close this paper without referring to one agency which not only has been of utmost importance during the present war but also in its continuance in the times of peace will have a still greater influence on the developments in science and industry. I refer to the National Research Council, the purpose and work of which is best explained by the following executive order issued by the President of the United States, May 11, 1918:

The National Research Council was organized in 1916 at the request of the President by the National Academy of Sciences, under its Congressional charter, as a measure of national preparedness. The work accomplished by the Council in organizing research and in securing coöperation of military and civilian agencies in the solution of military problems demonstrates its capacity for larger service. The National Academy of Sciences is therefore requested to perpetuate the National Research Council, the duties of which shall be as follows:

1 In general, to stimulate research in the mathematical, physical and biological sciences, and in the application of these sciences to engineering, agriculture, medicine and other useful arts, with the object of increasing knowledge, of strengthening the national defense, and of contributing in other ways to the public welfare.

2 To survey the larger possibilities of science, to formulate comprehensive projects of research, and to develop effective means of utilizing the scientific and technical resources of the country for dealing with these projects.

3 To promote coöperation in research, at home and abroad, in order to secure concentration of effort, minimize duplication, and stimulate progress;

tive in all cooperative undertakings to give encouragement to individual initiative, as fundamentally important to the advancement of science.

4. To serve as a means of bringing American and foreign investigators into active cooperation with the scientific and technical services of the War and Navy Departments and with those of the civil branches of the Government.

5. To direct the attention of scientific and technical investigators to the present importance of military and industrial problems in connection with the war and to aid in the solution of these problems by organizing specific researches.

6. To gather and collate scientific and technical information at home and abroad, in cooperation with Governmental and other agencies and to render such information available to duly accredited persons.

Effective prosecution of the Council's work requires the cordial collaboration of the scientific and technical branches of the Government, both military and civil. To this end representatives of the Government, upon the nomination of the President of the National Academy of Sciences, will be designated by the President as members of the Council, as heretofore, and the heads of the departments immediately concerned will continue to cooperate in every way that may be required.

The White House,
May 11, 1918.

(Signed) Woodrow Wilson.

57 The work and activity of the National Research Council will be discussed in one of the papers at this meeting, and I will not attempt to enumerate the various problems which it has undertaken. To make this paper complete, however, I will give an outline of the organization as it existed during the war period.

58 The Council was under the chairmanship of Dr. George E. Hale with three vice-chairmen, an executive secretary, a treasurer and two assistant secretaries. The executive board under the chairmanship of Dr. John J. Carty consisted of the officers of the Council, the chairman and vice-chairman of divisions and the chairman of sections of the General Relations Divisions together with six elected members. The Divisions of the Council were as follows:

1 The Division of General Relations
2 Military Division with its Research Information Service
3 Division of Engineering
4 Division of Physics, Mathematics, Astronomy and Geophysics
5 Division of Chemistry and Chemical Technology
6 Division of Geology and Geography
7 Division of Medicine and Related Sciences
8 Division of Agriculture, Botany, Forestry, Zoölogy and Fisheries.

Each Division was divided into committees and sections, covering special features of the work. There were over one hundred scientists representing various technical and scientific societies, educational institutions, commercial laboratories and manufacturers.

59 The work accomplished by the Council during the war has been so important and many of the investigations and researches which were not completed gave so much promise for the future that the Council has been reorganized to continue its work for the furtherance of science and its applications.

DISCUSSION ON ENGINEERING RESEARCH

THE following general discussion of the subject of engineering research has to do with papers Nos. 1688, 1689 and 1690.

P. F. Walker (written). Professor Greene has truly said that in most of our educational institutions the members of the faculty are so burdened with routine teaching that large amounts of research work cannot be expected. There is good reason, however, why the schools need to give active attention to scientific investigation. It is the schools that must produce the men to go into active professional work, including research. In the schools these men have implanted in them certain ideals which will remain, perhaps unconsciously, as determining factors in their lives. To promote research therefore is of vital importance in order that the student should be brought face to face with research problems, and thus imbibe something of the spirit of the investigator.

A matter in which the writer is at present personally interested and to which he is giving a great deal of attention is the state industrial problem. It is a research side of that new branch of the profession sometimes designated as industrial or commercial engineering. Every state and every community has problems peculiar to itself and it is a function of state educational institutions as well as of the engineering profession to interest themselves in such problems with the aim of rendering service. A combined survey and study of industrial possibilities is being made. This is mentioned because it is a kind of research not always thought of as coming within the scope of the engineer. To the writer, however, it seems that it is an activity which comes distinctly within the terms of that definition which states that engineering is the application of all of the sources of power to the use and convenience of man. In any community, and particularly in one where the natural resources are not yet developed by industries to the full extent that economic conditions will warrant, it is the business of the engineer to encourage and pave the way for new industrial development as truly as it is to develop the natural agencies for its successful prosecution.

F. J. SCHLINK (*written*). The importance of real research work and its possibilities in returning a quite extraordinary profit for a very moderate expenditure cannot be over-emphasized. If this end is to be attained, however, the venture must be undertaken with a broad vision and the investigator must be permitted within reasonable limits to carry on studies which may not promise immediate pecuniary benefits.

It is nothing less than astonishing to find, as we occasionally do, that a long-established product, supposedly fully standardized, is being manufactured without even a superficial knowledge of the simple engineering and scientific facts that underlie its performance. Some of the portable or hand-operated fire extinguishers are splendid examples of this class, and in these simple devices, which are not nearly so complex, from the designing and manufacturing standpoint, as an alarm clock or a lawn mower, the varied and manifest types of mechanical and operating failures presented are well-nigh unbelievable. One can say with a high degree of certainty that the expenditure of $500 in a real engineering study of the problems of material and function in any one of several such extinguishers would have made unnecessary the waste of many thousands of dollars in ineffective designs, some of which are an absolute menace in that they give the owner a false and unfounded sense of security against fire hazards.

The condition of American business in which large profits could be made without the necessity of careful and rigorous attention to engineering research and standardization is rapidly passing away in the face of the present perplexing industrial situation. The advancing costs of labor and material, with the concurrent unwillingness of the public to pay increased prices for the manufactured product it consumes, are bringing to every manufacturer a problem of almost crucial character, of holding down or reducing his manufacturing costs without depreciating his product. This difficulty must be met on the one front by research and standardization, and on the other by the efficient management of labor; and the former, though not so well advertized or perhaps even so highly regarded, is likely to prove at least as powerful an ally as the latter.

ARTHUR J. WOOD (written). Professor Greene states that one of the great needs of the present time is for research men and follows with a statement which in itself explains why there are no more men available. He says, "The love of the work will prove to be the incentive as in many cases the monetary returns are small." I believe that the author is conservative in saying "many cases," — it should read "most cases."

Monetary returns will never make a research engineer out of one whom nature never fashioned for such work, but it will produce results of unmeasured value from the one who has the love of and ability for research work if it frees him from financial cares which he otherwise must carry.

Surely the need is for more research men and the industrial laboratories look primarily to the colleges for their material, but what are the colleges, as a whole, doing to develop men along these lines?

In the period of awakening to the value of research there will be plunges into subjects without proper preparation or preliminary study and analysis, and some half-completed results of tests will doubtless be put forth as research, partly because the colleges cannot or do not pay salaries to teachers and investigators which would train the right men in the proper lines. The making of a research worker is a long-time process, and it calls for a normal, straight-thinking mind, well informed and evenly balanced, which must be left free to get results. One of the apparent needs for men in collegiate research is for more leisure in which to think out methods, to plan the work in fields not already well occupied and to keep the mind thoroughly saturated with the subject at hand. How much leisure should be granted? The man himself should be the one best qualified to judge.

Regarding industrial vs. pure research, there should be no conflict; no line of demarkation can be drawn between the two although they are essentially different. One is helpless without the other. No research is so "pure" but that it may some day lead to results of commercial value. It is unfortunate that some engineers do not have a true conception or an adequate appreciation of the value of scientific research work.

The Society through its Committee on Research and the various sub-committees may be an important factor in guiding some of this work. It may help to raise the standards of pure research so that industrial and educational institutions alike will accept the definition of research as given by the late Dr. R. H. Thurston as the "art of revelation and prophecy." The Society must be a leader in the great work and place research in its accepted and well-deserved place among engineering enterprises.

H. S. Coleman. The system of research at the Mellon Institute was formulated by the late Robert Kennedy Duncan and placed in experimental operation at the University of Kansas in 1906. In 1911 the system was inaugurated at the University of Pittsburgh, and in 1913 established on a permanent basis there through a gift, by Messrs. Andrew William and Richard Beatty Mellon, bankers of Pittsburgh, of a modern research building and an endowment to cover the general overhead expenses and salaries of the administrative staff.

Any company or association of manufacturers having a problem or group of problems requiring investigation may become the donor of an industrial fellowship by contributing to the Mellon Institute a definite amount of money, for a period of not less than one year. The foundation sum must be adequate for the purchase of all necessary special apparatus or other equipment as well as to furnish the annual stipend of the research man or men selected to work on the particular problem. The Institute houses the investigatory work, furnishes it with the use of its permanent equipment, affords library and consultative facilities, gives careful direction to the progress of the research, and provides an atmosphere which is conducive to productive inquiry. All results obtained during the course of the industrial fellowship belong exclusively to the donor.

The Institute is not, in any sense of the word, a commercial institution, being entirely independent and deriving no financial profit from any investigation conducted under its auspices. In fact, during the last fiscal year, it was necessary to draw upon the endowment fund for almost $70,000.

Up to the present time the engineering research carried on at the Mellon Institute has been rather limited in scope as the present building and equipment were designed mainly for chemical research. There are, however, three fellowships at present in operation involving engineering research, and plans are now under way for the construction of an engineering research building which will provide adequate facilities for extensive engineering investigation.

Investigations are usually worked out in three stages. First, there is the laboratory stage; then comes the unit plant or semi-commercial stage and finally the process is placed on a commercial basis in the plant of the donor. The fellow who has developed the process through the laboratory and unit-plant stages is usually at this time taken over by the company and placed in direct charge of the new process.

As a result of the investigations involving the development of new processes, a large and valuable collection of special equipment has been acquired. This equipment, in most cases becomes the property of the Institute at the expiration of the fellowship for which it was purchased, and is available for further use in connection with new problems.

John R. Allen. The American Society of Heating and Ventilating Engineers decided about a year ago to establish a research laboratory. After a thorough investigation of the available locations the committee in charge decided to locate this laboratory at the Bureau of Mines in Pittsburgh. The Bureau of Mines kindly agreed to supply the necessary laboratory space in their new experimental building.

The funds for carrying on the work of the laboratory have been largely supplied by the members of the society and by manufacturers interested in research of this character.

The work in general will not be of a commercial nature but the problems taken up will be of a fundamental character. The committee in charge of the work has laid down three principal activities for the Bureau:

1 The collecting of all the references covering research work along the lines of heating and ventilation. This will be a card index and will be particularly for the use of the Bureau but will be available for all members of the society.

2 The work of research, which is divided into two portions

— research at the laboratories in Pittsburgh and research in other institutions.

3 The standardizing of all instruments and methods of testing. The purpose of this is to establish uniform conditions of arriving at conclusions so that all results can be properly compared and to ascertain what degree of accuracy instruments should give in order that results may be reliable.

It is not the intention of the Bureau of Research to repeat work that has already been carefully and accurately done or to develop apparatus at the Bureau which is already well developed in other institutions. It is the intention of the Bureau to make use of all the available equipment in the country as far as is possible. Part of the work will be the collecting and editing of this material and the conducting of experiments along lines not hitherto conducted in other institutions.

The intention of the Research Laboratory in general is to give heating and ventilating engineers more definite data in regard to the main questions arising in their business so that the work of the profession can be done with greater accuracy and certainty than is possible at the present time.

ZAY JEFFRIES.[1] The Aluminum Castings Co. is engaged in a line of endeavor which for a long time has had for its object the refining of methods of production.

The Aluminum Casting Company's laboratory employs between 80 and 90 individuals. It is a two-story building 50 by 230 ft.; and its buildings and equipment are valued at approximately $150,000. The annual expenditure for this work alone is about $300,000. We are finding out a great many things in connection with non-ferrous alloys, but we are only advanced far enough to apply these fundamentals to our industry. Our work is new, and yet its application is even now producing results, and no doubt will have more extended use in the future.

CHARLES RUSS RICHARDS. Some 25 years ago the first attempt was made to provide for federal aid in the establishment of Engineering Experiment Stations, but for one reason or another the effort to secure congressional action was futile. A few years later,

[1] Director of Research Laboratory, Aluminum Castings Company, Cleveland, Ohio.

the officers of the College of Engineering of the University of Illinois succeeded in interesting the university authorities in engineering research, and on December 8, 1903, the Engineering Experiment Station was established by an act of the Board of Trustees. It has since undertaken industrial research in a great variety of lines.

The Engineering Experiment Station is an organization within the College of Engineering. The control of the station is vested in the Executive Staff which is composed of the Director and his assistant, the heads of the several departments of the College of Engineering, and the Professor of Applied Chemistry. The members of the faculty are encouraged to devote to research work as much time as they may have at their disposal. If any one desires to undertake the solution of an important problem, we attempt to relieve him of some of his exacting teaching or administrative duties so that he can give a portion of his time to the research. Most of the research work in the station, however, is carried on by the Research Corps composed of various full-time and part-time research assistants and special investigators who are employed for specific purposes.

In addition to the research work conducted from university funds, the Engineering Experiment Station has undertaken from time to time coöperative investigations of problems of importance to an industry or a group of industries, under an arrangement which provides that the coöperating agency shall pay the principal expenses connected with the investigation. At the present time there are four or five such investigations in progress.

As an institution supported by the state, it is necessary for us to safeguard the interests of the public in all research work which we undertake, whether it is at our own initiative or in coöperation with outside agencies, by publishing the results of the investigations and by reserving the ownership thereof.

The Engineering Experiment Station at the present time has an annual budget of approximately $60,000, but since it uses all of the facilities afforded by the College of Engineering, the annual expenditures for research work are considerably in excess of the regular budget. However, in view of the fact that our funds are apportioned to ten departments, the annual amount available for any one department is necessarily small. It is my hope that we may increase the number of coöperative investigations under the general direction of the Engineering Experiment Station, so that each member of our Faculty, who is a specialist in his particular line, may have an opportunity to direct a corps of special investigators on some important investigation, which will insure results of real value to the profession. It is only through coöperation of this sort that we can hope to undertake the solution of some of the larger problems, for no educational institution can supply sufficient funds to conduct research work which involves large expense. If we can secure the coöperation of the industries of Illinois and of the adjoining states in such a program, I am sure that the work of the station can be greatly extended and improved.

Charles H. Benjamin. The engineering experiment station at Purdue is but two years old. We now have a paid staff of workers, and a good field in which to work and our future is very bright. Our object in establishing the station was primarily to coördinate the research work which has been carried on at Purdue for so many years, to enable us to do it more satisfactorily and to publish the results more widely. Another purpose in forming this organization in coöperation with the industries of the state is that primarily we are interested in Indiana, its products, its manufactures and its future, and it is our first aim to collaborate with the men of Indiana to increase this productiveness. There is a certain advantage in a university laboratory. A university or engineering experiment station is commercially unprejudiced, and this removes from it any suspicion of commercial interest. Sometimes we are required to investigate problems that perhaps could be better investigated by a private concern, but that establishment feels that its own efforts would be misinterpreted and that the results from the engineering experiment station will receive more universal credence and support. The organization and our methods of work are very similar to those which Dean Richards has so well outlined. Among the investigations under way may be mentioned the testing of road materials and coöperation with the highway commissioners in the building of good roads, the efficiency of the carburetor, and the testing of farm tractors, in which we are coöperating with the agricultural school of the university. We expect to see a testing plant at the university which will bear somewhat the same relation to farm tractors as the locomotive laboratory does to the railroad, and our only difficulty is to take care of the great amount of work that it brings to us without solicitation.

In conclusion, I wish to express my appreciation of the work of the Research Committee. I do, however, want to urge caution, not on the part of the Committee, but on the part of some governing agencies in trying to shape research work. The research man is a genius. He is born, not made. He is peculiar and he must work in his own way and on his own initiative. He must not be interfered with and he must be allowed to follow a thing according to his bent.

J. R. Bibbins commended the spirit of the Research Session and the character of the contributions produced by Professor Greene's Committee. There were, however, certain broad aspects of the subject, involving state and even national policy, which seemed to have been forgotten in the special interest centering on research problems. These he brought forcibly to mind by the slogan "Millions for Agriculture, not one cent for Industry — Why?"

He then presented the following resolutions:

Whereas, it is a matter of generally accepted concern that any advanced policy of The American Society of Mechanical Engineers, in common with other complementary learned societies and associations, should incorporate prompt and extended recognition of scientific and industrial research as a specific means of advancing technology and proper industrial development of the nation; and,

Whereas, the Great War has fully demonstrated the principle of research, applied broadly as well as in detail, to be of inestimable benefit in advancing national welfare; and,

Whereas, the Committee on Aims and Organizations has specifically recommended that The American Society of Mechanical Engineers take a more active interest in this field; therefore, be it

Resolved, That it be declared the sense of this Research Session of the A.S.M.E., that Engineering Council shall be encouraged to undertake active support of a plan and organization of scientific and industrial research to the end that the following objects may be accomplished with all reasonable dispatch:

1 To secure the passage in the present Congress of a special act furthering nation wide research in State units through congressional appropriations for this purpose under the general coördination of Engineering Council, or other national agency thoroughly representative of the engineering profession.

2 To encourage trade, industrial, and utility associations to interest themselves in the advance of the Arts and the constructive benefits to be derived from research work in their respective fields and to coöperate with them in their efforts in these directions.

3 To encourage the various research institutions or instrumentalities now or to be established by the Federal and State governments in their close coöperation in this general research policy.

4 To encourage and assist in the establishment of organized departments of engineering research at the various universities, adequately equipped with material and personnel and to bring such department as closely as possible in touch with the vital problems of industrial development confronting the Nation.

5 To institute organized publicity with the industries of the country and ascertain broadly by a thorough canvass their vital needs, with a view to directing the research work of the country and the coöperative development of the industries through the agency of the technical laboratories both public and private.

6 To organize and support a separate department of the societies' activities, in close coöperation with similar departments of other technical societies, to act, through Engineering Council, or other representative national agency, as a permanent clearing house for all research work.

In support of these resolutions, Mr. Bibbins pointed out that the Chicago local section's desire to ascertain what work could best be done by the local sections within the limited time and facilities at their command, emphasizing the necessity of publicity and that the Chicago Committee believed in the necessity of each state developing itself as a unit (too much could not be done in this direction), encouraging by every possible means every form of intelligent and efficient research, both scientific and also industrial research of a more practical nature. There had been some discussion of the propriety of the state undertaking activities of this character, especially with reference to allying itself thus with industry. But in view of the high moral standing of the engineering profession, it seemed rather far-fetched precaution to deny to the people of the state opportunity of thus developing their industries to the maximum extent.

Following Mr. Bibbins' presentation of the resolution offered by the Chicago Committee, there was a general discussion which centered around the question of governmental supervision of research. Wm. T. Magruder cited the need of state research work in using Ohio coals in gas production to supplement and replace the fast-disappearing natural gas supply; considerable work along this line had been done already in Illinois. The resolution was also discussed by Dean A. A. Potter, who stated that while he believed the Society should encourage research, and while it is unquestionably the purpose of the Society to urge governmental recognition, it would nevertheless be a mistake to approve a resolution asking Congress to appropriate a certain amount of money to support research in certain types of institutions. He believed that THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS should express its approval of all kinds of scientific and industrial research, but should not approve any specific bill.

J. R. Bibbins emphasized the point that the Chicago Committee had no desire to complicate proceedings by attempting to classify institutions; it only favored the broad application of the principle, through publicity and national recognition of research by the passage of supporting legislation. The detail could be worked out by administrative authorities, possibly with the help of Engineering Council.

Doctor Mees, Director of the Eastman Kodak Research Laboratory, stated that he was in accord with Dean Potter's remarks and did not believe in urging Federal research. It was a question whether Federal research was desirable. The leaders of industry were against it, and the advice that he would give to the Society — he was not a member of it — was to pay for its own research.

Several members of the Society who favored financial assistance on the part of the Government next presented their views and urged that the Society endorse any measure which would obtain an appropriation from the National Government for the benefit of such institutions as might be selected.

C. H. Bierbaum also discussed the resolution. A bill of the kind proposed was proper enough from a theoretical point of view, but from a practical point of view he did not believe anything worse could be done than to have Congress pass a bill for Federal research, for there would then be a most inefficient distribution of funds. The securing of funds, however, for doing this work was, he believed, the smallest problem and the securing of men to do the work, the greatest. He was decidedly in favor of private or semi-private research work, but not for Federal control.

Albert Kingsbury urged a careful consideration of the resolution and suggested that the report of the Chicago Committee be brought to the attention of all members of the Society through suitable means of publication.

A substitute motion to the effect that the Society should not favor federal research was not seconded. The resolutions were then voted upon by individual items and carried.

DISCUSSION

Address on Organization of the Division of Engineering of the National Research Council

AT the Research Session of the Spring Meeting, Mr. Galen H. Clevenger, Vice-Chairman of the Division of Engineering of the National Research Council, was invited to give an address on the organization work of the Council and of the Division of Engineering with which he is directly connected at present, in charge of the headquarters of the Division of Engineering in the Engineering Societies Building, New York. Mr. Clevenger said in part:

Before discussing the work of the Engineering Division, it will be well to briefly review the history of the National Academy of Sciences and the National Research Council, pointing out the close relationship which exists between the two bodies. Early in 1863 after the Civil War had been in progress many months and serious and unexpected reverses had occurred, a number of leaders in science and engineering, recognizing clearly the need for a national organization embodying the whole range of science to advise the Government on scientific questions in connection with the conduct of the war, planned the National Academy of Sciences and through their efforts a bill incorporating the Academy was introduced in the Senate on February 21, 1863, and after passing both the Senate and the House was signed by President Lincoln on March 3 of the same year. Later the original bill was amended to remove the limitation on membership and to enable the Academy to receive bequests.

At this time, with but few exceptions, the now well-organized and powerful scientific and technical bureaus of the Government were not in existence, so that the new organization at once became and continued to be of great assistance to the War and Navy Departments throughout the rest of the Civil War.

During the years intervening between the close of the Civil War and the beginning of the war with Germany the Academy has on

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

many occasions advised the Government in scientific matters, the most notable case being perhaps in connection with certain problems arising in the construction of the Panama Canal.

WAR ORGANIZATION OF THE NATIONAL RESEARCH COUNCIL

In April 1916, following the attack upon the *Sussex*, President Wilson called upon the Academy to aid in organizing the scientific and engineering forces of the United States in the defense of the country. The Academy accordingly again turned to the work for which it was originally organized and which it performed so well during the Civil War.

In order to perform most effectively the greatly enlarged service possible under these new conditions, the organizing committee of the Academy after careful consideration recommended that a new body be formed, which would be so organized as to take every advantage of the very liberal charter of the Academy. Such an organization would therefore share fully in all the privileges of the Academy both at home and abroad and would have the further advantage of permitting of the widest range of freedom in the selection of its membership.

In accordance with this recommendation the National Research Council, comprising the chiefs of the technical bureaus of the Army and Navy, the heads of the civilian bureaus of the Government engaged in scientific research and engineering, investigators representing the educational institutions, research foundations, and representatives of industrial engineering research, was formed by the Academy with the active coöperation of the leading scientific and technical societies of the country.

Under date of July 24, 1916, President Wilson addressed a letter to the President of the National Academy of Sciences expressing his approval of the preliminary report covering the newly formed National Research Council and promising his support to the movement.

On February 28, 1917, the Council of National Defense passed a resolution asking the National Research Council to coöperate with it in matters of research for national defense, and soon after the Research Council was requested to act as the department of science and research of the Council of National Defense, its particular function being the organization of investigations on military and technical problems.

In July 1917 the National Research Council was requested by the Chief Signal Officer to organize a Division of Science and Research of the Signal Corps.

The war organization of the National Research Council consisted of eight divisions in addition to the Research Information Service; namely,

1 Division of General Relations
2 Military Division
3 Division of Engineering
4 Division of Physics, Mathematics, Astronomy, and Geophysics
5 Division of Chemistry and Chemical Technology
6 Division of Geology and Geography
7 Division of Medicine and Related Sciences
8 Division of Agriculture, Botany, Forestry, Zoölogy, and Fisheries.

The officers of the Council consisted of a chairman, three vice-chairmen, a treasurer, an executive secretary and two assistant secretaries. The organization of the Divisions differed somewhat, but in general each had a chairman, a vice-chairman and an executive committee.

WAR ORGANIZATION AND WORK OF THE DIVISION OF ENGINEERING

The Division of Engineering at this time comprised four sections: a Section on Metallurgy, a Section on Mechanical Engineering, a Section on Electrical Engineering, and a Section on Prime Movers. The work of each section was under a chairman, who was directly responsible to the chairman of the Division.

The Section on Metallurgy had for its principal work the solving of metallurgical problems arising in connection with the conduct of the war, more particularly those brought to it by the military. This work was accomplished through the medium of committees, whose personnel included leading authorities upon metallurgy.

The Section on Mechanical Engineering established a drafting room in charge of a chief draftsman at Research Council headquarters, and through the generosity of the Carnegie Institute of Technology, a machine shop at Pittsburgh under the direction of a foreman. These were used for the development of inventions referred to the Section by the Divisions of Engineering and Physics.

The Section on Electrical Engineering concentrated its efforts upon the problem of electric welding, more especially electric welding as applied to shipbuilding. This Section worked in very close coöperation with the Emergency Fleet Corporation, who financed its investigative work.

The Section on Prime Movers devoted its attention chiefly to the design and development of power plants for aircraft.

The efforts of each section were so directed as to be of the greatest service in the solving of the problems of greatest immediate need to winning the war; each has to its credit important achievements during the war period. (See Report of the Academy of Sciences for the Year 1918.)

PRESENT ORGANIZATIONS AND AFFILIATIONS

Permanent Organization of the National Research Council. Under date of May 11, 1918, President Wilson issued an executive order asking that the National Research Council be perpetuated, and in accordance with this request the permanent organization of the Council has been rapidly accomplished.

The membership of the Council consists of:

1 Representatives of national scientific and technical societies
2 Representatives of the Government, as provided in the Executive Order
3 Representatives of other research organizations and other persons whose aid may advance the objects of the Council.

The officers of the National Research Council are a chairman, one or more vice-chairmen, a secretary and a treasurer.

The Council is organized in Divisions of two classes:

A Divisions dealing with the more general relations and activities of the Council
B Divisions dealing with related branches of science and technology.

A Divisions of General Relations are:

I Government Division
II Division of Foreign Relations
III Division of States Relations

IV Division of Educational Relations
V Division of Industrial Relations
VI Research Information Service.

B Divisions of Science and Technology are:

VII Division of Physical Sciences
VIII Division of Engineering
IX Division of Chemistry and Chemical Technology
X Division of Geology and Geography
XI Division of Medical Sciences
XII Division of Biology and Agriculture
XIII Division of Anthropology and Psychology.

The affairs of each Division are administered by a chairman, a vice-chairman, and an executive committee, who are elected annually by the Division and confirmed by the Executive Board of the Council.

The purpose of the National Research Council is to promote research in the mathematical, physical, and biological sciences, and in the application of these sciences to engineering, agriculture, medicine, and other useful arts, with the object of increasing knowledge, of strengthening the national defense, and of contributing in other ways to the public welfare.

Affiliation with similar organizations abroad is rapidly bringing about an International Research Council.

The Division of Engineering consists of three representatives of each of the four founder engineering societies, the societies so represented being The American Society of Mechanical Engineers, the American Institute of Electrical Engineers, the American Institute of Mining Engineers, and the American Society of Civil Engineers; further, there is one representative each from the four more important non-founder societies, the societies so represented being the American Society for Testing Materials, the American Society of Illuminating Engineers, the Western Society of Engineers, and the Society of Automotive Engineers. In addition to the representatives of the engineering societies there are twelve members at large, making a total membership in the Division of twenty-eight. Eight members of the Division are also members of The Engineering Foundation.

The work of the Engineering Division has gone steadily forward during the reorganization period and to such an extent that the newly organized Division is now in full operation.

The Engineering Foundation and the Division of Engineering. The Engineering Foundation has from the beginning taken a very active and important part in furthering the work of the whole Council; indeed, in the earlier stages of the war organization, when the funds available for carrying on its work were very limited, The Engineering Foundation gave the services of its secretary and substantially its whole income to the support of the Council, this arrangement continuing until support was secured from other sources.

Recently a plan of close affiliation of The Engineering Foundation and the Division of Engineering has been approved by the members of these bodies and also by the Executive Board of the Council. In compliance with the terms of this agreement The Engineering Foundation has provided the Division of Engineering with an office in the Engineering Societies Building at New York, together with necessary clerical assistance. They further have agreed to make appropriations of their funds to aid specific undertakings of the Division from time to time as may be later determined, and in fact at the present time an arrangement has been effected whereby The Engineering Foundation undertakes the financial support of the work of the Committee on Fatigue Phenomena.

The Division of Engineering is not to be regarded as an instrument of research, but rather as a stimulator and coördinator of research. Its principal object is to get more and better research done in engineering, carefully avoiding the position of being a dictator, or of assuming credit for work which it has encouraged others to do.

The Division of Engineering now has nineteen committees working upon a variety of subjects. These are in various stages of organization. Every effort is being made to take up researches of broad general interest. At present twenty-one states extending from the Atlantic to the Pacific are represented on these committees and the number is rapidly increasing.

The National Research Council, like the National Academy of Sciences, was brought into being during a period of war and although growing out of a war-time need its utility in time of peace is now fully demonstrated. Although it received aid from the Government during the war it is not a Government bureau, and in the future it will be supported by private endowment.

The National Research Council as a federation of research interests of the United States covering the whole field of pure and applied science, and with the effective coöperation possible with foreign investigators through the International Research Council, is in a position through its Division of Engineering to perform a most valuable service in furthering Engineering Research.

No. 1689

THE ORGANIZATION AND CONDUCT OF AN INDUSTRIAL LABORATORY

By A. D. Little, Cambridge, Mass.
Member of the Society

and

H. E. Howe[1], Cambridge, Mass.
Non-Member

During the war industrial research in the United States was naturally stimulated, and as a result there now exists a deeper interest than heretofore in the applications of science to manufacturing processes. New laboratories will undoubtedly be built and many old ones reorganized in order to render more efficient service. It is the purpose of this paper to point out the organization and conduct of such a research laboratory.

The authors first outline the aims of a research organization, following which the divisions of the laboratory are enumerated and discussed, the laboratories of Arthur D. Little, Inc., being taken as a type. The methods of management, writing of reports and the commercial organization of the laboratory are also discussed at some length, and the paper concludes with a description of the building and equipment best suited to carry on this type of work.

PREVIOUS to the war there were about 375 industrial research laboratories in the United States, including those maintained by manufacturers for their own benefit, and commercial laboratories prepared to render similar service, continuously or intermittently to the establishments without such facilities. At the present time there are no figures available regarding the number of new laboratories established as a result of the war, but there is no doubt but that the war created a deeper interest in industrial research, and the application of science to manufacturing processes. It is also evident that those laboratories which existed before the war are displaying a greater interest in fundamental research, and in rehabilitating their organizations are paying far more attention to the research phases of their problems than they have been willing to do

[1] With Arthur D. Little, Inc.

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

For discussion see p. 51.

heretofore. This, then, is an opportune time to discuss the organization and conduct of an industrial laboratory and it is hoped that those interested in the establishment of new laboratories, or the expansion of old ones, may find something in this paper to assist them.

2 In discussing the industrial laboratory we may choose between the one organized for the purpose of exploring some small corner of the broad field symbolic of our ignorance and an establishment concerned with the greatest variety of problems.

3 A laboratory of the latter type should consist of a collection of special laboratories carefully articulated to produce results most efficiently, and the work common to all of them should be organized separately in a large general laboratory.

4 Fortunately for our country there are several such laboratories doing splendid work, and notwithstanding the care exercised to avoid undue specialization, nearly all of them contain departments which dominate, due either to stronger men or the greater appeal which these departments make to the company; or perhaps to a seemingly greater importance of their class of problems at the moment. The great majority of these laboratories are maintained in the plants of industry at an annual expense running up to two millions in at least one case, and with many spending hundreds of thousands each year.

5 Another plan which should be mentioned involves the training of men as a primary consideration, and the Mellon Institute, at Pittsburgh, affords a conspicuously successful example of what may be done in educational institutions in solving the problems of industry, while at the same time men are trained in research.

THE AIMS OF A RESEARCH ORGANIZATION

6 Broadly stated, the aims of a research organization should be:

a To find, develop and train men

b To create such a background in the public mind as shall insure support for research and the industrial utilization of research results

c To secure coöperation between different branches of science, as, for example, between chemists and mathematicians. (The fortuitous combination of the mathematical mind with the viewpoint of the chemist in Willard Gibbs laid the basis for physical chemistry. But such a combination in a single individual is very rare.)

d To avoid repetition and duplication of effort, first by rendering present knowledge readily available to research workers, second by applying clearing-house methods to research projects

e To stimulate research by emphasizing the importance of specific problems, making special grants, rendering material and facilities as generally available as possible

f To furnish a general staff for research which shall work out the plan of attack for major problems, assign the several lines to competent workers and coördinate and focus the whole

g To bring home to manufacturers the advantages of research with the view of promoting the establishment of private, corporation, and group laboratories

h To make and publish a census of available research facilities in men and equipment

i To survey the natural resources of the nation and direct research toward their development

To appraise our great industrial wastes and develop plans and methods for turning them to profitable use

7 As regards any research laboratory, it goes without saying that it is the personal factor which determines performance and this is preëminently true of the laboratory director. Sir Humphrey Davy truly said that his greatest discovery was Michael Faraday, and no greater problem is likely to confront a research laboratory than that involved in the discovery of a director. Successful laboratory directors may be of several types, but a militant optimism, contagious enthusiasm, controlled imagination and quick human sympathy are common to them all. Such a man will naturally in selecting his subordinates look for these personal qualities almost as carefully as he will weigh specialized scientific training, and having been thus guided in his selections will find it relatively easy to inspire throughout his organization those relations of good fellowship and that *esprit de corps* which multiply enormously the effectiveness of any working force.

8 Exceptional men are hard to find simply because they are exceptional, and the director in laying out the work of the laboratory and extending its personnel will endeavor to augment the output of the exceptional man through the coördinated effort of properly directed men of secondary capacity. Fairness in apportioning credit, frequent conferences, and opportunity for self-development are also essential to the attainment of high efficiency.

9 The so-called commercial laboratory devoting its efforts to industrial research and operated on a strictly business basis will best serve our present purpose, and that of Arthur D. Little, Inc., in Cambridge, Mass., will be taken as a type in the belief that much of interest will be found in this establishment, which is "dedicated to industrial progress." During the past thirty-three years this laboratory has grown from a partnership of two chemists to an organization of sixty people, and scheme after scheme has been devised for the management of the enterprise, only to find new conditions and rapid growth calling for constant revision. Being a corporation, it is managed by the usual officers with a board of directors, all of whom do not devote their entire time to the business.

THE DIVISIONS OF THE LABORATORY

10 Within such a laboratory there are two distinct sets of duties which may be designated as scientific or technical, and commercial or financial. These two divisions have at least two points in contact, one being through a service manager, and the other the department charged with obtaining new business for the organization.

11 Considering first the purely scientific side of the commercial laboratory, its fundamental duty is to interpret the results of pure science in the terms of industry. While the work of the commercial laboratory is of the same order as that done in any laboratory even where the dollar is never discussed, it must be conducted with full recognition of the fact that many industrial problems are as intimately concerned with economic questions as with scientific. In other words, while for instance a laboratory process in glass may be intensely interesting and of fundamental importance, the client can hardly be expected to be satisfied with a report unless a commercial method for operating it can be devised. The technical work should be in charge of the president, under whom various departments should be organized, so that each phase of a given problem may have the attention of a specialist, provided with adequate equipment to facilitate the work.

12 In this connection it may be emphasized that it pays to provide congenial, inspiring surroundings for the laboratory worker. The laboratory can be made attractive without being ornate, or involving unreasonable expense, and every effort should be made to have the workers reasonably happy. Under no other condition can the best work be expected, and it must be remembered that

the heaviest investment is in the time of these workers, the salary cost being much greater than that for equipment or material maintenance. Rewards other than monetary for faithful service also play an important part.

13 The departments into which the technical division are divided will naturally differ in each laboratory, but a fairly definite line can be drawn between research, engineering and standardized or routine work. It is advantageous to have all of the standardized work, including that incident to research and engineering, carried on under one department head, for in this way it can be done to better advantage both as regards efficiency and economy. There will be many occasions when individuals from each of the departments will confer on special problems.

14 The research department should be organized for both laboratory and small-factory-scale work. There will be a multiplicity of subjects, and since special facilities cannot be provided in advance of close acquaintance with the problem, the organization of departments for research along special lines will concern personnel more than a division of floor space or equipment.

15 Engineering will embody plant inspection, design, construction and operation, as much of its work will be in the field, although many phases of its problems will be worked upon concurrently in the laboratory by those departments best suited to handle the work.

16 The analytical department will be subdivided under such headings as textiles, fuels, food, metallurgy, and metallography, chemical microscopy, water, lubricants, construction materials, pulp and paper, fermentology, etc. Some of these subjects will require special accommodations, while others can share a large laboratory which provides space for certain apparatus kept in place for a large number of similar determinations.

17 Nothing is more expensive or demoralizing than experimentation in the plant. An industrial research laboratory should, therefore, be adequately provided with equipment of semi-commercial size. Infant mortality among processes is high in any case and the most critical period in their young lives is that covering the transition from the laboratory to the plant. They require and the research laboratory should provide a nursery to protect and foster them during this period of their development. Some large manufacturers have even found it desirable to operate in connection with and under the sole direction of their research laboratory a small plant in which

actual commercial manufacture is regularly conducted. Such extension of the laboratory's function permits the complete reduction to practice of new methods and the commercial demonstration of the sufficiency of the product before the innovations are introduced into the main plant.

18 Even when no such provision appears feasible, it is, nevertheless, highly desirable to have the industrial research laboratory actually engaged in some small scale, highly specialized, commercial manufacture, preferably of some product which it has itself originated. The least advantage of this procedure is that such manufacture of a properly selected product may frequently defray a substantial proportion of the expenses of the laboratory. The major benefits are the acquirement of a certain commercial sense by the laboratory staff, an appreciation of the conditions and difficulties of actual production, and finally the strengthening of the position of thé laboratory through the increase in its turnover and equipment. Such procedure, while perhaps not general, has been followed to great advantage by the research laboratories of the General Electric Company, the Eastman Kodak Company and Arthur D. Little, Inc.

19 It is easy to visualize the organization chart for such a laboratory, and a brief description of how a new piece of work will be handled may convey a still better idea of the method of management.

THE METHOD OF MANAGEMENT

20 The authorization for the work will go to the service manager, who sees all incoming mail, and to the authorization will be attached any correspondence or data bearing on the case, all of which will be given a case number for identification, and this number will be entered in a case register, which will indicate the name of the client, the subject of the problem, the date the authorization is received and the date when the work shall have been completed. The service manager, who must be familiar with the ability of each member of the staff, as well as with the work in hand, will assign the case to the division which can render the best service. Conferences will then be called, into which any member of the organization who can contribute anything to the solution of the problem in hand will be drawn, and outside associates or independent consultants may be included. The problem will then go into work by means of instruction sheets, setting forth what is to be accomplished, suggesting

methods of attack, relating any special circumstances, references to literature, and standard methods which may be applicable, and as much light as possible given to the individual who is to do the work. Accompanying the case there will be a tag bearing the case number, upon which a date at which it is expected the work can be completed, or a progress report made, must be indicated. The tag is then returned to the service manager. Through the means of data sheets, time slips and verbal reports the progress of the problem will be readily followed. This procedure will be followed in all the divisions, the individual reporting to his superior, and the service manager will be alert to insure prompt and efficient service to all clients.

21 At the completion of the work the report to the client varying in extent from a single printed form, upon which the results of analysis may be set down, to a bound volume of several hundred pages, will pass through the hands of all concerned, and will thus be distinctly the report of the organization and not of an individual in the organization.

REPORTS

22 Report writing requires skill, for it must be comprehensive and complete without padding. It should begin with a clear statement of the problem, followed by the conclusion reached as a result of the work, which may then be described in detail. Patents, cost data, tables, graphs, photographs and samples should be dealt with in an appendix, and in some instances descriptions of apparatus should be included. The whole must be carefully indexed, and a copy sent to the library to be bound and kept as confidential information in locked cases, but as part of the library it should be carded for the library card index. Obviously no fast rule can be laid down for writing reports, but it should be borne in mind that many of those who read technical reports are not interested in minute details, and that the subject-matter must be presented in a form that will be interesting and understood by the layman. It must also have its important points so emphasized that they can be readily picked out by those not caring to read the entire report, but at the same time it should include sufficient data to serve the purpose of a fully qualified technical man to whom the report may be referred at some later time.

23 This brings up the question of the library, which may easily be considered the backbone of the industrial laboratory. Its extent

will depend upon other library material available in the community, but there are few things which obstruct research more seriously than the absence of easily accessible proper library facilities. A few dollars spent in books and literature frequently saves as many hundreds otherwise spent in work of duplication. The useful periodicals must be provided, elaborate indexes will be found a good investment, also abstracts and patents; in short, every means for quickly locating literature references should be at hand. The current literature, with articles of interest indicated on an attached slip, should be circulated among the members of the staff whose names are checked on this slip, and some one, preferably a chemist, should have assigned to him the task of constantly reading the literature in order that no scrap of information shall escape. Such a chemist-librarian will conduct searches in other libraries, prepare abstracts, and in fact direct the information service for the laboratory, and through the laboratory to its clients. This will frequently mean recasting a scientific article in order to make it of practical value to the works manager or superintendent.

THE COMMERCIAL ORGANIZATION

24 Concerning the commercial division of the organization, the case we have described will have originated through the recommendation of a satisfied client, through magazine advertising, the literature of the laboratory, lectures and informal talks, direct appeals, traveling representatives, or through some similar channel. The publicity of the laboratory will be in the hands of the commercial department responsible for securing new business, and answerable to the president so far as technical matters are concerned, and to the treasurer on the financial phases of a prospective problem. The commercial department will answer inquiries, will present attractive problems for industrial research, and endeavor to obtain authorization from manufacturers for work contemplated along lines which are frequently made the subject of a conference. When the authorization is secured through personal interviews, by correspondence, and frequently by telegraph, the commercial division records become a part of the case, and nothing more is to be done by the division until after the final report. Then it is quite reasonable to make work satisfactorily completed the best recommendation for attack upon some new problem.

25 The commercial department will maintain a carefully classi-

fied mailing list, a follow-up system on prospects, and its own letter files. The treasurer, who has at hand data as to the cost of each man's work, will be consulted before any proposals involving money are made, and the financial division, under the treasurer, will attend to all matters of time cost on work, determine the overhead, the items of maintenance, depreciation, etc., all of which go into the total cost of doing business, something of which the average industrial laboratory remains in almost complete ignorance. By means of records constantly kept up to date, and revised in the light of experience, the treasurer will know accurately the percentage which must be added to the salary of each individual in order to arrive at the cost of his work, and will soon be able to rate the men according to their capacity and estimate something of what a particular undertaking should cost. This will be of great assistance in making suggestions regarding appropriations for prospective work, and also which members of the staff are of the greatest service to the organization. Costs will be determined with the help of time slips, which each individual will submit daily, indicating the client's name, the case number, the amount of time spent and how it was spent, that is, whether in the laboratory, in the library, traveling, etc. Another function of the treasurer's division will be the management of industrial enterprises where such service is desired, together with technical supervision or assistance.

BUILDINGS AND EQUIPMENT

26 With this general plan of management before us, we may now consider the equipment and space required for effective work. Experience has shown that a satisfactory building is one approximately 50 ft. by 150 ft. (planned so that another building may be easily connected in H-formation) and consisting of a basement, with three floors, the basement itself being so designed that it is as light and airy as the upper floors. Such a building with a wing housing the power plant and small grinding rooms (this being the connected wing of the H) has a total floor space of approximately 30,000 sq. ft. The building occupied by Arthur D. Little, Inc., is of this type and the various departments are allotted the following space: Analytical division, 5328 sq. ft.; research division, 3458 sq. ft.; engineering, 1155 sq. ft.; commercial department, 768 sq. ft.; management (meaning the management of outside enterprises), 538

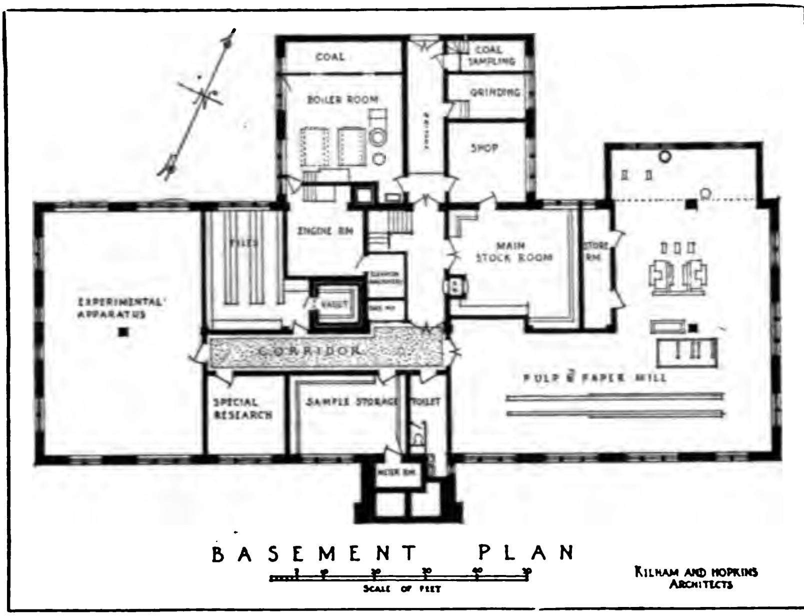
sq. ft.; special department devoted to pulp and paper, 4118 sq. ft. The portions of the building which are non-producing, such as miscellaneous offices, which includes stairs, corridors, halls, lavatories, etc., comprise 8000 sq. ft., and the space unassigned amounts to 3815 sq. ft. This includes laboratory space provided for emergencies and expansion but not in constant use.

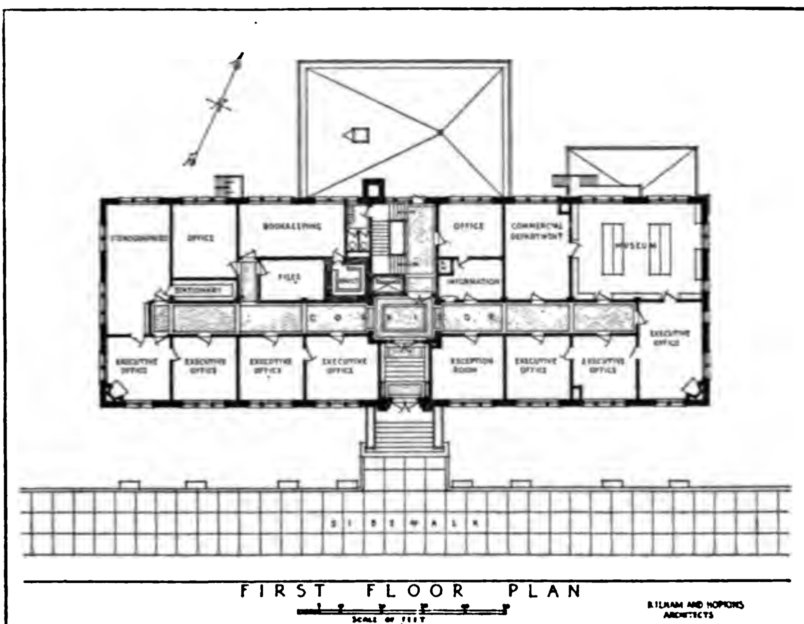
27 These area measurements form the basis of apportioning overhead due to interest on investment, depreciation, repairs, insurance, upkeep, etc., and among the departments is distributed the charge for carrying 768 sq. ft. devoted to an industrial museum and 1536 sq. ft. devoted to the library.

28 The basement will provide room for the power plant, the current sample room, the general stock room, two very large rooms for small-factory-scale equipment, two small rooms for coal and other crushing and grinding operations, and a machine shop in which the physical testing machinery can be installed. A laboratory for testing construction materials, such as cement, may also be placed in the basement to advantage, and here, too, a room and vault can be set aside for inactive letter files and records.

29 The first floor will provide a series of offices each of about 250 sq. ft. area, two larger ones which may be used by officers of the company, consulting engineers and others, a reception room, and an information booth with switchboard. The museum can also be on the first floor, together with the rooms devoted to the commercial department's work. Quarters for the financial division, with ample vault space, and room for current correspondence and the general stenographers' office complete the floor. The engineering division might also occupy rooms on the first floor.

30 The second floor may be properly devoted to research and the library. It is frequently advantageous to be able to segregate research problems, and it will be well to provide a series of small rooms, say of approximately 250 sq. ft. in area, which can be fitted up in accordance with the requirements of the problem, and easily dismantled at the conclusion of the work, to be refitted according to the next undertaking. A branch stock room should be located on the second as well as the third floor, and these should be served by elevator from the general storeroom in the basement. There are always a number of scattering problems in research that can be handled in one laboratory, and so a special-problems laboratory should also be provided on the second floor with an office for consultation purposes. Finally a large room, say of 1500 sq. ft., should





BASEMENT AND FIRST-FLOOR PLANS OF THE LABORATORY OF ARTHUR D. LITTLE, INC.

also be available for large undertakings, and as a space for emergency overflow.

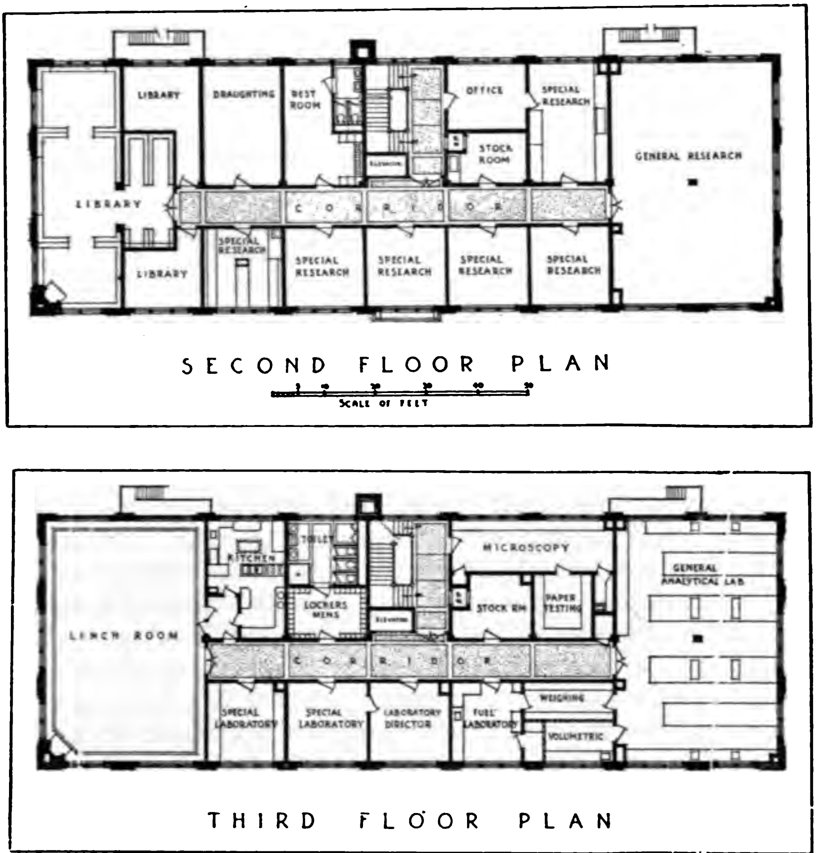
31 The third floor may comprise the general analytical laboratory, a room of about 1500 sq. ft., adjacent to which should be a room for titration and the balance room (each of approximately 125 sq. ft.), a fuel-testing room and a special room for extraction. A branch stock room and the offices of the head chemist and assistants should also be on this floor. The optical room should be placed where north light can be obtained, and a small dark room must be provided, as well as a specially equipped room for the physical testing of paper and textiles. The kitchen can adjoin the assembly room. A locker room for the men and a rest room for the women must also be provided. Space under the roof can be used for fans and a ventilating appliance, water tanks, etc.

32 Such a building as has been thus briefly described cost about $200,000, 1917, and $50,000 will provide general equipment. There is always something new to buy for a laboratory. Such an establishment will provide working space for approximately 150 people, and more could be accommodated if necessary, depending largely on the type of work being conducted. The cost of operation and maintenance, based on a staff of sixty, will be about $20,000 per month.

33 While the type of building described may not appeal in all cases, it is certainly true that better work is done in a separate building provided for the purpose than if an attempt is made to remodel an old building or provide space in some existing building. Laboratories started in this way are certain to develop, if at all successful, and moving is very expensive, principally because it interferes with the efficiency and production of the establishment, which is a direct loss due to the high salary cost. The permanent investment in a laboratory building and equipment is smaller than the salary investment, and any enterprise of this sort should be planned with reference to the fact that it is the men's time and accumulated knowledge that form the stock in trade.

34 Not only does the success of an industrial research laboratory depend upon its equipment and environment, but far more defininitely upon the capacity and ability of the director, and his immediate associates. The constant problem is to obtain hearty coöperation without over-organization and without in any way dampening the enthusiasm of the individual. After all, the greatest progress in science has been made through individual

effort, and it is the function of the industrial research laboratory and its organization to provide and maintain conditions under



SECOND- AND THIRD-FLOOR PLANS OF THE LABORATORY OF ARTHUR D. LITTLE, INC.

which such individual effort will achieve the greatest results, utilizing coöperation as far as circumstances in each instance may warrant.

No. 1690

INDUSTRIAL RESEARCH LABORATORY ORGANIZATION

By C. E. K. Mees,[1] Rochester, N. Y.
Non-Member

The great value of scientific research, both to the industries and the nation at large, is now generally recognized. The industrial research laboratory is an important factor in maintaining the supremacy of an industry, and its success depends to a considerable degree upon its relation to the other departments of the company with which it is associated. In this paper these statements are discussed, and the author presents his views regarding the establishment and function of industrial laboratories, giving in connection with the latter three annular diagrams. The form and operating costs of industrial laboratories are also discussed.

THE triumphs which have already been won by research laboratories are common knowledge. The incandescent-lamp industry, for instance, originated in the United States with the carbon-filament lamp, but was nearly lost to this country when the tungsten-filament was developed, only to be rescued from that danger by the research laboratory of the General Electric Company, who fought for the prize in sight and developed first the drawn-wire filament and then the nitrogen lamp; and we may be sure that if the theoretical and practical work of the research laboratory of the General Electric Company were not kept up, the American manufacturers could by no means rest secure in their industry, as, undoubtedly, later developments in electric lighting will come and the industry might be transferred, in part if not completely, to the originators of any improvement. Manufacturing concerns and especially the powerful, well-organized companies who are the leaders of industry can, of course, retain their leadership for a number of years against smaller and less completely organized competitors, but eventually they can insure their position only by having in their employ men who are competent to keep in touch with and

[1] Director of the Research Laboratory of Eastman Kodak Company, Rochester, N. Y.

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

For discussion see p. 51.

to advance the subject, and the maintenance of a laboratory staffed by such men is a final insurance against eventual loss of the control of its industry by any concern.

2 The success of an industrial research laboratory depends to a considerable extent upon its position in the organization and upon its relation to the other departments of the company with which it is associated.

3 If industrial enterprises had been organized afresh with the research laboratory as a definite part of the organization, it is probable that by this time some general opinion would have been formed as to the position which the laboratory should occupy, but in fact nearly all industrial research laboratories have been added to organizations already formed, and their relations to the other departments of the organization are usually closely associated with their origin.

METHODS OF ESTABLISHING LABORATORIES

4 Laboratories are established in many different ways. If there is a technical scientific expert in the executive of the manufacturing company he may feel the need of a laboratory and become its director, and in this case the laboratory will necessarily be very closely associated with the work of the executive who initiated it.

5 A laboratory may also be established under a separate director, not himself associated with the executive officers of the company, but as a reference department for the executives, and in this case also it will be very closely associated with the officers of the company and will tend to be concerned more with questions of policy and the introduction of new products than with any other of the problems of the company.

6 In a large company a research laboratory may be established as a separate department, having its own organization and being available as a reference department for all sections of the company, in which case its activities will cover a very wide field, but at the same time it will not have as direct an influence upon the policy of the company as will happen if it is closely associated with one or more of the executive officers.

7 The earliest research laboratories grew out of the works testing and control laboratories and were therefore responsible directly to the works manager. More recently, laboratories have generally been established as independent departments of the company and responsible only to the general manager.

8 The executive official to whom the laboratory should report will depend upon the nature of the work to be done. There may be industries in which research work is required for only a single department, and in this case the research workers should be responsible to the head of that department; there are others in which the interest in the research is confined to the works, and in such cases the laboratory should be responsible to the works manager; but in most technical industries research work will have a great bearing not only on the methods of production but even on the general policy of the industry, and in such cases it is necessary that those who direct research should be in touch with, and responsible to, the executives who control policy.

9 The position of the research laboratory in an industrial organization is perhaps best determined by the criterion that the research department should be responsible to the officer of the company who is in charge of the development of new products. If the introduction of new products is in the hands of the works organization, then the research department should be responsible to the works manager; if there is a definite development department, or, if new products are introduced through the agency of some definite executive, then it is to that executive that the research department should be responsible. The research laboratory, in fact, should primarily be associated with development.

FUNCTIONS OF THE LABORATORY

10 The chief functions of the laboratory are as follows:

1 The provision of information regarding the technical and scientific matter in which the industry is interested, and the supply of this information in a form suitable for the education of the employees, of the customers, and of the general public.

2 Service in the form of the provision of specifications and standards for materials, the making of analyses and tests, assistance to the works in regard to difficulties and to customers in the relation of problems arising from the use of the product.

3 The development of new processes or products, utilization of by-products, the development of new departments of the industry.

11 These functions may be expressed by means of annular diagrams such as those shown in Figs. 1, 2 and 3. Fig. 1 shows

the information diagram. The information originating from the library and from the research staff is issued in the form of abstracts, reports, scientific publications and monographs, and first goes to the executive, manufacturing, purchasing and sales departments for their information; then to the advertising and educational departments to be placed in the form required for publication in the scientific and general press.

12 The organization of the development work is shown in Fig. 2, where the work is shown to be founded upon pure research done in the scientific department, which undertakes the necessary

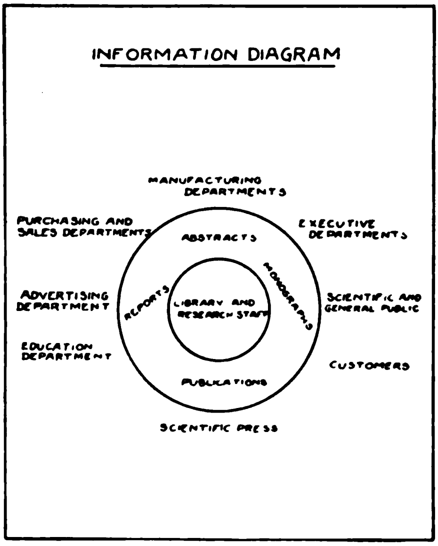


FIG. 1 INFORMATION DIAGRAM, FUNCTIONS OF AN INDUSTRIAL RESEARCH LABORATORY

practical research on new products or processes as long as they are on the laboratory scale, and then transfers the work to special development departments to form an intermediate stage between the laboratory and the manufacturing departments. These development departments are really small scale manufacturing departments which may be operated either by the works department or by the laboratory; but which are controlled, as regards the work done in them and the method used, by the laboratory itself, being run as experimental departments in order to develop a new process or product to the stage where it is ready for large scale manufacture.

13 In Fig. 3 the service diagram shows the scientific divisions as the operating centers, each of which supports and controls the necessary service departments, which prepare specifications and standards, undertake testing and analysis, the investigation of works troubles, complaints of customers, and suggestions from the sales department, the results of which are communicated to the departments interested.

14 The laboratory organization will therefore consist of a section of administration which will be responsible for the direction

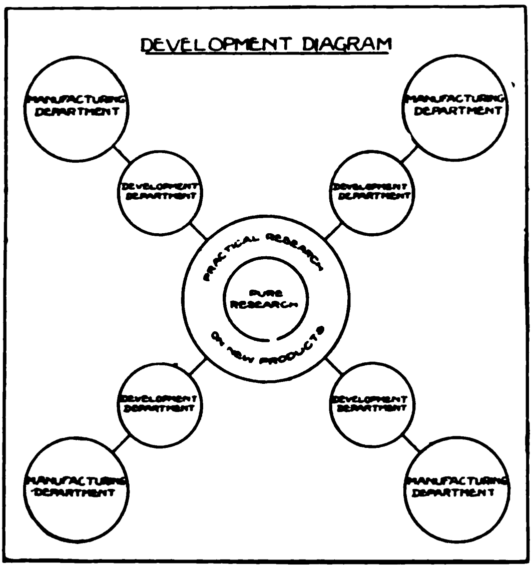


FIG. 2 DEVELOPMENT DIAGRAM, FUNCTIONS OF AN INDUSTRIAL RESEARCH LABORATORY

of the work, the control of accounts and the issuing of reports; a section of information, which will operate the library, prepare abstracts of the literature, keep in relation with the patent department and constitute its technical wing, and prepare reports and publications of all kinds; and the scientific section, which will carry on the operation of the laboratory work.

FORMS OF ORGANIZATION

15 There are two forms of organization possible for the scientific work of the laboratory. For brevity these may be spoken of as the "divisional" system and the "cell" system. In the divisional

system the organization is that familiar to most businesses. The work of the laboratory is classified into several divisions; physics, chemistry, engineering, and so on, according to the number necessary to cover the field, and each of these divisions has a man of suitable scientific attainments in charge of it. In a large division each of these men will in turn have assistants responsible for sections of the division, all the heads of divisions finally being responsible to the director of the laboratory.

16 Under the alternative or cell system the laboratory consists of a number of investigators of approximately equal standing in the

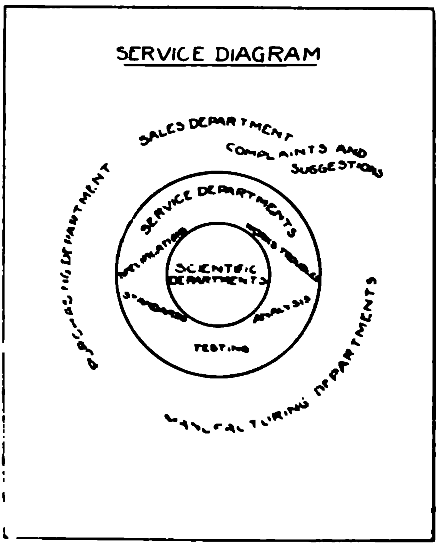


Fig. 3 Service Diagram. Functions of an Industrial Research Laboratory

laboratory, each of them responsible only to the director, and each of them engaged upon some specific research. Each such investigator of course may be provided with assistants as may be necessary. [illegible] these two systems of [illegible] any laboratory. It is [illegible] the other hand [illegible] effective.

[illegible] is best directed [illegible] of all the men [illegible] have the further [illegible] of the details of the direction of the scientific work, it will keep him constantly informed as to all the work going on in the laboratory.

COST OF RESEARCH LABORATORY

19 From various sources, but chiefly from the convenient list of American laboratories given by Fleming, there can be found the cost of a research laboratory per scientific worker employed, at the time when that list was prepared. It might seem that there would be very great variations in this, but, provided that the laboratories are all of the physical and chemical type, there is a surprising agreement between the figures, which show that the cost of building and

FIG. 4 THE LABORATORY OF THE EASTMAN KODAK COMPANY

equipment for a laboratory was then between $3000 and $4000 per man. From the same sources the annual cost of maintenance of such a research laboratory appears to be slightly lower than the first cost. Probably $3500 per man is a fair estimate of the cost of maintenance, and of this we may take 60 per cent as representing salaries and wages and the remainder all other expenses. These figures must, of course, now be increased in proportion to the rise in costs.

20 As laboratories are organized and experience gained in the types of laboratory suitable for different industries it will doubtless be possible to lay out definite schemes of organization for a laboratory suitable for the requirements of any industrial under-

taking. At the present time, however, it is possible to do this only in the most general way, and it is necessary to consider each case independently, taking into consideration the requirements of the particular industry involved.

21 Fortunately data on this subject are being accumulated at the present time and the whole question of laboratory organization is being studied carefully both here and abroad. In this country the National Research Council has appointed a special committee to promote industrial scientific research and this body is now engaged in drawing up a scheme for the establishment of an alloys research laboratory and is considering other lines of industrial research all of which will surely meet with the approval and support of The American Society of Mechanical Engineers.

No. 1691

RESEARCH WORK ON MALLEABLE IRON

By Enrique Touceda[1], Albany, N.Y.
Non-Member

The author presents an account of four years of research work undertaken for the American Malleable Castings Association as a plea for industrial research among manufacturers and as a striking example of what such research can accomplish. He sketches the organization and purpose of the Association and shows how the quality of the product of its members has steadily increased since the beginning of the research work. Malleable-iron castings, due to lack of uniformity and dependability, were rapidly being replaced by other materials. There were many fallacious ideas and theories regarding the physical properties of such castings and the methods of annealing them. Records of tests of 1*-in. bars from seven different concerns made by the author in* 1911 *showed that the average ultimate strength was* 39,882 *lb. and the elongation under* 5 *per cent. A report dated March,* 1919, *to the members of the Association, each of whom regularly submits test bars from some one heat of each day's runs, showed that* 44 *per cent of the test bars submitted during that month had an ultimate strength over* 52,000 *lb. and an elongation of* 14.67 *per cent, indicating the progress made since research work was undertaken.*

It is further stated that the average of test bars of the Association from January 1, 1917, *to March* 31, 1919, *has shown an ultimate strength of* 51,000 *lb. and an elongation* 12.5 *per cent. The records of tests show, contrary to generally accepted theory, that the elongation increases with the ultimate strength. The purpose of the Association, however, is not to increase ultimate strength and elongation but to increase the uniformity of a product upon which the engineer can rely, and this is being accomplished through exhaustive research and advice to members through the consulting engineer of the Association.*

A description is given of the process of manufacturing malleable iron, of the air furnace, and of the annealing ovens and the annealing process. The structures of iron containing free carbon and iron containing combined carbon are shown by micrographs and the metallurgy of cast iron is carefully explained with abundant micrographs of typical structures.

The effects of the time element in cooling through the critical temperature, of successive anneals, of varying percentages of carbon, sulphur, silicon, phosphorus and manganese and of subsequent heating to high temperatures are clearly described and illustrated. Picture-frame fractures are also discussed.

[1] Consulting Engineer, Albany, N. Y.

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers, 29 West 39th Street, New York.

The author closes by exploding three popular theories with regard to malleable iron. He shows that the strength of malleable iron is not confined to its skin but that this may be machined off without destroying strength, if the quality of the iron is as it should be. Secondly, he shows that it is possible to eliminate the carbon throughout the entire specimen and not merely near the surface. Lastly, he shows that thick as well as thin castings can be annealed successfully.

IT is the writer's belief that three factors mainly operate to darken the vision of many of the manufacturers regarding the value of research work. The first is fundamental. Stated brutally, it is ignorance as to its possibilities. Following this regrettable fact is their obsession to the thought that the prime and essential requisite for success lies in the advertisement, with oftentimes the accent on the last four words. The third factor has reference to the extent to which research has been discredited through the employment for that purpose of men not qualified through temperament, proper training or resourcefulness, to undertake such work — the square peg in the round hole, the neophyte, and at times the quack. While the revelations born of the world's war have in large measure done much to open the drowsy eyes of the manufacturer, it is nevertheless certain that hard and diligent work still remains to be done before the majority will be aroused from their lethargy.

CONDITION OF THE INDUSTRY PRIOR TO RESEARCH

2 Prior to the time four years ago that research work was undertaken in the interests of the American Malleable Castings Association, the industry was in a more or less chaotic condition. There had been at least three years of serious business depression, ruinous competition as a consequence had been running its insensate course, but back of and beyond all this, was the damning accusation of the engineer, that the material, except in the case of a limited number of concerns, was not only lacking in dependability, but of low strength when dependable. In railway-car fabrication particularly, the number of malleable-iron castings used had dwindled from a very large quantity per car, to an almost insignificant number consisting mainly of unimportant details. Malleable cast iron was rapidly being replaced by the steel casting, and in other directions as well, the latter was encroaching on the legitimate field of the former, and incidentally placing it in an exaggeratedly false position, for the reason that when substitutions were made the patterns were redesigned and made much heavier to accommodate the less-fluid casting properties of that metal. When a steel casting failed, the

this product conclusions drawn from experiments made on white-heart malleable iron, such as used on the other side.

FORMATION OF THE ASSOCIATION

6 Among the various manufacturers of malleable-iron castings, at the time referred to, were some twenty-five who were progressive enough to understand the benefits that accompany coöperation, and that one might better have an intimate friend with whom to compete than an enemy. These had formed an association that had been in existence some ten years or more. The association had as its objectives the exchange of ideas in the direction of business economy, improved works practice, the study of proposed foundry and factory enactments, the securing of more favorable insurance rates, the study of problems relating to cost, labor, housing and sanitation, and finally the forming of friendships that are the natural outcome of frequent and close personal contact. It is to these men largely that credit should be given for the renaissance of the industry. They decided to enlarge their field of action and, irrespective of cost, determined to go the full road along the lines of metallurgical research. They determined as well that every statement made as to progress would be conservative and accompanied by data that would be incontrovertible, which course they have followed to the letter.

7 The preliminary part of this paper will be descriptive of the steps that were taken to bring system out of chaos, the second will deal with some metallurgical details of the malleable-iron process, and the third with the fallacies that have been handed down in the manner already indicated. The first step taken towards systematization was to make a hasty survey of the different plants. At the conclusion of the survey, papers were written by the consulting engineer in connection with matters deemed of most importance at the time to the membership, and in these recommendations were made as to suggested improvements in works practice generally. These papers were sent to the secretary, who had them printed in uniform bulletin form for distribution. In these bulletins to date a very wide field has been covered, as they contain quite complete details in regard to physical properties and tests, the metallurgy of the process, air-furnace and annealing-oven construction and practice, combustion, special investigations of difficulties encountered from time to time by the various members, and other matters too numerous to mention.

METHOD OF PHYSICAL TESTING

8 When the research work was started, it was found that by far the majority of the members had no system of testing the quality of their product, aside from the twisting and bending of a casting, in order to ascertain its ductility, or the bending over of test lugs attached to castings. Consequently there was no way in which could be compared the quality of product of one member with that of another. To be candid, there was available no information of value that could be given to an engineer who might be seeking information of this character. Mr. Benjamin Walker, who at the time was vice-president and general manager of the Erie Malleable Iron Company, of Erie, Pa., a number of years previous to this, had devised for the purpose of testing the quality and uniformity of his product, what he called a test wedge. The wedge was 6 in. long by 1 in. wide, tapering from $\frac{1}{2}$ in. at its base to $\frac{1}{16}$ in. at its other end. His practice was to distribute a series of these wedges throughout the annealing oven, and at the conclusion of the anneal subject them to test — accomplished by holding the butt end of a wedge in a narrow-jawed tongs, placing it upright on an anvil, and striking its top end with a 6- or 8-lb. sledge. In this manner the thin end of the wedge would gradually curl up under these repeated blows, and it is apparent that the more blows the wedge would stand before fracture, the shorter would be the butt left in the jaws. The shortness of this butt he considered was a measure of the metal's ductility.

9 It was decided that an attempt would be made to standardize this test for adoption by the Association. With this end in view a machine (Fig. 1) was so designed that a weight of 21 lb. when raised to a height of 3.33 ft. above the top of the wedge, when placed in position for test on the anvil of the machine, would be automatically tripped to deliver a fairly constant blow on the thin end of the wedge. The blows were counted, and for convenience the number delivered before rupture took place, was recorded as the *blow efficiency* (assumed to be a measure of toughness), while the length of the butt (assumed to be a measure of ductility) was measured and expressed in terms of *butt efficiency*. These were arbitrary terms, understood by the members, and intended for their use only. It will be noted that the test is one of great severity, for the reason that the first blow must be borne by a section but slightly larger than 1 in. by $\frac{1}{16}$ in., and equals, expressed in pounds of static pressure, over 1350 lb. As the wedge curls and consequently shortens, subsequent blows

11 In this manner it is possible to learn just how each member is progressing, and if it is considered by the Research Committee that his progress has not been as rapid as it should have been, the consulting engineer is requested to pay him a visit with the object of aiding him more quickly to better his condition.

AVERAGE PHYSICAL PROPERTIES PRIOR TO RESEARCH

12 It is pertinent at this point to present in as fair and impartial a manner as possible a comparison of the physical properties of malleable iron manufactured within the past few years with that made in the period prior to 1913. Before entering into this matter the writer wishes to go on record as stating that any data in connection with the ultimate strength of malleable iron is valueless as far as serving to show its real worth, unless accompanied by information regarding the ductility of the metal as measured by the elongation. If one knows how, there is no difficulty whatever in uniformly making a metal of 85,000 lb. ultimate strength, provided ductility be sacrificed down to what would be represented by a 5 per cent elongation. Further on the structure of such material is shown. In the past there have been occasional records of tests that have run somewhat similar to these figures, and such tests have been quoted by others in different articles, but in the light of present practice this is not considered good malleable iron, unless it is to be used for special purposes where ductility is not of consequence and a high ultimate strength is imperative. When such high ultimates were obtained it must obviously have been by accident rather than design, because if 5 per cent elongation was a rather high average at the time, it follows that it would have been accompanied by a high ultimate strength had it been known how to obtain it. There are still stronger reasons for making this statement. For many years the writer has had an unusual opportunity to learn either at first hand or on good authority what character of product most of the concerns made, and aside from one concern who had always enjoyed an enviable reputation for the uniformity and excellence of its product and another very large company whose plants were very painstaking in their methods of manufacture, the ordinary run of malleable iron was undoubtedly inferior. The record of tests made in this laboratory during the period mentioned do not exceed three hundred. They do, however, represent the product of many different concerns. As some of the bars were of square section,

some rectangular, and others round, it is plain that no uniformity existed in their dimensions. The latter vary anywhere from ½ to 1 in. in diameter, while the former for the most part are 1 in. square and 1 in. by ½ in. The great majority of these tests show that the ultimate strength was under 39,000 lb. per sq. in., while the elongations were for the most part under 3.5 per cent. There are instances of fairly high strength, slightly over 48,000 lb., while the highest elongations, ran 7 per cent in 4 in.

13 It happens that the writer has a record of tests made in 1911 on bars made by seven concerns that were deemed at the time to be unquestionably among the very best producers of malleable-iron castings. These founders were each asked to make 20 of the very best bars they could produce for test, 10 to be 1 in. in diameter, and 10 to be ½ in. in diameter. In these tests the average ultimate strength of the 70 bars of 1 in. diameter is 39,882 lb., and the average elongation exactly 5 per cent. The lowest ultimate is 31,990 lb. and the highest 45,560 lb., the lowest elongation 1.7 and the highest 9.8 per cent. In the ½-in. bars, the average ultimate is 41,693 lb. and the average elongation is 5.5 per cent. The lowest ultimate is 33,600 lb. and the highest 47,430 lb., lowest elongation 1.2 and highest 6.3 per cent. Inasmuch as each of these seven concerns were informed that what was wanted was 20 test bars that would represent the very best product they could make, and inasmuch as these manufacturers were considered among the best of the producers, it would appear that the writer is warranted in assuming that the foregoing tests would represent a high rather than a low average.

14 Table 1 is reproduced from Dr. H. M. Howe's book, The Metallography of Steel and Cast Iron, pages 96 and 97. In the first series of this table there is a very good bar, though the writer is inclined to doubt the accuracy of the figures under the elastic limit column. In the second there is a bar, slightly better than the other. The one with 8.2 per cent elongation is also good. The second series furnishes a guide as to what was deemed to be unusually good malleable iron several years ago.

THEORIES OF MALLEABLE IRON

15 In Hatfield's Cast Iron in the Light of Recent Research, 1912, page 213, is a table containing the results of 14 tensile tests of bars 1 in. by ⅜ in. in section. The bars run very uniformly and the material, while of low strength, is ductile. The latter should

certainly be expected in bars that are but ⅜ in. thick. The lowest ultimate strength is 42,750 lb. and the highest 51,200 lb., the lowest elongation 10 per cent and the highest 15.3 per cent. Under this table appears the following: "If attempts are made to increase the maximum stress obtained from such iron, the elongation would appear to have to be sacrificed, and to a considerable degree. In illustration of this the following records are given."

Maximum stress, tons per sq. in.	Elongation, per cent in 2 in.	Reduction of area, per cent
18.9	12	9
21.1	13	8
24.0	9	8
26.5	7	5
27.8	5.5	3.5
29.0	5.0	4.2
34.3	3.0	2.5

16 Under this table the following words appear: "The cause of the increase in tonnage is the retention of increasing proportions of combined carbon. This combined carbon stiffens the material and incidentally reduces its ductility." The writer believes that he was the first to prove and furnish indisputable evidence as to the falsity of the statement that the ductility of malleable iron decreased as its ultimate strength increased. While Hatfield is correct in assuming that the ultimate strength is increased and the ductility decreased with an increase in combined carbon, the statements indicate clearly that he was, at the time at least, not familiar with the manner in which malleable iron can be obtained having the characteristics of high strength accompanied by high ductility. It can be stated, that in normal malleable iron as made today the higher the strength the higher will be the ductility, and in this particular this metal is unique. In a paper read by Dr. Richard Moldenke, before the American Foundrymen's Association in 1903 can be found these words: "The tensile strength of malleable castings should run between 42,000 and 47,000 lb. per sq. in.; castings showing only 35,000 lb. are serviceable for ordinary work. It is *not advisable to run beyond* 54,000 lb. per sq. in. *for the resilience is reduced,* and one of the valuable properties of the malleable casting impaired. The elongation of a piece of good malleable will lie between 2½ and 5½ per cent." In the Mechanical Engineers' Handbook, edited by

Prof. Lionel S. Marks and published as late as 1916 appears the following by the same author: "These castings *should not be machined*, as the interior is not as strong as the metal at and near the surface. Tensile strength 35,000 to 48,000 lb. per sq. in. European malleable cast iron, made by a somewhat different process, is not as sensitive to machining, the castings which are thin only, are practically

TABLE 1 PROPERTIES OF MALLEABLE CAST IRON (H. M. HOWE)

		Tensile strength, lb. per sq. in.	Yield point, lb. per sq. in.	Elongation		Contraction of area, per cent
				per cent	inches	
Malleable castings. Standard Properties						
E. Schoemann. Open-hearth furnace.		44,230	.....	3.9	.....	.....
		48,640	.....	4.5	.....	.....
W. P. Putman		39,638	28,326	7.03	2	10.51
		50,849	41,792	10.15	.....	20.92
Kent, Master Car Builders' Association, 1891 Size of specimen	1.52 × 0.25 in.	34,700	21,100	2.0	4	....
	2.0 × 0.78 in.	25,100	15,400	1.5	4	....
	1.54 × 0.88 in.	33,600	19,300	1.5	4	....
	1.52 × 1.54 in.	28,200	.....	1.5	4	....
Kent		32,000	.....	2.0	4	....
Touceda. Avg. strength of commercial		41,000	.....	.....	.....	....
Malleable castings. Unusually good properties						
C. H. Gale		60,000	.....	.....	.....	....
		70,000	.....	.....	.....	....
H. R. Stanford, average of	3 tests 13/16 in. diam.	55,100	.....	5.2	.....	5.5
	3 tests 13/16 in. diam.	64,500	.....	2.8	.....	1.3
	2 tests, 13/16 in. diam.	69,100	.....	4.0	.....	2.6
	2 tests, 13/16 in. diam.	56,700	.....	8.2	.....	8.4
	3 tests, 13/16 in. diam.	51,600	.....	7.0	.....	7.7
	42 tests, 13/16 in. diam.	49,810	.....	6.61	.....	6.23
S. B. Chadsey		45,810	.....	6.25	4	....
		55,230	.....	10.55	.....	....

decarbonized in the annealing process, whereas in the America black-heart malleable iron only the skin is decarbonized, the metal adjacent for about ¼ in. partially so, and the central portions contain the full carbon percentage of the original hard white casting." The writer has italicized certain parts of the matter quoted in order to devote if possible a little attention to these parts later on. It is believed that the foregoing fairly sets forth the state of affairs as to physical properties during the period mentioned.

IMPROVED PHYSICAL PROPERTIES DUE TO RESEARCH

17 Let us see what improvements have been made within recent years, that is, since the research work was started for the Association. We will not select the best month's record, but will reproduce in part the very last report sent to the members, that is, the record for March 1919.

18 Table 2 covers the physical tests for ultimate strength and elongation on bars received during the month of March 1919. It would be impossible, almost, to consider the figures in this record and fail to note that as the tensile strength increases, so does the elongation. These monthly records have been kept in this manner

TABLE 2 ANALYTICAL EXAMINATION OF PHYSICAL TESTS FOR ULTIMATE STRENGTH AND ELONGATION ON TEST BARS SUBMITTED DURING MARCH 1919

Limits of ultimate strength	Per cent of bars	Per cent elongation
Under 40,000 lb.	0.40	4.50
Between 40,000 and 42,000 lb.	0.88	6.86
Between 42,000 and 44,000 lb.	2.79	7.67
Between 44,000 and 46,000 lb.	6.47	8.13
Between 46,000 and 48,000 lb.	10.29	9.50
Between 48,000 and 50,000 lb.	17.16	10.09
Between 50,000 and 52,000 lb.	17.95	11.55
Over 52,000 lb.	44.06	14.67

for four years or over and there is no exception to this rule. It will be noted that only 0.40 per cent of the total bars received tested under 40,000 lb. per sq. in. As a matter of fact, only 10.54 per cent were under 46,000 lb., while 44.06 per cent stood over 52,000 lb. with an average elongation of 14.67 per cent. The best individual record showed an average of 59,681 lb. ultimate and 21.47 per cent elongation. The worst was 39,942 lb. ultimate and 4.20 per cent elongation. The latter record belongs to a member who but very recently joined the Association, and bears out quite well the thought that the writer has been endeavoring to convey.

19 The members who submitted bars classified under Railway Work were twenty-two in number. The average ultimate strength and elongation of test bars submitted by the eleven members having the highest averages are found to be 53,559 lb. and 15.56 per cent,

respectively. Carrying through the same operation with the twenty-six members who are not thus classified, it is found that the average ultimate strength and elongation of these thirteen are respectively 52,327 lb. and 12.42 per cent. Taking the average of these twenty-four members, we find that the ultimate strength is 52,943 lb. and the elongation is 13.99 or practically 14 per cent. Lest our intention be misunderstood, it should be explained that our effort is not directed toward securing an increased ultimate strength and elongation so much as uniformity of product. The aim is to secure a product that the engineer will readily acknowledge possesses excellent physical properties, which vary but little from heat to heat. Within what can be considered quite narrow limits, this is what these particular twenty-four members are doing, while most of the others are not far behind. It may also be of interest to state that from January 1, 1917, to March 31, 1919, the average ultimate strength of the test bars of the Association as a whole has been over 51,000 lb. ultimate and the elongation 12.50 per cent.

INFLUENCE OF WAR CONDITIONS

20 In considering the last statement the following facts should be taken into account. War conditions during 1917 and 1918 made it quite impossible to secure appropriate pig iron and fuel. It is only fair to state that most of the companies were greatly handicapped during this period. It was solely and only through an intimate knowledge of the metallurgy of the process derived from the research work that made such a showing possible. Aside from this the membership has been and is constantly growing and the total average is and for some time to come will be necessarily affected as a consequence, as it takes some few months to get a new member in line. It is not unfair to assume that a still better showing could have been made had the times been normal, and had the membership been confined to those only who were members when the research work was first started. At the beginning of our investigations an elongation of 10 per cent was considered to be an indication of a superior product. As our knowledge of the metallurgy of the process increased, accompanied by better air-furnace practice and annealing-oven conditions, the elongation particularly began to climb. An elongation of 20 per cent is not now looked upon as unusual; elongations of 25 per cent occur with considerable frequency, while we have had numerous bars that have run as high as 30 per cent and

several of 31 per cent, which for an untreated cast-iron product we believe to be quite extraordinary.

METALLURGY OF CAST IRON

21 The raw material from which malleable-iron castings are made is pig iron, but this must be of suitable composition for the process. The usual elements or "impurities" (as they are frequently called) in pig iron are carbon, phosphorus, sulphur, manganese and silicon. If any of these elements combine with the iron, or combine with each other in definite proportion, compounds will be formed and it must always be kept in mind that compounds have very different physical properties from those possessed by the elements that form them. For instance, pure iron is so soft that it is difficult to machine it in such a manner as to leave a nice clean surface, as the chips have a tendency to shear or tear off before the edge of the tool has cut through, while graphite, the form in which free carbon exists in pig iron, is certainly very soft; and still, when these two unite to form carbide of iron, in the proportion always of about 6.67 per cent carbon to 93.33 per cent iron, they yield the hardest substance that can be produced from iron or steel by any known method. Now it happens that there exists a preferential and reciprocal attraction between some of these elements. For instance, if a piece of pure silicon is dropped into a bath of pure iron, or even iron contaminated with phosphorus, sulphur and manganese, when solidification takes place the silicon will be found to exist in the iron not as such, but united with it to form a definite compound called silicide of iron, because any tendency it has to remain by itself is overcome by the iron, for which it has a greater attraction than for any of the other elements present. In the event that any element in pig iron remains uncombined, that is, fails to unite with the iron, or with any of the other elements present, then the element is said to exist "free."

22 *Compounds of Iron.* In the absence of much silicon, carbon will always unite with the iron to form the compound called carbide of iron, a structural constituent known as cementite. We have already stated that silicon unites with iron to form the compound known as silicide of iron. Phosphorus in commercial pig iron suitable for use in the manufacture of malleable iron always combines with the iron to form the compound phosphide of iron. Sulphur and manganese have a reciprocal attraction for each other, greater

than either has for the iron or the other elements, and in consequence we can consider that the conditions are always such commercially that the sulphur and manganese will unite together to form manganese sulphide. Any manganese in excess of this requirement will unite with the carbon to form what we will at present call manganese-iron carbide (manganiferous cementite). It can now be stated that of the five impurities or elements referred to, none, except the carbon, can occur in the iron in the free state, and the latter can never exist free, as graphite, in whole or in part, unless it happens that there is present in the iron an amount of silicide of iron sufficient to prevent the formation of iron carbide. As indicated, silicide of iron has a tendency to break up, that is, render unstable, the carbide of iron that forms during, or shortly after solidification, causing it to dissociate into its two original soft constituents, iron and carbon; but as we will see later on, this action will not start unless a certain amount of silicide of iron be present.

23 *Influence of Time Element.* It must be borne in mind that all reactions are governed more or less by a time factor. If the pig or the casting, although containing high silicon, be rapidly cooled the iron may become rigid so quickly that the carbon will be denied the time to separate out as graphite in spite of the presence of an amount of silicon that would have precipitated the carbon under conditions of normal cooling. Sulphur, on the contrary, acts to encourage the union between carbon and iron; that is, it acts to stabilize the carbide of iron. Inasmuch as sulphur unites with manganese to form sulphide of manganese, a compound of the nature of slag and supposed to be inert, the writer fails to understand just how it functions to stabilize the carbide. He acknowledges that this appears to be the case even when there is present at times more manganese than sufficient to satisfy the sulphur. He acknowledges also his ignorance as to the cause, but believes that the sulphur does not unite with the manganese until after it first has exerted its influence in some manner unknown on the carbon.

24 *Free and Combined Carbon.* When the carbon exists free, it occurs more or less uniformly distributed throughout the mass of metal, in plates of varying shape, size and thickness as shown in the micrograph of Fig. 2. When in this condition, the fracture of the pig iron usually will be coarsely crystalline and very black, the iron will be very soft and easy to machine, and it will lack high strength. If, on the other hand, the carbon is present entirely in the combined form, the fracture of the pig will be white and vitreous, it will be

impossible to machine it due to its extreme hardness, and while it will have a much higher ultimate strength than when the carbon is in the free state, it will be brittle. The great difference in structure when the carbon is combined and when it is free can be seen by comparing the micrograph of Fig. 2 with that of Fig. 3. Between these two limits it is possible to have pig iron in which the carbon will exist in both the combined and graphitic form in any proportion of the total carbon content, while the appearance of the fracture, machinability and strength, obviously will depend upon what part of the total carbon content remains combined and what part remains graphitic. If then we see a sand-cast pig (one that has cooled normally) whose fracture is white we know immediately much about its physical characteristics and a few things about its composition; that is, we know that it is extremely hard and brittle, that its silicon content must be low, and that all the carbon must be in the combined form. If, on the other hand, the fracture is very black and coarsely crystalline, we will know that the iron is very easy to machine, that is will not have a very high ultimate strength, that the silicon is not very low but on the contrary is probably quite high, and that most if not the entire amount of carbon exists in the free state.

25 *Effect of Silicon.* The foregoing practically signifies that whether the carbon will be in the combined or free state in a pig iron or a casting depends primarily upon the percentage of silicon present, though the part played by the rate of cooling must also be considered, as we may have thick or thin castings or castings with thick and thin sections. Consequently, through the control of the silicon, the condition in which the carbon will remain in the iron can be determined. As the control of the silicon is easy, so is the control of the condition in which the carbon will exist in the casting.

26 Ignoring for the moment the influence of the rate of cooling, it has been stated that in order to obtain an iron that will be white in fracture, the silicon must be low enough to lack a tendency to break up the carbide of iron into its original two components. White iron is made up structurally of two elemental constituents, carbide of iron (cementite), and carbonless iron (ferrite), the former existing in part free, and the remainder forming a mechanical mixture in definite proportion with all of the carbonless iron, to make a constituent to which the name pearlite has been given. In Fig. 3 the white constituent is the cementite existing free, while the dark areas are the pearlite. Many years ago it was discovered that if particles of carbide of iron, in intimate contact with carbonless iron of the

FIG. 2 MICROGRAPH SHOWING FREE CARBON IN CAST IRON

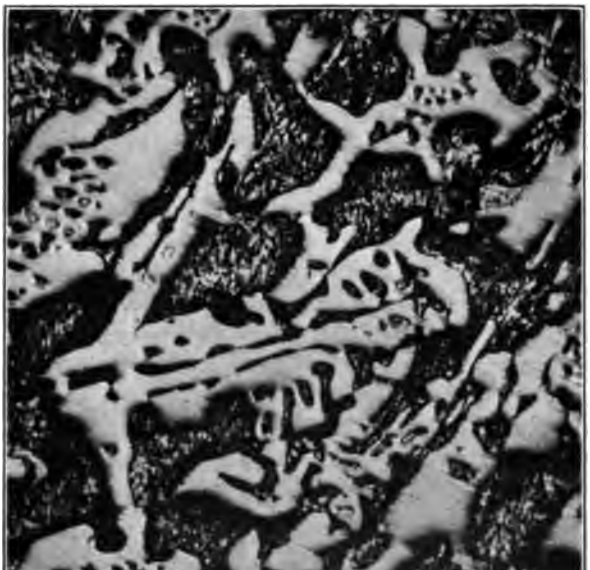


FIG. 3 MICROGRAPH SHOWING COMBINED CARBON IN CAST IRON

character referred to, were maintained at a temperature of bright redness for some 40 to 60 hours and then allowed to cool with extreme slowness, that this very hard constituent could be, through this treatment, broken up into the two very soft constituents through whose union it was formed.

PROCESS OF MAKING MALLEABLE CASTINGS

27 It can now be stated that the process for making black-heart malleable castings involves two steps. The first step consists in making a casting in which the totality of the carbon will exist as carbide of iron, when the iron will have a structure shown in Fig. 4, which structurally is like Fig. 3 but which contains less free cementite, because air-furnace white iron has an average carbon content of but 2.40 per cent as against an average of 3.50 per cent in white pig iron. In this step, then, is produced a casting white in fracture, hard and as brittle as glass. The second step consists in subjecting this white-iron casting to a heat treatment such as will serve to break up this hard carbide into its two original components, both of which are very soft, and hence from a white-iron casting we can obtain through heat treatment one that possesses the properties of strength, toughness and ductility. Fig. 5 shows the structure of a normal well-annealed piece of malleable iron. The white ground mass is the carbonless iron (ferrite), while the dark continent is the carbon that precipitated out during the anneal. If Fig. 5 be compared with Fig. 4, an idea will be gained of the profound change that has taken place in the structure during the annealing process.

28 As previously pointed out, the raw material for the manufacture of these castings is pig iron, though pig iron does not constitute the entire charge, as not only must the sprue from the previous heats be melted, but it is more than likely that when these two alone are used the carbon in the mixture will be too high to yield a white iron of suitable composition to produce the strongest product. Consequently, in order to lower the carbon to the necessary limit, steel or other very low-carbon scrap must be used to bring about that end. In by far the greater majority of cases this mixture is melted in a reverberatory furnace, commonly known as an air furnace, but the cupola, the open-hearth and the electric furnace can be and are used for that purpose. Inasmuch as the iron for more than 95 per cent of all of the malleable castings produced is melted in the air furnace, we will confine ourselves to that particular apparatus. A photograph of the furnace is shown in Fig. 6.

DESCRIPTION OF THE AIR FURNACE

29 The furnace consists essentially of a fire pot, hearth and stack. Some furnaces have a solid roof, the charge being "peeled

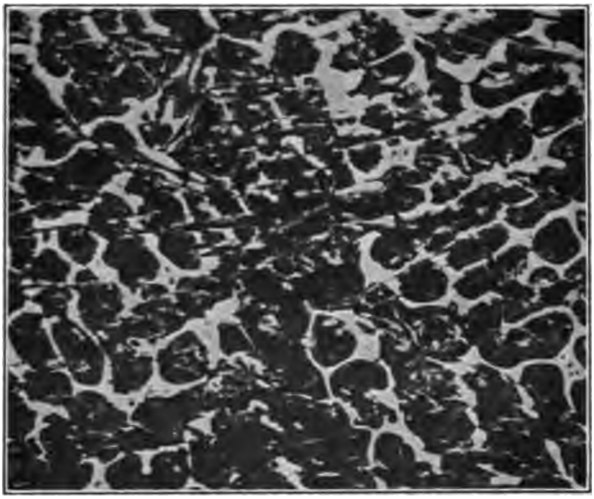


FIG. 4 MICROGRAPH OF HARD, WHITE CAST IRON

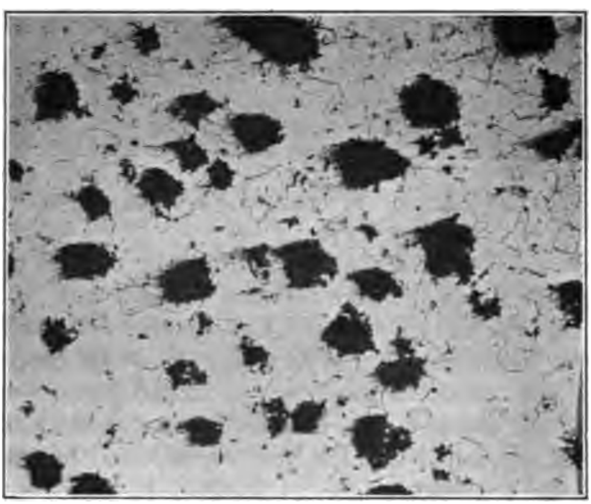


FIG. 5 MICROGRAPH OF NORMAL, WELL-ANNEALED MALLEABLE IRON

in" through the charging door, but in the larger number of cases the roof is made up of bungs that can be removed during repairs and a sufficient number of them lifted off when necessary for the purpose of admitting the charge. While a few of the furnaces operate with natural draft, most of them use a forced draft of from 3 to 4 oz. pressure. Although oil is advantageously used for fuel in a few instances, where its cost is not prohibitive, bituminous coal having a volatile combustible of from 25 to 35 per cent is the heating agent used by the majority. In order to burn the volatile products of the coal, as well as the CO generated from it, when, as should be the case, a deep bed of coal is used, secondary air is admitted through a series of tuyeres or a continuous tuyere, which is located far enough in front of the grate bridge wall and so inclined that the air will

FIG. 6 TYPICAL AIR FURNACE

enter and be deflected about 15 in. from the base of the bridge. In this manner a maximum temperature of about 2800 to 2950 deg. fahr. can be eventually obtained. Considerable time and thought have been expended in an endeavor to see if it would not be possible to regulate the amount of secondary air to just insure perfect combustion by the use of a CO_2 recorder. After considerable experimentation it was found that the plan was defeated by the inability of the recorder to act with sufficient promptness. The furnaces vary in capacity from about 7 tons for the small furnaces to 35 tons for the larger ones. The best are very inefficient and in practice the fuel ratio is about three of iron to one of coal. The average practice does not exceed 2.5 to 1. In numerous cases waste-heat boilers are used and while this conserves heat, it obviously does not add to the furnace efficiency. The average furnace has a capacity of about 15 tons.

THE CHARACTER OF THE AIR-FURNACE CHARGE

30 It has been explained that the object of the first step in the process is to obtain white-iron castings, or those in which the totality of the carbon is in the combined form, and we have seen that this object can be attained through a control of the silicon; that if the silicon is high, the carbon will be precipitated in the iron as graphite, which will yield a gray-iron fracture, while if it is low, the carbon will combine with the iron and the fracture of the casting will be white. As pig iron in which all the carbon is combined can be easily made and purchased, it would appear logical to assume that inasmuch as the sprue which, of necessity, forms part of the charge must be white, why not use this character of pig, when all that will have to be done will be to melt the charge, superheat it to a temperature that will successfully run the castings and let it go at that. Actually this is what is done when the inferior cupola product is made. To understand why it is impractical to do this in the air furnace, it is necessary to have a previous knowledge of what takes place in this apparatus during the melting of the charge.

OXIDATION IN THE FURNACE

31. The electric furnace is the only apparatus in which a high temperature can be attained with a reducing atmosphere. A high temperature in any other melting furnace consequently implies an oxidizing atmosphere. This in turn means that during the melting and superheating of the metal in the air furnace, those elements in the iron that will oxidize and remain oxidized will be eliminated to an extent that will depend upon the time the charge is exposed to the oxidizing influence of the furnace atmosphere. Practically, all of the constituents are oxidized in part, but only four remain in that condition. These four are the iron, silicon, manganese and carbon. Oxidation actually starts as soon as the charge on the furnace hearth begins to get fairly well heated up. The bulk of the oxidation takes place, however, just as the iron begins to melt, for at this stage the molten iron as it runs off the melting sides of the pig is freely exposed to the oxidizing furnace gases, and presents a large surface in proportion to its weight, with the resultant elimination of

some silicon, manganese, iron and carbon. The iron is oxidized to ferrous oxide, the manganese to manganous oxide, the silicon to silica, and the carbon to CO which escapes as a gas. As the silica has acidic properties, while iron and manganese oxides are basic in their action, they will unite together to form a slag consisting of a double silicate of iron and manganese, although the composition of the slag as tapped is much more complex, being modified by the sand on the pig and sprue and the unpreventable erosion of the hearth side walls and bottom. During an average normal heat there will be a loss in silicon of 0.35 per cent actual, a loss in carbon of about 0.35 per cent actual, and an average loss of about three-fifths of the manganese.

REASONS FOR NOT USING WHITE PIG-IRON CHARGE

32 The reason why it is impractical to use white pig iron can now be made clear. In the making of coke pig iron in the blast furnace, the amount of silicon and sulphur that will be in the product is largely a function of the furnace temperature. When this is low, there will be produced a pig in which the silicon will be low and the sulphur high. With a hot furnace the conditions are reversed and there will be obtained a pig high in silicon and low in sulphur. As a rule, a low-silicon coke pig means one that is prohibitively high in sulphur, but even if this were not the case, pig of this character could not be successfully used in the air furnace to make white-iron castings for the manufacture of malleable iron, because such castings must have a silicon content between certain limits. If the silicon is too low, then on annealing we will fail to obtain a casting whose fracture will be normal. If then we used a pig as low in silicon as obtains in the case of white pig iron, and with this were forced to use 35 to 40 per cent of sprue that must of necessity be still lower in silicon than the charge from which it was made, then we would obtain metal too low in this element to yield good castings. The situation, however, is worse than this, because to lower the carbon to the necessary limit steel scrap must be used, which contains practically no silicon, while by the time the molten metal is hot enough to be tapped from the furnace 0.35 per cent of the silicon in the charge has been eliminated. Fortunately there is no necessity of using white iron, because it is a perfectly simple operation to convert pig iron that is gray in fracture, and in which the carbon exists mostly ;ee, into castings in which all of the carbon is combined.

CONTROL OF SILICON CONTENT

33 In Fig. 7 can be seen the fractures of six air-furnace test sprues. No. 1 of gray fracture was cast very shortly after the charge was completely melted, while No. 6, white in fracture, was cast when the metal was just hot enough to run the castings. No. 2 was cast about 25 min. after No. 1 and the others at succeeding equal intervals. In comparing the fractures in the order in which the sprues were cast, it can easily be seen that as the silicon and carbon content in the bath were being gradually lowered, the graphitic carbon lessened by degrees, until finally the silicon became so low that it no longer possessed the power during solidification to drive the carbon out of combination with the iron. It is simply a matter of holding the molten iron in the air furnace a sufficiently long time to eliminate the silicon down to a point where the amount that remains in the bath will be insufficient to precipitate any carbon when solidification takes place. Irrespective of how much silicon, within reason, were in the charge, it would be possible to finally eliminate enough to accomplish this object, but as this would involve a great waste of fuel, it would not be commercially practical to do it. The logical thing to do, and the thing that is done, is to charge a mixture with a silicon content just high enough to admit of the carbon combining, coincident with the arrival of the bath to pouring temperature. While too high a silicon content will result in a waste of fuel and valuable time, too low a silicon will not only defeat the obtaining of a superior product but the castings ordinarily will be unsound.

PHOSPHORUS CONTENT

34 The usual practice is to have the phosphorus content under 0.20 per cent, but if all the other elements are correctly proportioned and the white-iron castings correctly annealed, a much higher phosphorus content can be used and a very good product obtained. The writer is convinced that in the absence of combined carbon, considerable liberty can be taken with this element in sections that do not exceed $\frac{3}{4}$ in. It may also be stated that there seems to be no advantage to the quality of the product to run the phosphorus as low as 0.10 per cent, while there is a disadvantage in so doing that results from a lessening fluidity of the iron. Inasmuch as a pig of 0.20 per cent phosphorus can be purchased as cheaply as one containing 0.30 per cent, there is no commercial advantage gained in using the latter.

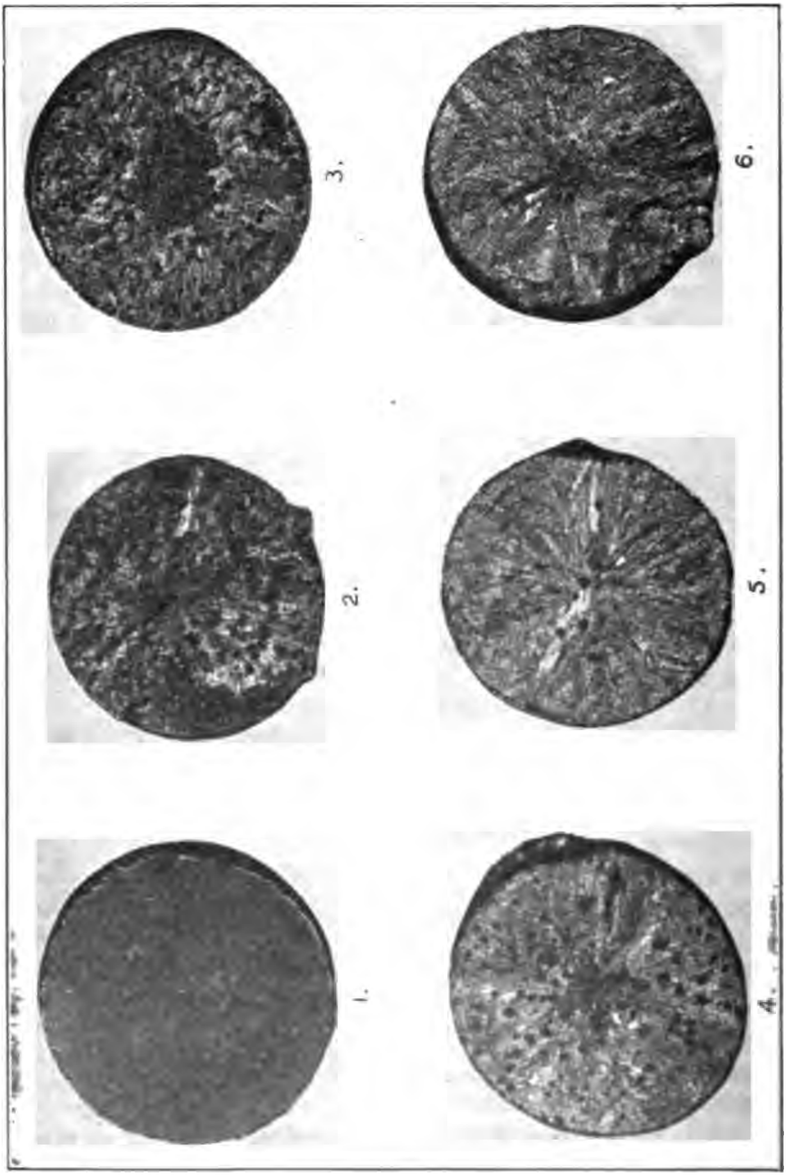


FIG. 7 FRACTURES OF SIX AIR-FURNACE TEST SPRUES

SULPHUR AND MANGANESE CONTENT

35 In considering the proper sulphur content, we must at the same time take into account the manganese, since due in a measure to their reciprocal attraction these elements unite to form manganese sulphide, as already explained. Provided the sulphur is properly balanced by the manganese, one need not fear injury from this element up to, say, 0.10 per cent, but approaching this point and beyond it to 0.12 per cent as the highest limit, great care must be taken not only to see that it is properly balanced by the manganese, but that the annealing temperature be kept low. Just how low can be better understood when the rationale of the annealing process is briefly entered into. As far as the writer has been able to ascertain, there is absolutely no advantage to be gained and, as a matter of fact, considerable disadvantage in striving for a low sulphur content in the castings. It is getting increasingly difficult to purchase malleable pig iron that will average much below 0.65 per cent in manganese and it is not easy to obtain it this low. When this percentage in the mixture or charge is exceeded, it is quite difficult if not impossible to obtain best results, unless some sulphur be present to control subsequently its action through the formation of sulphide. Under present average conditions the writer would prefer to see 0.05 sulphur in the product than one-half that amount.

36 The manganese content in the product should not be lower than 0.18 per cent nor higher than 0.36 per cent, the former when the sulphur is at its low, and the latter when it is at its high, limit. This does not mean that a good product cannot be made with a slightly lower manganese provided the silicon is not abnormally low, or with one slightly higher, but as a departure is made from the figures given, greater care must be exercised all along the line. An excess manganese makes the material very sensitive to heat and when the manganese in the hard-iron casting is high, the annealing temperature must be kept at the lowest point at which the carbon can be precipitated.

CARBON CONTENT

37 Concerning the desirable carbon content for the hard-iron castings, the writer believes that the higher the carbon the more easily can the carbon be precipitated in the anneal, but the weaker the product. It is impossible to obtain such a product as is now being turned out if the carbon be high. In looking over the literature of

even recent date one can find statements to the effect that a superior product can be made with a carbon content of over 4 per cent, which is further evidence in regard to what has been considered a superior product by some authorities. The literature of the subject is fraught with just such statements but invariably unaccompanied by data to back them up. To cover the ground briefly, it can be said that the lower the carbon, up to a point at which the carbon can be successfully precipitated, the more ductile and trustworthy will be the castings. If, however, the carbon is run too low, trouble will be experienced from lack of fluidity and in complicated castings

TABLE 3 CARBON ANALYSIS OF TEST BARS OF ELONGATION OVER 20 PER CENT AND ULTIMATE STRENGTH OVER 52,000 LB.

Total carbon, per cent	Elongation, per cent	Total carbon, per cent	Elongation, per cent
1.72	21.00	1.50	20.50
0.72	20.31	1.43	21.09
1.18	22.00	1.52	22.66
1.54	23.00	1.46	21.88
1.46	21.00	1.39	21.09
1.36	20.00	1.35	21.50
1.85	22.00	1.81	25.78
2.03	20.31	1.50	20.31
0.82	25.00	1.51	21.00
1.31	25.50	1.64	20.50
1.72	21.00	1.78	20.50
1.83	22.00	1.76	20.00
1.97	20.50	1.19	21.00
1.77	20.00	1.49	23.50
1.41	20.00	1.62	22.50
1.55	20.00	1.70	21.00
1.61	20.00	1.42	20.50
1.75	25.00	1.52	22.50

contraction cracks will be in evidence, while if the carbon is run still lower, a steely fracture will exist in the castings. A carbon content of 2.35 per cent in the hard iron will yield sufficient fluidity for most work and after such removal of the carbon as will take place during the anneal a tough, strong product will result. A few figures in this connection will prove illuminating.

38 Table 3 contains the carbon analysis of many bars in all of which the elongation was over 20 per cent and the ultimate strength over 52,000 lb.

39 In order that there will be no doubt as to the reliability of the carbon determinations, it can be stated that the drillings were

secured by milling off the metal at the reduced part, the milling cutter passing over the entire cross-section until sufficient drillings were obtained for the carbon determinations in duplicate. The drillings were then very thoroughly mixed. The carbons were run through by combustion in a platinum tube with all precautions deemed essential for trustworthy work. It can be further stated that all of the bars were regular ones sent in for test to this office, and it was unknown to any one that these analyses were to be made.

40 Without practically writing at too great a length on the subject, it would not be possible to cover completely certain facts of interest in connection with air-furnace work, proper construction of furnace, furnace operation and suggestions for further improvement in this direction.

ANNEALING OF HARD IRON

41 When a piece of air-furnace hard iron is gradually heated, a temperature is finally reached where many of its properties are very different from what obtains at a very slightly lower temperature. It will cease to be magnetic. Its structural composition will be different. The size of the crystals will be much finer than was the case under this particular temperature. It can be carburized beyond its original carbon content if packed in a carbonaceous material and held at this temperature for a sufficient length of time, while if packed in material that yields oxygen it can be decarbonized almost completely if the piece is thin. Also the carbide of iron can be broken up into its two soft constituents at the temperature referred to. This temperature is called the critical temperature, or critical range, and for air-furnace hard-iron castings it is in the vicinity of 1440 deg. fahr. It is the lowest temperature at which hard-iron castings may be successfully annealed. This statement must be modified by the further statement that in an oven under perfect control this temperature is the one that would be selected. In practice it would not be safe to adhere too closely to it, for the reason that should the castings while being held "at temperature" fall under the critical range, it would undo in large measure what had been accomplished above it. In the annealing of the castings one of the things to be avoided is oscillating temperatures, or temperatures alternating above and below the critical range. For this reason it is necessary to select a temperature some 100 to 150 deg. fahr. above the critical, say, 1550 deg. fahr. in which event, even if due to carelessness the temperature does drop a little, it will not be

liable to fall to a dangerous point. There is another reason why this latitude is deemed essential, though this does not obtain today to the extent it did formerly, due to the improvement in oven construction and operation. This has to do with the fact that in large ovens it requires considerable ingenuity to arrange flue openings, drafts, etc., in such a manner that the temperatures in all parts of the oven will be uniform, for which reason it is necessary to make sure that the temperature at the coldest corner is somewhat above the critical range, which will serve to safeguard oscillations in these locations.

PREPARATION OF CASTINGS FOR ANNEALING

42 In order to anneal the hard-iron castings that have previously been barreled or sand-blasted, chipped, gates ground off and inspected, they are packed in cast-iron pots where they are surrounded by an oxidizing packing. The packing has a dual function: to furnish oxygen through whose agency the castings will be decarbonized to the extent that is possible, and to avoid kiln warp, that is, prevent the castings from distorting. The pots, or stands, are sectional and each comprises a casting which forms its bottom, upon which four or five sections are superimposed. Each section consists of a rectangular or circular "ring" by which name they are known, whether they are of the former or latter shape. These rings are about $1\frac{1}{4}$ in. thick and vary in size at different plants, depending upon the dimensions of the castings to be annealed, but they would average, if rectangular, about 14 in. by 24 in. by 14 in. high. In building up a stand, a ring is placed on the stand bottom and then carefully filled with castings that are surrounded with packing. When the ring is completely filled, it is hammered on the sides with a light sledge in order that the packing will run down and fill in all voids. The second ring is then placed upon the first one, and this is filled in the same manner, which procedure is followed until the stand of four rings is completed. In Fig. 8 can be seen an annealing oven, partly filled with stands that are each four rings high. The top ring is filled with castings only to about two-thirds of its height, for if they were brought to the top they would be exposed to the oven gases. Instead, the top third of the ring is filled with packing and this in turn is covered with an iron plate. The top, and all joints in the stand, are then mudded or luted in order to prevent the entrance of the oven gases, after which the stand is lifted up by the charging truck and placed in position in the oven.

DESCRIPTION OF ANNEALING OVENS

43 The ovens are usually of rectangular shape, and vary in capacity from 15 tons for a very small oven to 50 for the largest ones. Their average capacity is about 25 tons. The usual fuel is bituminous coal, but hard coal, powdered coal or oil are used. The matter of construction must be dismissed with the statement that these ovens are being standardized and designed with a determination of securing uniformity of temperature throughout, and a great deal of study and experimentation is being devoted to this proposition. The flues are not only being properly proportioned for the

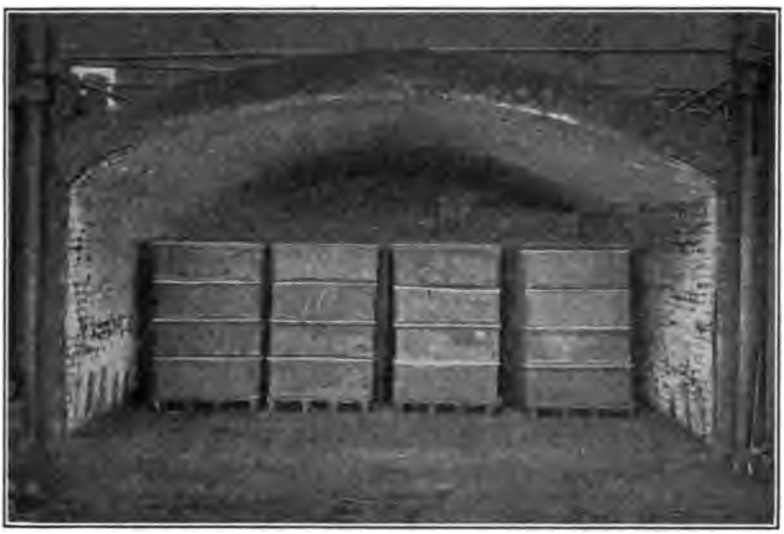


FIG. 8 ANNEALING OVEN PARTLY FILLED WITH ANNEALING POTS

draft used, but the flue openings so dimensioned that the heat can be drawn to any part of the oven in amounts sufficient to equalize temperatures, while provision is made whereby they can be easily kept clean. In the plants of the Association practically all of the ovens are under pyrometer control, and equipped at a central station where it is possible at any moment to ascertain the temperature at the hottest and coldest part of any oven, while a master pyrometer is used as a check on those that are permanently located. In this manner it is rather difficult for things to go wrong without detection. Air-furnace and annealing-oven operations have also developed from extremely crude methods to intelligent control, and this in large measure accounts for the improvements in the uniformity of the product.

LENGTH OF ANNEALING TIME AND TEMPERATURE LIMITS

44 In annealing, the castings are brought "to temperature," that is, to 1550 deg. fahr. or as high as 1600 deg. fahr. if thought best, as rapidly as it is deemed they can absorb the heat. Too great a forcing of the heat during this period is avoided, for if it is done the rings expand much more rapidly than the material within them, which leaving a space between ring and contents will allow the packing to bleed down from the top towards the bottom of the stand, lessening the compactness in the upper rings. In average practice it takes about 48 hr. for the oven to arrive "at temperature." The temperature of anneal is then maintained for a minimum of 48 hr., the time recommended being 60 to 72 hr. Firing is then stopped and the oven sealed tight in order that the castings will not cool faster than from 8 to 10 deg. per hr. while passing through the critical range. To safeguard this very important detail, this rate of cooling is maintained until the pyrometers indicate, that the oven temperature is less than 1100 deg., for on cooling, the castings are liable to be some 200 deg. higher than indicated by the pyrometer in the oven. After the temperature has been lowered to that point an opening is made in the front of the oven in order to allow it to cool more rapidly, for the reason that once the castings are at a temperature under the critical range no change can take place in their structural composition, so the only remaining precaution is to see that the castings do not cool so fast that internal strains can develop in them. From the foregoing, it can be readily seen that the average length of anneal occupies about seven days. From theoretical, as well as practical considerations, the writer does not believe that there is a possibility of safely lessening the time for this operation, by much more than one day without taking chances. He has designed an oven in which the temperature of 1600 deg. can be easily attained in 25 hr. and he is aware that when the composition of the hard-iron castings is such that the hard carbide is in its most unstable condition that even less than 48 hr. will suffice for the precipitation of the carbon, so that these two periods can be reduced somewhat, but danger is ever present if liberties are taken with the cooling through the critical range. In order then to safeguard the consumer as well as his own reputation,

the manufacturer should make no serious attempt to shorten the anneal unduly. The annealing capacity should be such as to make the attempt unnecessary. If it is made, however, then the pyrometer element should be inserted directly into the center of the pot, placed for that purpose in contact with the side wall of the oven, in order better to determine when the castings have actually arrived "at temperature" and the moment when he can commence to record the time the temperature can be started on its downward course, which procedure will enable him to operate more closely and accurately.

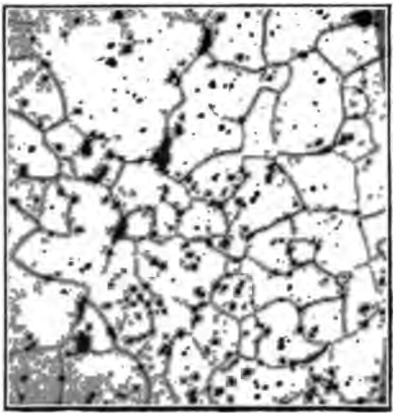
A CONVERSION RATHER THAN AN ANNEALING PROCESS

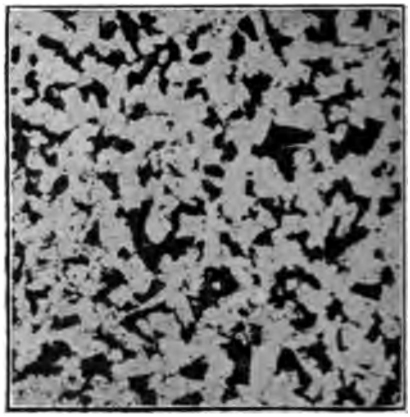
45 From what has preceded it should be evident that the second step in the manufacture of these castings should not be known as an annealing process, but more appropriately as a conversion process. The dominant function of an anneal is to obliterate coarse crystallization or an unsuitable one, and replace it by the most suitable that it is possible to produce in the object treated, and incidentally remove internal stresses. Annealing does not imply structural changes in the piece when cold, aside from grain size and grain refining. In the annealing of malleable-iron castings the dominant object is to convert white, hard iron, in which all of the carbon is combined, into a soft, tough, ductile iron in which no part of the carbon is in that state. In order to achieve this, it is necessary to maintain for a sufficient length of time a temperature just in excess of the critical range, which, as has been pointed out, coincides closely with that at which grain refining occurs, so it happens that during the conversion both objects can be practically attained.

STRUCTURAL CONSTITUENTS IN IRON AND STEEL

46 Before briefly entering into the rationale of the annealing operation and subsequently taking up certain facts in detail concerning the finished product and the conditions that influence the appearance of the fracture and physical properties, some particulars regarding the structural constituents in iron and steel should be presented in order that the exposition may be intelligently followed. Steel and iron crystallize upon solidification, and if a piece is properly polished and etched, the crystalline boundaries can be developed and seen under proper magnification. If a sample of pure iron is

thus treated, its structural compositions is considered to consist of 100 per cent ferrite, because the latter name has been given to iron which contains no carbon. (See Fig. 9.) If the iron were contaminated with the other four impurities to which we have alluded, it





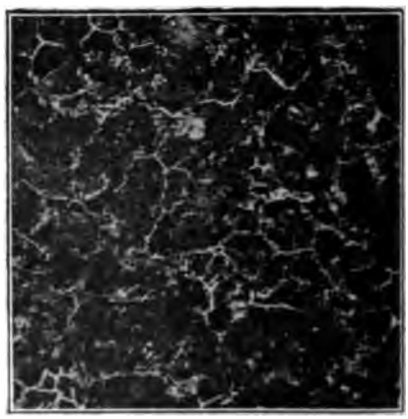


FIG. 9 MICROGRAPH OF 100 PER CENT FERRITE SAMPLE

FIG. 10 MICROGRAPH OF STEEL CONTAINING ABOUT 0.20 PER CENT CARBON

FIG. 11 MICROGRAPH OF PEARLITE IN MALLEABLE IRON

FIG. 12 MICROGRAPH OF STEEL CONTAINING 1.10 PER CENT CARBON

would still be considered to consist of 100 per cent ferrite. For example, ferrite has the property of dissolving silicon and its physical properties can be changed thereby, under which condition however it would still remain ferrite.

47 *Effect of Varying Proportions of Carbon.* We have already seen that when carbon is added to pure iron it unites with it to form carbide of iron in definite proportion, to which compound has been given the name cementite. It happens, however, that cementite on forming invariably insists upon mechanically mixing with ferrite, in the proportion of about 12 per cent of the former to 88 per cent of the latter, and will never exist free unless there is a deficiency of ferrite to satisfy it in accordance with the above ratio. As the carbon content in iron is gradually increased from zero up to about 0.90 per cent, structures are obtained in which obviously the percentage of ferrite decreases from 100 per cent to zero, while the pearlite increases from zero to 100 per cent. Any carbon steel therefore that contains less than about 0.90 per cent of carbon consists of free or excess ferrite and that amount of pearlite that the carbon content was able to make. Fig. 10 shows a steel which contains about 0.20 per cent of carbon, the white constituent being the ferrite and the dark the pearlite. Under a carbon content of 0.90 per cent, however, while not all of the ferrite will be used up by the cementite to form pearlite, all of the latter will be, so that none of it can under these conditions remain free, for it will all be used up in forming the mechanical mixture with ferrite, which in a slowly cooled steel will consist of alternate layers of these two constituents in the definite proportion of 12 per cent cementite and 88 per cent of ferrite. When a steel contains about 0.90 per cent carbon we find through calculation that it will form 12 per cent of cementite, and as the remainder of the material must be ferrite and figures to 88 per cent, such a steel must contain 100 per cent of this mechanical mixture, to which the name pearlite has been given, the appearance of which in malleable iron can be seen in Fig. 11. All carbon steels, therefore, in which the carbon is less than 0.90 per cent consist of pearlite and some excess ferrite existing as such. If the steel contains more than 0.90 per cent of carbon, then its structural composition consists of pearlite and the amount of cementite that was formed in excess of what was required to make pearlite, the pearlite growing less and the excess cementite greater with each increase in carbon content. Fig. 12 represents the structure of a steel containing about 1.10 per cent of carbon, the white rivers being the free cementite and the dark ground-mass the pearlite. The structural composition of both air-furnace white iron and white pig iron therefore consists of pearlite and excess cementite, the former shown in Fig. 4 and the latter in Fig. 3, the white constituent being the excess

cementite and the dark the pearlite. While these constituents are stable under the critical range, on passing through and over it they undergo a change, and a new one is formed called austenite.

48 *Austenite.* This constituent differs greatly from the others that we have considered, particularly in the fact that it is of indefinite composition; that is, no matter how poor in carbon the product may be, or how rich, we will obtain austenite on passing over the critical range, for in the former case we obtain an austenite lean in carbon and in the latter one rich in carbon. In order that carbon may be precipitated during the anneal, the iron must be in the austenitic condition, and if it were not for the fact that austenite is of indefinite composition the precipitation of the carbon would not be possible. The mechanism of the operation can be made clear by stating that during the anneal, as soon as the hard iron attains its austenitic condition, graphitization starts with considerable slowness, the austenite crystals rejecting some of their carbon in a minute nucleus and in so doing becoming leaner in carbon. This operation continues "at temperature" until the austenite is practically carbonless and the carbon nuclei have balled up and grown through a segregation of the carbon into nodules. Coincident with this operation some of the carbon diffuses out of the iron due to the oxidizing action of the pot atmosphere, which fact in turn would not be possible were it not that the austenite was of indefinite composition. It is a well-known fact that it is not possible to heat any iron product, for such operations as rolling, forging, annealing or hardening, in an oxidizing atmosphere without surface decarbonization. In these operations, however, the material is as a rule exposed to a temperature exceeding that of the critical range for a comparatively short interval of time, while in the case of the annealing of malleable iron we have seen that the shortest time to which the product is subjected "to temperature" is 48 hr. so that diffusion has ample opportunity to exert a very pronounced influence on the carbon content.

49 That diffusion takes place right to the center of such sections, as are generally found in quite the heaviest malleable castings, the writer has demonstrated in many instances, and when the mechanism of the operation is considered, it can be seen that this could hardly be otherwise as the principle involves the simple fact that if we have a layer of iron with a lower carbon content than its contiguous layer, carbon from the latter will diffuse into the former until equilibrium is established between the two. As the surface of the casting during the anneal is being continually impoverished in

carbon, there must be a travel of the carbon from the center toward the surface, for it is correct to assume that once the process is started that part of the section from the surface to the center consists of an infinite number of layers, each slightly lower in carbon than its contiguous one. The underlying principles in the manufacture of cement steel, and in case-hardening are just the opposite of those involved in the anneal, and there is abundant proof that the carbon can diffuse into the iron right to the center of the section. As the rate at which the pot atmosphere acts to remove the carbon from the immediate surface and the rate of diffusion of carbon from one layer to another is not the same, we can expect to see an almost carbon-free surface rim or frame in the section. It can be stated that were it not for the fact that the precipitation of the carbon was taking place coincident with its elimination, all of the carbon could be removed from the iron.

MATERIALS FOR PACKING ANNEALING POTS

50 In connection with the material used as a packing, considerable could be written if time permitted. The object to attain is to secure one that will evolve oxygen just as fast as the carbon can be eliminated, and no faster. If one is used that is too strong in this particular, then the surface of the casting will be attacked and we will have scaled castings, which are very troublesome and costly to clean. In addition to this we will run into another trouble that will be referred to later. If, on the other hand, it is too inert, we will fail to eliminate sufficient carbon and will not obtain the highest degree of ductility. The usual packing consists of a predominating proportion of inert material, such as ground air- or blast-furnace slag, pulverized firebrick, etc., to which has been added iron oxide in some form, such as rolling-mill scale, hammer scale, etc. With a correctly sized inert packing it is possible to secure excellent results, provided the voids are of such capacity as will contain the right proportion of air to furnish oxygen for the reaction. This scheme is in a measure rather difficult to operate uniformly unless after each anneal the packing is again sized.

51 It is the practice in some plants to anneal without packing in what are known as muffle ovens. Provided precautions are taken to have a tight muffle and the latter is well filled with castings to allow of its holding but little air before the muffle is sealed, very good results can follow, provided further that the carbon in the

hard-iron casting is low. As a matter of fact very creditable results can be obtained from castings that are very low in carbon (not so low as to prevent carbon precipitating) even if no carbon is eliminated, which means that the lower the carbon content in the hard-iron castings, the less dependent one need be in connection with the extent of carbon elimination. This matter is rather important in some instances where a reliable grade of malleable that can be machined at very high speed is desired. A rim of decarbonized iron can be so soft that in some machining operations the metal will crumble and tear in front of the tool edge and generate so much heat that it will be softened. Many samples have been sent to this office with the statement that the material was hard, which when examined were found to possess the character of skin referred to. The skin was dead-soft, and the castings proved hard to machine, due to the reasons stated. The general worth of a casting, its machinability and peculiar physical characteristics depend so much upon the structural make-up of the finished product, that some time must be devoted to a consideration of the factors that act to prevent either the structure or fracture or both from being normal.

APPEARANCE OF FRACTURE OF TEST PIECES

52 It takes a lengthy experience before one can tell from the fracture of the iron what may have caused its abnormal condition, which is one direction in which great progress has been made. When a normal, well-annealed piece of malleable iron is subjected to a steady, direct pull in a testing machine, a point is reached at which the crystalline grains of which it consists are elongated permanently, the stretch continuing until fracture takes place. If the fracture is examined it will be seen that the grains have elongated into finely pointed spines which gives what is called a "tooth" to the fracture. When light falls obliquely on such a fracture, there is produced a play of colors that yields a sheen caused by a reflection from the points and sides of the spines and the shadows that fall between them. As the grains in the decarbonized rim are more ductile than the rest of the metal in the section, they will elongate to a greater extent, and if the fracture is held in certain directions to the light, the width of the decarbonized rim can be seen by contrast, its color appearing under those conditions a little lighter than the rest of the section. (See Fig. 13.) This explanation is made and entered into because such a rim or border must not be confused with and mistaken

for the character of fracture that has a well-defined frame, that is, a border having not only a sharp line of demarcation between it and the core of metal which it surrounds, but an appearance wholly distinct from it. If the writer were asked to pass judgment as to

FIG. 13 FRACTURES SHOWING WIDTH OF DECARBONIZED RIM

the quality of a piece of malleable iron, based upon either the appearance of its fracture, or what would be shown by a polished and etched section under magnification, he is positive that he could render a more reliable opinion in the case of the former than would

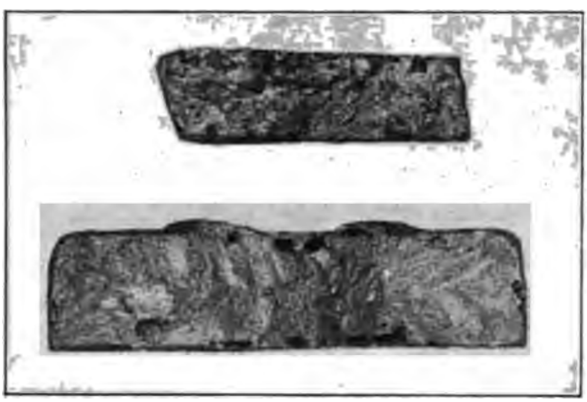


FIG. 14 FRACTURES DUE TO LOW SILICON ACCOMPANIED BY LOW CARBON AND MANGANESE

obtain in the case of the latter. The reason lies in the fact that even in a non-ductile product it is possible to have an absolutely normal structure, one which consists of a matrix of ferrite, throughout which are uniformly distributed nodules of free carbon, such as shown in Fig. 4, while if the fracture is as has been described, a normal structure at least can be predicted. If the ferrite is not ductile, then the crystalline grains will not elongate, with the result that we obtain a structural appearance that would be interpreted by those not familiar with the facts as belonging to a piece that had been insufficiently annealed. It has already been stated that when the silicon is too low a steely fracture will result, and also that the metal is liable to be unsound. In making this statement the writer is assuming that the silicon is low, not because too little was used in the charge, but low due to excessive elimination in the air furnace.

FIG. 15 MICROGRAPH OF SPECIMEN WITH 0.33 PER CENT SILICON AND 0.089 PER CENT MANGANESE

In such cases the low silicon will be accompanied by low carbon and manganese. Such fractures are shown in Fig. 14. In Fig. 15 can be seen the structure of a piece that had a silicon as low as 0.33 and a manganese of 0.089 per cent. The fracture of this piece was uniformly bright and coarsely crystalline. It had no frame. The structure consists of a matrix of pearlite and throughout it are distributed particles of undecomposed hard carbide (cementite) and small, well-rounded nodules of graphite, or temper carbon, as that carbon is called which separates out during an anneal. The casting from which this piece was taken was in an oven in which castings of correct composition annealed perfectly, and the presence of the particles of undecomposed carbide simply means that the anneal was not carried on for a sufficiently long time for this character of product. The casting, however, would have been very inferior even if all of the carbide had been broken up.

PICTURE-FRAME FRACTURES

53 Very frequently low silicon-carbon-manganese of certain compositions will yield what are known as picture-frame fractures, such as are shown in Fig. 16, which are typical and have the following composition: Silicon, 0.54; phosphorus, 0.162; sulphur, 0.053; manganese, 0.108; total carbon, 2.01. This piece when polished and etched showed the following characteristics: a decarbonized surface border, an inner ring of coarsely laminated pearlite, and within this a core corresponding in structure to that of normal

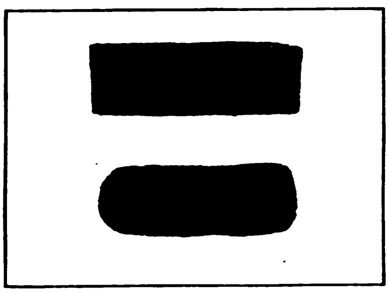


FIG. 16 PICTURE-FRAME FRACTURE IN LOW-SILICON-CARBON-MANGANESE SPECIMEN

malleable iron. Fig. 17 shows the decarbonized border surrounding the pearlitic ring, Fig. 18 the structure of the pearlitic ring, and Fig. 19 the core within the pearlitic ring. It is the presence of this ring of pearlite whose ductility is so much less than that of the metal in either the decarbonized border or core that produces on fracture the sharp line of demarcation between frame and core. While in this particular fracture the frame is fiery bright and finely crystalline and the core black, there are picture-frame fractures that show various color characteristics of frame and core, but it will be found that invariably the frame has its pearlitic ring of greater or less breadth. The pearlite is not always coarsely laminated, but as a rule has the appearance and consists of an amount of pearlite that would correspond to a 0.35 per cent or 0.45 per cent normalized carbon steel.

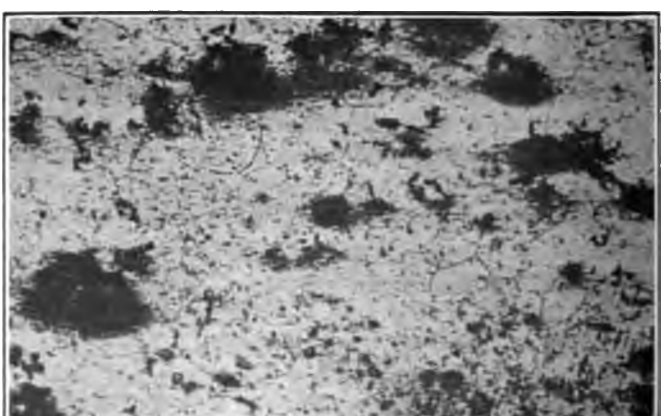


Fig. 17 Micrograph of Decarbonized Border

Fig. 18 Micrograph of Pearlite Ring

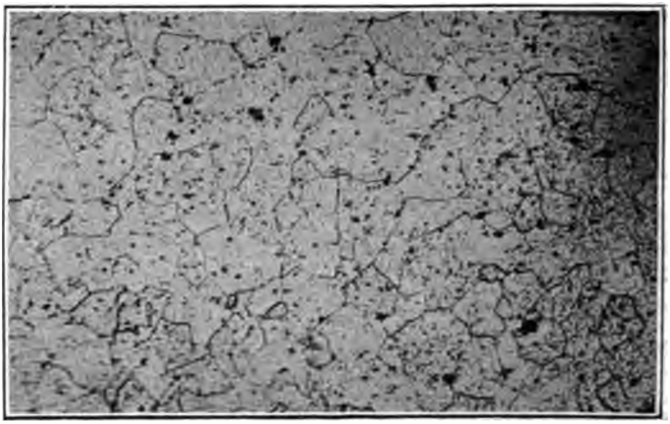


Fig. 19 Micrograph of Core within Pearlite Ring

54 If the sulphur in the hard iron is unduly high and particularly if not well balanced by the manganese, the castings will almost invariably show a picture frame on fracture, and especially is this true if the temperature of anneal is too high for such a composition. If the manganese is too high and not well balanced by the sulphur, the same result will follow. In each case there is an appearance to the frame and core indicative of which is which. The frames shown in Fig. 20 are rather typical of the latter. In this case the frame is dove-colored, while the core is black but sparkling.

55 While time is not available to enter into a full discussion of what has been discovered in regard to picture-frame fractures, there are some points that can be recorded. There are some compositions

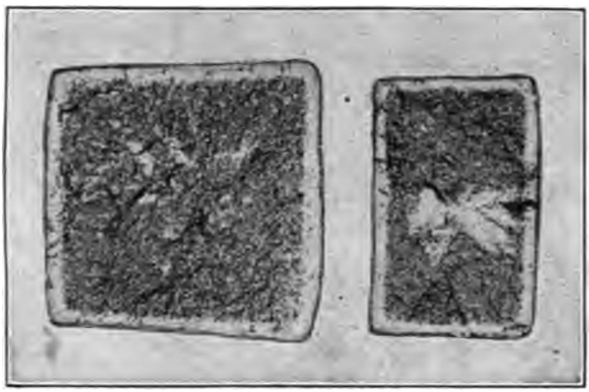


FIG. 20 PICTURE-FRAME FRACTURES IN HIGH-MANGANESE SPECIMEN NOT BALANCED BY SULPHUR

that unquestionably have frame-producing tendencies. These compositions will not produce a frame when annealed in an atmosphere that is not oxidizing. The surface structure of the hard iron has nothing to do with the problem as the writer has had $\frac{1}{4}$ in. ground off of one side of hard-iron samples and upon annealing the frame was in evidence equally on all sides. It is believed that the following facts are pertinent to the situation: Not only do certain compositions affect the ductility of ferrite, but the same is the case with a pearlitic structure. We can have ferrite that will elongate into very long spines and ferrite that will fail to elongate at all. We can have a pearlitic grain that can be ductile and those that are not. The foregoing is stated because the writer believes that whether or

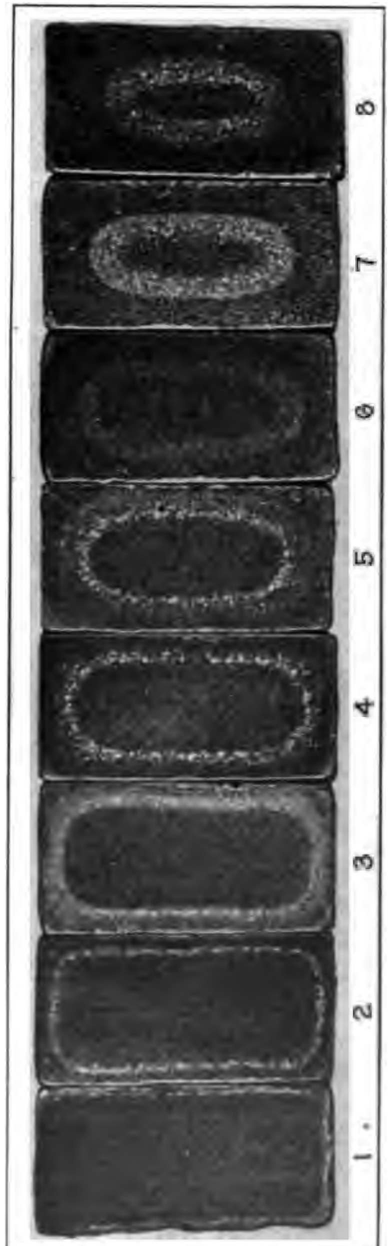


Fig. 21 Pearlite Ring in Successive Stages of Annealing

not a frame will be produced in the fracture depends upon the breadth and ductility of the pearlitic ring, because a slight pearlitic ring can be present within a decarbonized border without a picture-frame fracture being produced. It is his belief that whether there will be a pearlitic ring or not depends upon the rate of surface decarbonization, as compared with the rate at which a dissociation of the cementite takes place. When conditions are such that there will exist a region between the decarbonized surface border and the core that will have a carbon content of about 0.90 per cent, equilibrium seems to be established in this region, and if any carbon

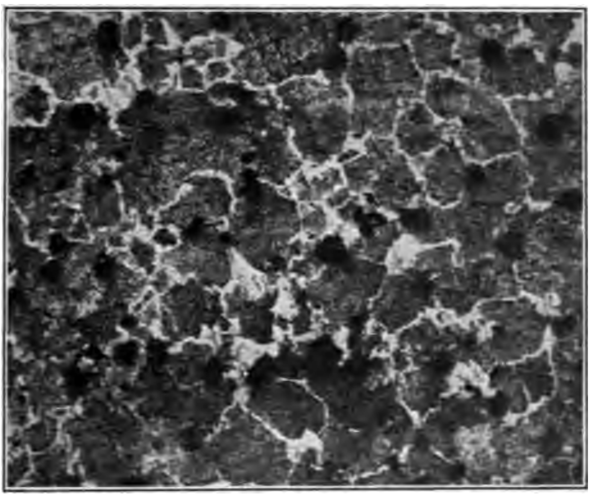
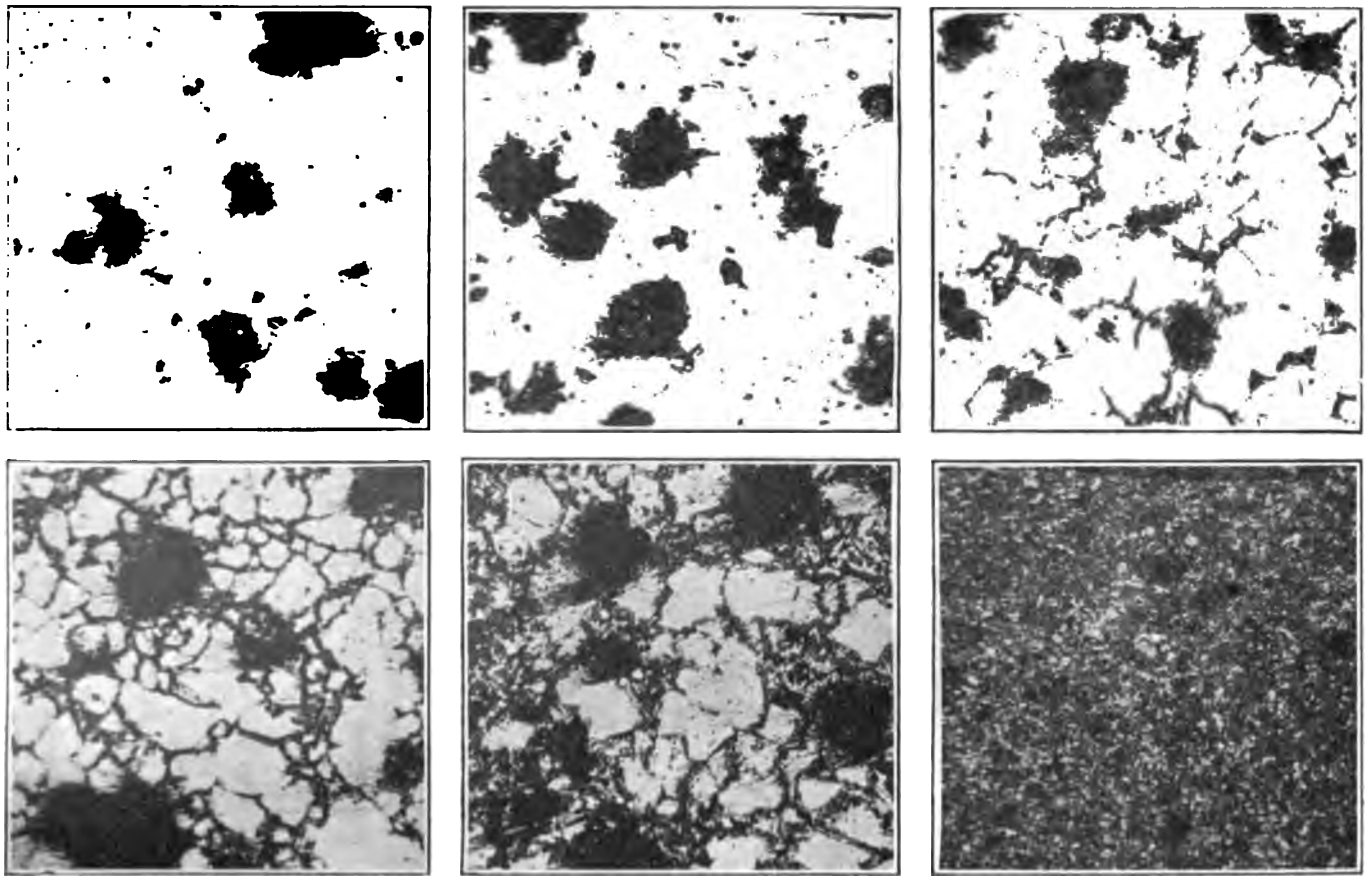


FIG. 22 MICROGRAPH OF MALLEABLE IRON OF HIGH ULTIMATE STRENGTH

passes from this region to the decarbonized border it is replenished by carbon from the core.

THE PEARLITIC RING

56 The writer believes that such is the case, and is of the opinion that perhaps the samples shown in Fig. 21 may have a bearing on the case. A well-annealed bar was cut into eight pieces. A section from the first piece was polished and etched, and the other pieces all packed together and given another anneal. The second specimen was then prepared like the first. The remaining six were then given a third anneal, and the third piece polished. As this procedure



FIGS. 23 TO 28 MICROGRAPHS SHOWING EFFECT ON STRUCTURE OF HEATING TO VARIOUS TEMPERATURES

Upper Row Fig 23, Unheated; Fig 24, 1400 deg.; Fig 25, 1450 deg.; Lower Row Fig 26, 1475 deg.; Fig 27, 1500 deg.; Fig 28, 1675 deg.

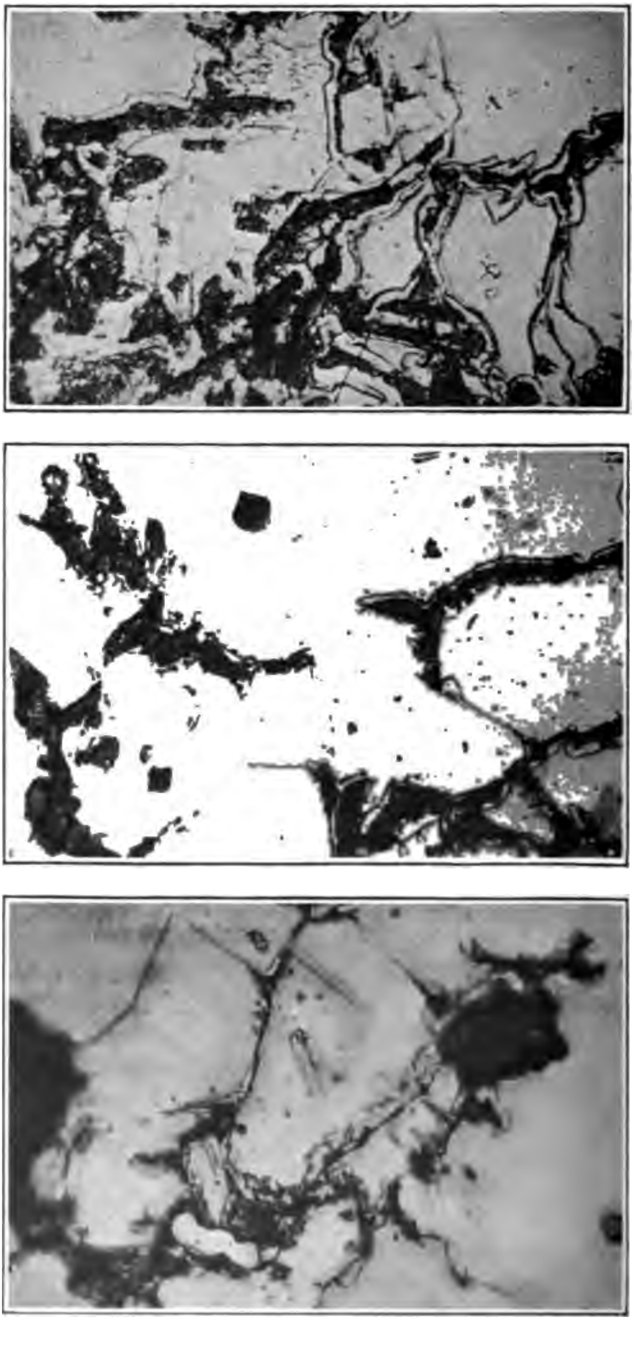
was continued, it follows that the eighth bar had eight separate and complete anneals. It will be noted that this very faint pearlitic ring which shows whitish and very faint in the once-annealed bar, is very distinct and well defined in the second and that it is wider and further in toward the center. It will be seen that with each anneal the pearlitic ring has widened and has a smaller periphery. As it was found that with each anneal the total carbon content decreased, it is plain that once the pearlitic ring has formed, it does not act as a seal for the passage of carbon from core to surface and that the width of the pearlitic ring is built up and added to more quickly from the core than it is robbed of carbon by the decarbonized border. The matter can be summed up as follows: If conditions are such that a region containing about 0.90 per cent or less carbon is formed, this region while permitting carbon to migrate or diffuse through it will at the same time be incapable of having its carbon precipitated. Under proper conditions the region can alter its position and increase in extent. For such a region to have a start it is essential that there be a very substantial difference in carbon content in two parts of the section.

STRUCTURE OF HIGH-STRENGTH MALLEABLE IRON

57 Malleable iron of very high strength accompanied by a ductility that can be considered good enough for certain purposes, has already been alluded to. While no effort has been made to exploit this product as yet, it would appear to the writer that there is a very large field in which it could be used to advantage. In Fig. 22 can be seen the structure of a sample that stood an ultimate strength of 84,000 lb. and had an elongation of 5.20 per cent. This material can be made with uniformity and it is believed from experiments that have been under way for some time that a 90,000 lb. ultimate and a 10 per cent elongation might be uniformly maintained. It will be noted that the structure consists of a ground mass of pearlite, in which are more or less uniformly distributed nodules of temper carbon. The structure readily explains why the product is of high strength.

EFFECTS OF HEATING MALLEABLE IRON

58 As it is frequently necessary to heat the finished product for the purpose of straightening it, for galvanizing and other purposes, it may prove instructive to see what happens when an annealed



Figs. 29, 30 and 31 Magnification, 750 Diameters, of Crystalline Boundaries of Figs. 25, 26 and 27 piece of malleable iron is heated up to and beyond the critical range. Ten pieces were cut from a normal, well-annealed malleable-iron bar. Fig. 23 shows the structure of this bar at ×200. The ten pieces were then placed in an annealing oven in the writer's laboratory that can be controlled with great accuracy. When the pyrometer registered 1250 deg. fahr., one piece was withdrawn and allowed to cool in the air. When the temperature reached 1300 deg. fahr. another piece was withdrawn. The other pieces were withdrawn at temperatures of 1350, 1400, 1450, 1475, 1500, 1550, 1600 and 1675 deg., respectively, Fig. 24 showing structure at 1400 deg., Fig. 25 at 1450

Fig. 32 Example of Large Malleable-Iron Casting

deg., Fig. 26 at 1475 deg., Fig. 27 at 1500 and Fig. 28 at 1675 deg. All of these micrographs were taken at a magnification of 200 diameters. It is apparent that no change has taken place in the structure up to 1400 deg., but that somewhere between 1400 and 1450 deg. the structure starts to alter in appearance. An examination of Figs. 25 to 28 shows that increased amounts of pearlite result as the temperature is increased, and that in Fig. 28 nearly all of the temper carbon has been dissolved. It follows that for straightening, brazing and other operations that necessitate the heating of a malleable-iron casting, the temperature used should be well under 1400 deg. fahr.

59 In order to obtain some idea of the mechanism of what takes place, micrographs at a magnification of 750 diameters were taken of the crystalline boundaries of pieces corresponding to Figs. 25, 26 and 27, which are shown in Figs. 29, 30 and 31. It would appear that as the carbon is dissolved the solution takes place through the amorphous iron in the crystalline boundaries, a fact that the writer has not heretofore seen mentioned. This is possibly the manner in which the carbon starts to diffuse into iron during cementation. What might be written in connection with the numerous observed facts pertaining to this product would fill many pages. The writer has lacked time to even take up the matter on which he has written in the way in which he would have liked to have presented the subject, but he believes that enough has been covered

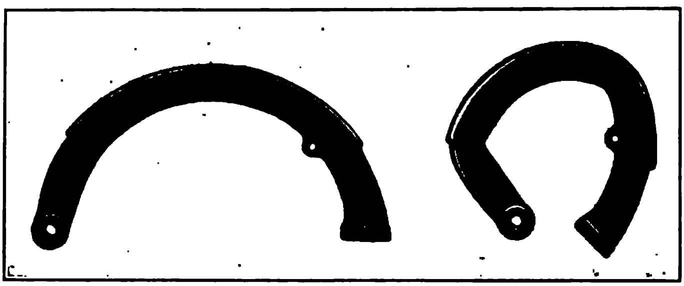


FIG. 33 EXAMPLE OF ABUSE MALLEABLE CASTINGS WILL WITHSTAND

to illustrate that through research and through that means only can the manufacturer make real and permanent progress.

60 For the benefit of those who labor under the impression that malleable iron is a product unsuited for any but small castings, Fig. 32 is shown. One casting of which the writer has a photograph is 5 ft. long, 23 in. high and some of its sections are 3 in. thick. That the metal, when well made, can stand great abuse is illustrated in Fig. 33; while the problem in regard to disproportionate sections that confronts the manufacturer, coupled with intricacy of design, and too often the intricacy of ill design, is well illustrated in Figs. 34 and 35. Irrespective of how good the metal may be, as shown by physical tests on bars and wedges, patterns are furnished so outrageously out of proportion that the good qualities of the metal can easily be destroyed and too frequently the metal is blamed when the design is at fault. These troubles are overcome as far as

possible by a thorough study of the best method of gating the casting, in order that no evidence of shrink will be present in any part. Research has made considerable advance in this direction and well-placed, well-proportioned shrink heads are now generally used. Not

FIG. 34 EXAMPLE OF CASTING OF DISPROPORTIONATE SECTIONS

infrequently, in the case of such castings as the ones referred to, the sprue, runners and heads weigh about as much as the casting.

COMMON FALLACIES

61 Before taking up the matter of fallacies, which during the past few years have been pretty well exploded although some still linger

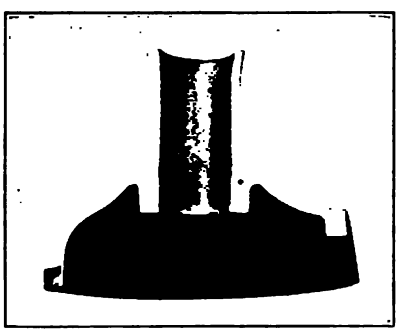


FIG. 35 EXAMPLE OF CASTING OF DISPROPORTIONATE SECTIONS

in the minds of a few of the engineers and consumers, the writer would like to make some further remarks by way of explanation concerning what was stated in the first part of this paper in connection with the literature of the subject. Enough has been shown to make clear the fact that in order to make a good quality of malleable iron, it is absolutely necessary that the white-iron-casting composition be correct. Many very interesting and laborious researches have been made in connection with the precipitation of the carbon during the anneal and in numerous other directions on hard-iron samples whose physical properties would be so low as to be worthless when annealed, due to their impossible composition. Had the same work been done by the eminent men who have carried through the experiments referred to on a hard-iron composition that was normal in all particulars except in connection with the particular element they were investigating, the metallurgist in this particular field would have been greatly benefited and his path made easier.

62 The fallacies that have been handed down and accepted by many of the engineers and consumers as true, are numerous, but

TABLE 4 TESTS OF MALLEABLE-IRON BARS WITH DECARBONIZED SKIN REMOVED

Mark	Ultimate strength, lb. per sq. in.	Per cent elongation in 2 in.
12-2-1	52,084	17.50
12-3-1	47,182	10.00
12-3-2	51,107	17.50
12-4-1	56,732	14.00
12-5-1	46,482	7.00
12-5-2	52,246	23.00
12-6-1	47,889	19.00
12-7-1	48,080	18.00
12-7-2	49,610	18.00

the following only will be touched upon as they are the most important:

a The strength of malleable iron lies in the skin. When it has been removed the remainder of the metal is found to be very inferior and not dependable.

b During the anneal, the elimination of the carbon is confined to the surface, and the amount removed from the rest of the section is inconsequential.

c When the section of a casting exceeds $\frac{5}{8}$ in. in thickness, it cannot be annealed throughout.

63 Concerning item *a*, the data in Table 4 will prove of value. Nine regular test bars were machined until the decarbonized surface was removed. These bars were all from different heats and marked as indicated in the table.

64 As the writer did not have duplicates of these bars, he was unable to make a comparison between the machined and the bars as cast and lacked time to run through a set for illustration, but the experiment should be unnecessary in any event in view of the above. It

Fig. 36

Figs. 36 and 37 Bars Magnified Seven Diameters to Show Carbon Distribution

is obvious and must be acknowledged that the metal in the decarbonized skin is more ductile than the core, so when a bar fails it must be conceded that it is the core that has parted first, for the reason that the metal in the skin has not at the instant of fracture reached its maximum elongation. Aside from the foregoing we have the practical evidence that presents itself in the case of the automobile industry in which thousands of tons of machined malleable-iron castings are used annually on parts that receive in service great abuse, such as wheel spindles, etc. On the other hand, the writer not only admits that when the skin is machined off some malleable-iron castings the remaining part is worthless, but admits as well that the castings would be such with the skin on. This, unfortunately, will continue to be the case until the purchasing agents cease to shop around and a contract is made on price as the basis rather than quality.

65 Taking up item *b*, the writer can, without encumbering this paper with the large amount of data he has on the subject, prove the falsity of this contention. In the figures quoted in Table 3 for bars of over 52,000 lb. ultimate strength and over 20 per cent, elongation, there will be noted two bars, one of which has a carbon content of 0.72 per cent and the other of 0.82 per cent. Aside from this there are fourteen with a carbon content of 1.50 and under. In Par. 39 an explanation is given of the manner in which the drillings were taken for analysis. It has already been pointed out that the carbon in the hard iron must be kept up to a certain figure, failing which the castings will not only misrun, but contraction cracks will spoil them. If we assume, in the case of the first two bars referred to, that the carbon was reduced by one-half, then in the bar that had but 0.72 per cent carbon, the carbon in the hard iron from which it was cast must have been 1.44 per cent, and we all know that it would be almost impossible to run such work, say nothing about subsequently annealing it. In a $\frac{5}{8}$-in.-diameter annealed bar such a low carbon content is unusual, but it proves the point that is being made, nevertheless. The writer has polished the section of two $\frac{5}{8}$-in.-bars and has photographed them at about seven diameters. These are shown in Figs. 36 and 37. They will furnish a fairly good idea as to how the carbon is distributed throughout the section, and indicate that the carbon does not vary by uniform gradation from surface to center, but in one region can vary slightly from what it may be in another. This does not signify that in the regions of highest carbon content the carbon has not been lowered through diffusion into its contiguous region, for many investigations have shown that this is just what does happens.

66 Item *c* can be disproved in a few words. We know that in order to break up the hard carbide in white iron it is simply necessary

that the casting be not only heated until the iron is in an austenitic condition, but maintained at that temperature for a certain interval of time. To state that thick castings of white iron cannot be annealed is to state that they cannot be brought to a uniform temperature throughout and maintained at that temperature. Such a claim would be an absurdity.

No. 1692

INDUSTRIAL PERSONNEL RELATIONS

BY ARTHUR H. YOUNG,[1] CHICAGO, ILL.
Non-Member

THE present focusing of attention on personnel relations will result in a new era in industry — as much an epochal change as we have had through various fundamental causes heretofore; such as the change from the original craftmanship to larger shop organization, made possible by power development, power transmission, changes wrought by methods of communication, and refinement of the methods of transportation, the era of consolidation of interests, and so on, and one has to stop and wonder why at last we have come to the consideration of the human factor.

We have probably had as great a refinement as would produce any revolutionary changes in machinery and methods and practices in everything except that relating to the human factor. All of us have witnessed the birth and the growth and the final development of the safety movement, and more recently, too, of employment management.

I refer to these two developments, particularly, as factors in personal relations of today, and illustrating how rapidly changes are coming about. I think it was only back in 1906 that the first organized effort in safety was made. It is generally credited to the South Chicago plant of the Illinois Steel Company, under the leadership of Mr. R. J. Young. He was a sort of genius who suddenly discovered that accidents were preventable — that it was no longer necessary to kill and maim men as a part of the making of steel.

DEVELOPMENT OF THE SAFETY MOVEMENT

At first not much attention was paid to his proposition, but it rapidly gained support, due to its humanitarian appeal, its evangeli-

[1] Manager, Industrial Relations Department, International Harvester Company.

Address, slightly condensed, delivered at the Industrial Relations Session of the Spring Meeting, Detroit, Mich., June 17, 1919, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

For discussion see p. 186.

cal aspect. The first attack was naturally very largely along mechanical lines — the removal of projecting set screws and other parts of revolving machinery, the railing of platforms, the guarding of gears, changes in construction, marking of aisle spaces, and all such things.

A very disappointing result was had, because at the end of two or three years, after several hundred thousand dollars had been spent, at that particular plant accidents had not been reduced over 20 per cent, and generally speaking, today safety engineers rate the mechanical factor or the correction of engineering practices as not contributing more than 20 per cent to the possible efficiency of the safety movement.

It was immediately discovered that the real problem was psychologically to teach a naturally careless man to become habitually careful, and that more than 75 per cent of our accidents were preventable by a correction of habits of thought — of thinking safety all the time. To illustrate: It is perfectly natural to cross a street just exactly as a chicken does — to look neither to the right nor the left, but step right out from the curb; and it is only by a process of education that we learn to pause at the curb and look up and down the street and then walk into the middle of the street and then glance again, and finally to cross the street. That is a process of education which will prevent accidents, and it was what was immediately arrived at in industrial safety engineering.

The shaping up of that problem brought its immediate solution, and brought into play a most interesting development from the laboratory of safety engineers. There were movies, bulletin boards, mass meetings, and finally the formation of public safety councils, the introduction of the study of safety in the schools. Its serious consideration by civic police departments and other parts of society is really the product of the industrial safety engineer, because he realized that in order to make a man think safety all the time it was necessary to work on him not only while he was in the shop, but at home as well. He carried his message to the man in the shop through the children whom he encountered at school, and during the hours that he was walking to and from the plant or riding to and from the plant by public safety bulletins and so had a continuous propaganda working 24 hours a day in order to make him think safety.

Immediate results were achieved. Taking again, for example, the South Chicago plant of the Illinois Steel Company, with which

I happen to be familiar, back in 1906 the frequency of fatal accidents in that plant was 47 during the year, or about one a week. By 1913 that rate had dropped to 7, and it has since stayed at about that figure.

As in fatal accidents, so in the less serious accidents, the frequency rate dropped in the same proportion; and in the Illinois Steel Company, there has been a reduction in the frequency of accidents to employees on duty of over 85 per cent since the 1906 rate. That was duplicated in nearly every plant of the United States Steel Corporation, and in thousands of other industrial concerns as well.

There have been a number of by-product results which have directly contributed to the growth of personnel movements and industrial relations, as we term them today, coming out of this safety movement. In the first place, it was found that not only did this human interest conserve lives, but that it was a paying proposition; originally it was an interest in the cause of humanitarian principles, and safety was regarded as somewhat of a fad by a great many employers.

After a sufficient period of experience on which to establish definite figures, it was found that safety was really "good business." The Steel Corporation has published the most interesting figures along this line, showing a net saving almost exactly equal to the cost of the safety work, or 100 per cent on the investment.

There is a striking unanimity in the form of organization for efficient safety work. The original movement in 1906 resulted in the formation of shop committees; as soon as it was realized that the problem was one of education of the workman and that his interest must be aroused, means were sought to turn the problem over directly to him, and I believe the first safety committees were organized at the South Chicago plant of the Illinois Steel Company.

Shop committees — the workmen themselves — were given charge of the safety work. It was their duty to investigate the cause of each accident, fix its responsibility, and make recommendations for a prevention of its recurrence. In addition, they were to make regular inspections of the plants, and by their foresight and recommendations were to prevent, as far as possible, the recurrence of accidents.

Some form of a shop committee is almost unanimously used by safety workers today in all industrial establishments of any size. It may be of the workmen themselves, it may be workmen in one committee reporting also to a committee of foremen, or it may be a joint committee, or it may be a joint committee of workmen, foremen and officers of the company, but the successful safety program of today includes shop committees.

In addition to the definite return on a cash basis of the safety work, there have been several interesting by-products. The first is the reduction of labor turnover. The Steel Corporation's reports, recently published, show that since 1906 they prevented, roughly, 25,000 fatal or serious accidents. They rate as serious any accident which disables a man more than 35 days, or results in a permanent disability, such as the loss of a thumb, or an eye, or other member of the body.

It is reasonable to assume that such an injury would require the replacement of the trained worker, and Captain Fisher and other writers on the subject have given some interesting figures as to the cost of replacement of trained workers. They vary anywhere from $10 to $100, depending somewhat on the way it is figured, and somewhat on the worker. Of course it is fair to say that steel-mill workers are rather well trained, and probably $100 as the cost of replacing a trained man with an inexperienced man is not an over-estimate. If we multiply 25,000 prevented replacements of trained workers, because of the efficiency of a safety movement, by $100 we have a substantial by-product, due to reduced labor turnover through efficient safety work.

There was also another manifestation. As is generally known, the Steel Corporation has been facing a suit for dismemberment. While the hearings were on several years ago there appeared voluntarily a number of old employees who petitioned that the corporation be not dissolved, because that would mean a return to the days of ruthless competition when they did not have standardized safety programs. Of course, no one knows what effect that action had on the decision, but it presented concrete evidence of the boosting of the morale of the workers through safety work.

THE QUESTION OF EMPLOYMENT MANAGEMENT

Turning for a moment to employment management, I will cite the circumstances of my first employment. At the age of thirteen years I went to work in a steel mill during vacation time and reported to the engineer who turned me over to his assistant. The assistant said, "Do you know anything about oiling?" Being thirteen years old, naturally I knew everything about anything.

He said, "All right, here is the cylinder oil, and there is the beeswax and some sand. Here is a ladder. Go to it!" And that was the extent of my instruction in oiling.

I well remember the first day. An emery-wheel explosion occurred. It was frequent in those days, just as frequent as crane runway accidents were, and those things that are not heard of any more.

The third day we had a fire caused by an overheated bearing. I discovered the fire when it first started, threw a bucket of sand on it, and put it out.

I still have distinctly a feeling of horror as I think of the days that I climbed on a rickety ladder, unsafe as unsafe could be, up to a high-speed shaft. In those days we oiled while the machinery was in motion, and the high-speed pulley on one side of me without any protection whatsoever, and couplings and collars on the shaft, had projecting set screws and bolts galore. There was absolutely no thought given to safety.

I also think of the possibility of a great financial loss through the neglect properly to educate a new man in a rather important and dangerous job. If that fire had not been discovered within the first five minutes it would have swept through the shop.

PROCEDURE OF A PRESENT-DAY EMPLOYMENT MANAGER

Contrast such methods with the employment program in effect in many shops today. In every well-managed institution there is an employment manager, not necessarily known by that title, but functioning as such, to whom a request to fill the vacancy of oiler would be referred. In those days if there had been a vacancy the assistant engineer would have gone to the gate and picked a man from the waiting crowd, and a crook of a finger would have put that fellow on the pay roll. The paymaster would have been informed that this man was oiling.

Today there would be a process of selection. Instead of the group hanging around at the gate, there would be a well-furnished employment office with a waiting room, and when the request for an oiler was received by the employment manager, he would go out and seek a man who had some experience or who was adaptable for that position, and he would also know in full the duties and just what specifications he would have to meet in getting the right man for that job.

If the man whom he chose were a foreigner that man would be interviewed across a desk by a man who could talk in his own language, and be invited to tell in full his past experience. If he lied about it, that would show up in the replies to the letters of investigation that are sent out as a rule.

I am not one of those who believe that the employment manager is the final judge of the fitness of a man for the job; that rests wholly in his performance on the job, and the foreman is probably the final judge there, but certainly the employment manager can by intelligent first sifting, by coarse sifting, get rid of many palpably undersirable applicants. The applicant who seems after that sifting to be the man for the position would then be given a complete physical examination, not necessarily to reject him if physically impaired, but to make sure that he would not be assigned to a job that would further injure him if he had any physical disability.

If he were acceptable after that physical examination he would probably be told something of the policies of the company; be given an introduction to his employer — the corporation — and be told of its safety program and handed a book of rules of conduct. He would probably be told that this rule book was made up by the Safety Committee, employees in the shop, and that if he did not fully understand it and wanted to quiz anybody on it he had only to seek the nearest old-timer who had been concerned in the revision of the book, and he would be given full and complete information. Undoubtedly, he would have greater respect for the rules when he learned that they were made by the men themselves.

Then he would be told of the promotional opportunities, — that it was the policy of the company to promote from the ranks, not to hire anybody at a given wage unless they were satisfied that nobody then in their employ at a lower wage could be promoted to fill that vacancy. He would be quizzed particularly as to his ambition. What did he want to be? He would not necessarily be put upon the job as an oiler permanently, but in an efficient filing system for recording applications his ambition would be registered. If he aspired to be a machinist and had stated that he had had certain experience as a bench hand and was attending night school, that would be recorded; and later, when a machinist was desired, preference would be given those applications from employees already in the service. Probably at that time the employment manager would send for this chap and say to him, "When you first went to work here six months ago you were put on as oiler. I understand you are still doing it, and you said you were going to night school to learn to read blueprints and take up manual training, and so on. Just how profficient are you as a machinist? Now I have a vacancy and you might be of use." And possibly he would be fitted into that job.

When he was hired there would be entered on his employee's record card a pretty complete history of him, his social condition, whether married or single, how many children he had, just how many dependents he had, where he was born, his age, his schooling, his military service, if any, whom to notify in case of sickness, result of physical examination and so on, for statistical purposes and then he would be sent to the job on sort of a personally conducted tour. He would be taken by an agent of the employment office, watchman, or special guide, and first shown the gate nearest his work so that any unnecessary hazard of traveling through a long plant might be avoided in coming to work.

In the meantime, if he were a single man he would be assisted in getting a proper boarding house. If he were a married man, temporarily located in a boarding house, the employment agency would probably show him a map of the vicinity of the plant and the desirable residential districts for his type, and give him a list of reputable real estate men as an assistance to him in locating permanently, and show him the street-car lines whereby he might best get to work. Finally, when delivered to the foreman for whom he was to work, he would be introduced by his own name and learn the name of his foreman. Nowadays, the worker, whether he be foreigner or American born, is introduced to his foreman by name and the guide makes sure that they understand each other.

The foreman probably repeats "John Sobrinski — did I get it right? Well, my name is John Smith. I am your foreman." And the last word of the personal guide is, "The labor supervisor particularly requests you, Mr. Foreman, to give this man any special instructions necessary in the particular hazards of his job." And then and there the foreman, either himself or through a fellow-workman, tells this man something about his job, and introduces him by name also to one or two of his neighboring workmen, so they will know each other, and the man can go to his fellow-workmen and seek and get information.

Some employment agents go still further, and during the process of hiring a worker give him a movie show picturing the complete operation of the plant — taking it from the receipt of the raw material, through the various manufacturing processes, on through shipping, just as general as it may be or just as complete as it may be in the space of time available, and increasingly that time available is lengthening.

We used to brag in employment management about hiring a man in a minute, or half a minute, or in a number of seconds, or about the number we could put to work in a year. But that has changed; now we are beginning to boast of taking an hour and a half, and even a week and a half to put a man properly on his job.

Now, it needs no more than an appreciation of that contrast to bring to mind how well sold a man may be on his job, how much he may be made to feel that he is a constructive part of that establishment, if he is put to work by the modern method, as contrasted with the former. If when he is put on the job the foreman calls his attention to the movie pictures of the particular operation that he is engaged in, and if it only be wheeling cinders away from the central boiler plant — if he is told at the time that that is the central boiler plant that furnishes the power to run the plant that does the things that he saw in the movies and that these cinders are the refuse from the boiler that generates all that power — then and there he links himself to the plant as a constructive force, as an important part of that machine. This is very necessary if we are going to settle this unrest that we now feel.

The unrest in industry today is evidence of a social, a political unrest that must be met freely, frankly and squarely, and corrected. In fact, it is manifested as Bolshevism in its ultraradical form. It is no less than that. Probably when the history of the Bolshevik movement in Russia is finally written, and we know just what its causes were, it may be found that a large contributing cause was the fact that the citizens of that country had no part whatever in their government. Things were done *at* them and *for* them, but never *by* them. They were never taken "in on the know."

While it is possible that their condition as citizens of that country under despotism was as good, their living conditions and all other conditions just as fine as they themselves might have done through democracy, they never had that feeling because they had not been consulted. It is simply a statement of basic psychology to say that we are only mildly interested in the things that other people are doing for us, but we are intensely interested in the things that we are doing for ourselves.

Going just a little further, if we have seen, as in the safety movement, the efficiency that may be gained through consulting the workmen themselves on such a matter of common interest, is it not perfectly logical to go to the men themselves for consultation on other problems of mutual interest — recreation, sanitation, health, and then the controversial matters of hours and wages?

It has been stated by Professor Hoxie that naturally the aims and purposes of employee and employer are antagonistic; that the employer seeks long hours and low wages as a means to a low cost of output, and that conversely the employee is constantly seeking shorter hours and higher wages, which must necessarily mean a decreased output.

There is abundant experience in the industrial world today to show that the reduction from twelve to ten hours in the length of time employees are working has not decreased production — that the reduction from ten to eight hours has been accomplished in factories without any decrease in the output, and that today reductions are being made from nine to eight hours without any decrease in the output. Shorter hours do not necessarily mean reduced output. Neither do higher wages necessarily mean increased cost, if they mean a higher standard of living, a better mental and physical development. I firmly believe that there is a common ground upon which employer and employee can meet for the consideration of their problem, and it need not be a controversial affair either in the consideration of wages and hours any more than any consideration of safety or health or recreation or plant canteens or anything of the kind..

THE INTERNATIONAL HARVESTER COMPANY'S PLAN OF EMPLOYEE REPRESENTATION

Our plan is known as the Harvester Industrial Council plan of employee representation, and was offered to the employees of the company on March 12 of this year. It was a frank invitation to them to participate in the determination of the policy of the company on all matters of mutual interest, including wages and hours, on an equal basis. They were invited to elect, by secret ballot, one-half of the membership of a works council whose function was to determine the policy of the company on the various items I have enumerated. The management appointed the other half of the membership. A guarantee of equal participation was had by the adoption of a unit ballot system. There are only two ballots cast. A majority of the employees' section determines their attitude and casts a unit ballot, as also does the majority of the management representatives.

These unit ballots have the same value, regardless of the number present on each side. The employees are guaranteed the right to a free performance and a free action in all of their activities as employee representatives. If there is any question of discrimination on their part, they may appeal directly to the president, and if not satisfied with his adjudication of the matter, it may then be arbitrated upon the selection of an arbitrator, mutually agreeable, whose decision would be binding upon both parties.

The plan was not put into effect at any plant which did not vote by considerable majority for it. It was first offered at the seventeen American and three Canadian plants. It was adopted at seventeen and failed of adoption at three. The day after the failure to adopt at these three plants petitions were circulated asking for another ballot and an opportunity of coming under the plan. The statement was made that the employees did not fully understand the plan on the previous day because it was written only in English. We have since published it in foreign languages because the employee representatives find it exceedingly difficult to convince foreign-born employees of the exact meaning of a certain clause if they themselves have to translate it.

An error was also made in the form of ballot. The ballot stated "For adoption. Against adoption." Some very good friends on the outside of the plant who were working against the plan told many foreigners to vote on the bottom line. When they saw the phrase "Against adoption" it did not mean anything to them and they voted according to instructions on the bottom line.

There was no positive effort on the part of the company to "sell" the plan. It was simply a dignified announcement. Each employee was furnished with a copy of the plan with a short résumé of a facsimile of the ballot, and asked to indicate, after three days of consideration, his wish for or against it. At the present time the plan is in operation in nineteen of the twenty plants of the Harvester Company.

Another fundamental included in the plan is the guarantee of the protection of the employee against any discrimination because of race or sex or membership in any religious body or labor organization.

FUNCTION OF THE WORKS COUNCIL

The function of the works council is limited to the determination of the policy of the company with reference to wages, hours, recreation, health, sanitation, restaurants, and other matters of mutual interest. A policy having been determined, its execution lies wholly with the management. But the manner of execution being open to question at any time, it may be brought up through the works council. In other words, we have given to the employees equality in participation in the legislative and judicial functions, but not with reference to the executive. That, we believe, still lies wholly with the superintendent and foreman, and it is further accented in the procedure in bringing matters before the works council.

The plan states that any employee desiring to bring a matter before the works council shall present it first to the secretary, who shall ascertain whether it has been presented to the superintendent through the regular channels. If this has not been done, he shall see that it is done promptly. If the adjudication of the matter by the superintendent is not satisfactory to the employee or employee representatives, it then and then only comes before the works council. It must be presented in writing by the secretary to all members of the council at least three days before a regular meeting. The decision of the works council is final and binding.

When it agrees upon a matter — and it can only function through an agreement because the two ballots cast in opposition to each other completely deadlock the decision of the works council — it is forwarded to the superintendent for execution. In case the council deadlocks, it is then in order to reopen a discussion or propose an alternative or compromise resolution. If the deadlock still continues the matter is then referred to the president of the company, the highest executive officer, who is given ten days in which to propose a settlement acceptable to the majority of the employee representatives.

THE GENERAL COUNCIL

If he fails to do that within the following five days he may elect to put it into arbitration direct, and arbitration is by mutual consent before a disinterested and non-partisan arbitrator, if one can be chosen. If not, each side selects one. If they agree, the matter is settled and their decision is final and retroactive. If they cannot agree and choose a third arbitrator, a majority of the three is binding on both parties; or the president may elect to throw it before a general council — provisions being made that by such a reference, or in the event that a matter is introduced into a works council which is common to more than one other plant, the president may indicate the other plants which are interested, and call a general council of those plants, whereupon the works council originating the proposition ceases its consideration of the matter.

In a general council the employee representatives of each of the plants designated by the president send at least two of their representatives. For a general council they select one for each thousand employees or a major fraction thereof, but in no case less than two. The president names a number of management representatives, not greater than the total of employee representatives, who function exactly as the works council in regard to the method of ballot, etc.

Provision is made that either council may be recessed at any time and the employee representatives and management employees be privileged to withdraw and canvass their factions. In a general council a recess may be taken to enable the plant representatives to consult with the other members of their works council. Provision is also made, in order to do this, that the superintendent shall convene the works council in whole or any part of the employee representatives for conference with the representatives who have been elected to the general council. The traveling, hotel and other expenses of the representatives are paid while in the performance of this work.

After consultation with their plants upon a matter pending, the general council may be reconvened, and decision is then binding upon all plants affected. Provision is made that all employees serving on the works councils shall be paid for the time they are so serving. They are privileged to call before them any employee of the plant to give testimony in a case under consideration, and the time of the employee so summoned is paid.

Provision is made, however, that in case it is not acceptable to the employee representatives to receive their money from the company, they are at liberty to arrange for a pro rata assessment of the employees directly. That was because of the objection which had been made by organized labor and other students of the subject to the general plan of paying employee representatives while serving as such.

Copies of the plan are available, and no less than 658 specific requests for them have come from other industrial establishments, from colleges and universities, from individuals, and from various sources.

RESULTS ACCOMPLISHED BY WORKS COUNCILS

We have been most agreeably surprised at the splendid results accomplished through these works councils since March 12, but surprised only at the rapidity with which they were accomplished, because there was no question in our minds as to the ultimate result.

There were elected by the employees 148 representatives in the 19 different plants. The average age of those employees is 38 years and 10 months. They are mature. One hundred and twenty-seven of them are married, and only 21 are single. Their average length of service is 7 years and 7 months, so they are relatively old-timers and mostly sedate married men. One hundred and two of them are native-born Americans, and 46 are foreign-born naturalized citizens.

Ninety-seven per cent of all of the employees who were present and eligible actually voted for or against the plan when it was proposed for adoption. The results at the various plants ranged from an almost unanimous vote for it to a vote of sixty to forty against it in two of the plants. One or two of the plants showed only a scant majority for the plan, considering the number of men involved, and we were somewhat puzzled at the time as to just what the result would be. For instance, at one plant the majority on the whole plan was only 200, and in one particular department — we will say the malleable foundry — the vote was 265 against the plan and 125 for it. And we thought that those 125 should be the only ones who would participate in the subsequent nominations and election, and we might then have a condition where only 65 of them would elect a representative for practically 400 men, which we could not feel was true representation. We were very much relieved, however, to learn that 98 per cent of all of the employees present and eligible actually voted at the nominations.

In the department just mentioned every man working on that day — nearly 400 of them — participated in the nominations, and again in the elections. On the final ballot for elections a nominating ballot was provided first in order that the elective representative might have at least a majority vote of his constituents. It can easily be seen that if we had a department of 300 men and only one elective ballot and there were, say, 30 candidates running pretty evenly, possibly a man with only 25 or 30 votes would win the election. The nominating ballot makes it sure that at least a majority will be had by the successful candidate.

The average vote cast for the successful candidates at all plants was 68 per cent of the total, so that the winning candidates represented, or were elected, by more than two-thirds of their constituents. With a single exception, every local plant superintendent has written in response to an inquiry that if he had been permitted to select the employee representatives he would not have been able to choose a more satisfactory and more truly representative group than did the employees by their own secret ballot.

The Harvester Company has always operated under the open-shop principle, but a number of union men have been elected as employee representatives and are serving on works councils. Our experience shows that these men appreciate as readily as non-union employees the constructive possibilities of the plan and there is no indication that their participation in the coöperative activities of the council is not fully as satisfactory as that of the non-union representatives.

One of the first results under the plan was, naturally enough, a demand at several of the plants for shorter hours and increased wages. As one old-timer said, it looked very much as if the company was giving a sort of a Christmas party when it passed around those booklets saying that the works council would determine wages and hours.

With a single exception these requests were withdrawn voluntarily by the employee representatives upon presentation of the management's side of the case, which was to the effect that this was not an opportune time for such action — that the agricultural-implement business was a competitive industry. The management was able to show that wages and rates were as high or higher than in similar industries in the vicinity, and that only through constructive work in this council, through a greater efficiency in the reduction of the costs, would it be enabled to pay higher wages and still remain in a competitive market. If the employees were willing to do their part the management would do its part, exchange figures with them and show exactly what conditions were at any time. When it would be felt that the time had come to consider it again, this would be done.

Thus far under the "Harvester Industrial Council" plan the works councils have been able to reach mutually satisfactory conclusions on all matters proposed with a single exception. That exception was in reference to a demand for a wages and hours revision, affecting about 25 per cent of the employees of one of the plants. The proposition as put up by the employee representatives did not meet the approval of the local management. After an extended discussion that was wholly friendly and frank, the ballot of the works council resulted in a tie. This was probably due to "an agreement to disagree," because both sides felt that the matter was one which could well be referred to the president for settlement and were entirely willing that this course should be followed.

Automatically the matter came before the president who was able to make a compromise offer to the employee representatives which met with their entire approval and at a special meeting, held four days after the original action had been taken in the works council, the proposal of the president had received the unanimous approval of the employee representatives and the matter was settled to the satisfaction of all concerned. In fact, it has resulted in a marked advance in the morale of the plant and the friendly attitude of the employees toward the council is doubtless more firmly established by the manner in which this matter was handled.

Probably 75 per cent of the actual business of the works council is transacted outside of the regular meetings. Many suggestions or complaints have been brought to the attention of employee representatives and referred by them direct to the foreman or superintendent, who have promptly cared for the matter to the complete satisfaction of all concerned. This has been particularly true with reference to correction of ventilating equipment, improvement of shop practice, more convenient time of the weekly payday, occasional reviews of piece-work rates and similar matters. This functioning of the council has been of especial value in acquainting the employees with the principles of time and motion studies, the reasons for adoption of certain shop rules, etc.

The employees have universally seemed appreciative of the opportunity to familiarize themselves with the facts as to any situation and have been exceptionally fair in passing judgment after complete discussion. They have displayed particular interest in production problems and seem to realize that the basis of larger returns to them for their labor lies in an increased production and more efficient operation of the shops.

Some of our plants are in the middle west side of Chicago, right in the heart of the radical district. Another in particular is out on the far south side, where probably you have read of race riots in Gary and Indiana Harbor and that vicinity.

I believe firmly that there is a well-organized propaganda on foot to start Soviet organizations in this country, and to play up the rebellious spirit in the foreign element in the vicinity of some of our shops.

The foreign members of one of our works councils desiring to meet this movement, asked for information which enabled them to get up a report showing the participation of the foreign-born employees in military service of this country. They drew parallel columns of the number of foreign-born employees per thousand of the men at the plant; and the number of each nationality per thousand who engaged in the military program. The average for the whole plant, of all the employees engaged who went into military service was 22 per cent. It was much higher at the other plants. The average of all the foreigners was 22.2 per cent, practically the same, but particularly the average of the Polish people against whom sentiment had been directed, and against whom a definite campaign was being waged in playing up their racial spirit, was 36 per cent. The average participation of the Poles was greater than that of the Americans. Then they drew some conclusions: that the foreigner, while he was not a citizen, and therefore not bound by the same ties as the American, had contributed almost the same to the military program, and individual nationalities to a greater extent than had the Americans; that his contribution had really been much greater from the fact that he was not a citizen, not bound by those ties; and they drew the conclusion that the anti-foreign sentiment which was being engendered was all wrong and not justified by facts.

Now, that was done in the works council committee on publicity without any instigation by the management whatsoever. It was the thought of the men themselves as a method of competing with this insidious and destructive propaganda.

RESPONSIBILITIES UNDERTAKEN BY WORKS COUNCILS

We had a restaurant which was not very satisfactory. It was losing money for us and not getting the results which we wanted. We turned it over to the works council and the patronage has trebled since that time. It no longer shows a deficit. It was frankly and freely turned over to the men to manage. They hire their own manager, they make their own rules, they set their own prices, and they have learned something about the restaurant business that we wanted them to know, and something of its trials and tribulations, and we have "sold" the proposition entirely.

There have been one or two requests for reinstatement of discharged employees. There are functionalized employment bureaus and all recommendations for men on the part of foremen are reviewed at the employment office. There are no restrictions in our plan as to what may come before the works council. In two cases — the only cases thus far that have been presented — the works council by unanimous vote upheld the decision of the employment manager and refused to reinstate the employee who thought he had a just grievance.

The safety program has been rejuvenated. We thought we were getting up efficient bulletins and we thought we were carrying our message all the way down the line. We thought that our safety program was satisfactory, and we couldn't go much farther with it. Since the inauguration of the works council, however, our accident-frequency rate has steadily fallen month by month. The character of the bulletins has changed somewhat. They are written in shop parlance, and by the men on the safety committees.

The sub-committee organization activities of the works council have been wonderfully well arranged by the employees themselves without any action of the management. They have unanimously said that they do not want a grievance committee. They do not want any committee on wages and hours. Those are subjects that they want to come out for a full discussion in the works council, and their reasoning was somewhat along these lines: They said, "If we had a grievance committee which only went to the boss on controversial matters, he couldn't help but get down on us a little bit, and believe that we never think of anything except how to make trouble. And we don't want to have that put up to just one or two men, we want the whole council to work on that; and then, too, we have this feeling that if a man comes to us with a grievance or a request for piece-work revision, or revision of prices or hours, we don't ourselves want to say to him, 'Well, you go over in the blacksmith department and see John Doe, because he is on that committee. We don't want to have anything to do with that.' Those are things they want us to get into, and if we do, we want to do it to our complete satisfaction."

The sub-committee activities, so far as grievances and hours and wages have been concerned, are always taken up by the council as a whole, and it has meant that no one or two individuals sat themselves up as business agents in the plant.

The question has been asked how much of this human relations problems lies within the field of the mechanical engineer. Engineering, to my mind, has been the consideration of an exact science, and these matters of personnel in the industrial sense are not reducible to stable factors at all. In the whole scale of industrial relations we are always dealing with the human being — a body with a mind, a creative force, and a soul, and therefore not reducible to any sort of stability as regards its valuation. You can't get up any sort of a formula which will solve a given condition, and repeat on it. And I don't believe the personnel problem ever should be treated as a science and coldly analyzed or formulated.

I believe that its direction and its constant progress should be made by men, not necessarily technical in their education, although we do need the assistance of every bit of science that can be brought to bear, but more particularly men who are given over to the idea of real service.

There are many employment managers functioning today in splendidly furnished, mahogany-lined offices, and who receive the requisitions from foremen for help and who analyze the jobs' specifications quite technically, and select by means of phrenology, and goodness knows how many other "ologies," applicants from the waiting line, if there is such a thing, or by scientific advertising, if the waiting line is not there, and possibly fill those jobs.

But I believe the work is best done by men who forsake the office and don't hire through an employment window but get out and interview a man across a desk, and look upon him as another human being, one who is coming into a strange land, into a strange company, and make him thoroughly at ease; who will sell him his job as a constituent part of a great assembly, who will look after him, not in an apparently disinterested way, but in an atmosphere of good-fellowship after he goes to work, and who will look upon his problems with a light of experience and know exactly, or nearly exactly, what his thoughts are, what his suspicions are — and those I do not think can be had by a science as well as by experience.

I think the employment engineer of today — the personnel manager of today — should first of all be chosen on personality. That is certainly more than fifty per cent of the necessary make-up. He should be of an engaging personality, a man who can converse with other men and get them in turn to converse with him; who has a sympathetic ear; who has a certain poise; and withal, who is a good, keen judge of human nature; and then on top of that, as much technical training as he may have.

No. 1693

THE STATUS OF INDUSTRIAL RELATIONS[1]

By L. P. Alford, New York, N. Y.
Member of the Society

At the Annual Meeting of the Society of 1912, a report was presented by the Sub-Committee on Administration, of which the late James M. Dodge was Chairman, and L. P. Alford, Secretary, on The Present State of the Art of Industrial Management. This report was replete with information upon the broad aspect of the management problem as it then existed in the industries of the country.

During the seven years which have intervened since the preparation of this report, the question has been studied from many different angles and has come to be viewed in quite a different light from that in which it was regarded when the original report was prepared. In consequence, the Committee on Meetings and Program appointed Mr. Alford a committee of one to prepare the following new report upon the subject for presentation at the Session on Industrial Relations at the Detroit meeting.

This report not only comprises a review of the new aspects of the problem which have recently developed, but also a historical summary of the progressive stages in the development of industrial relations since the period immediately following the Civil War. It has proved to be inevitable that after any great economic disturbance like that produced by the Civil War, or the present period of unrest following the world conflict in Europe, there should be unrest and uncertainty in the field of labor and employment; and it thus seems appropriate at this time to outline briefly the most important transitions which have occurred in this field, beginning with the time of the Civil War and including the situation which now exists, so similar in character, but greatly amplified.

IN the presidential address made to this Society in 1882 is the following remarkable statement of the responsibility of engineers in solving the problems involved in the relations of employers and employees:

> In singular and discreditable contrast with all the gains in recent and current practice in engineering, stands one feature of our work which has more importance to us and to the world, and which has a more direct and controlling influence upon the material prosperity and the happiness of the nation than any modern invention or than any discovery of science. I refer to the relations of the employers to the working classes and to the mutual interest of labor and capital.

[1] Report prepared at request of Committee on Meetings and Program.

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

It is from us, if from any body of men, that the world should expect a complete and satisfactory practical solution of the so-called "Labor problem." More is expected of us than even of our legislators. And how little has been accomplished.

2 Dr. Thurston was speaking of conditions as they were a full generation ago, yet his words are as true today as they were then. The world is looking to the engineers for leadership in these matters "and how little has been accomplished."

3 The topic of industrial relations is so complex and far-reaching that any treatment within the space of a professional paper must of necessity be restricted to certain aspects of the problems. So in attempting to outline the present position of the body of fact that is comprehended in the term "industrial relations," the scope will be limited to a survey of the more important developments of the last thirty or thirty-five years, and to a statement of a few of the more outstanding tendencies revealed by events of the immediate present.

4 To shorten the paper, certain of the supporting facts have been largely grouped in two appendices or are referred to in foot notes. Free use has been made of the writings and work of others and an earnest attempt has been made to give due credit, but in the examination of many sources of information it may be unwittingly that violations of courtesy have crept in. If such are discovered, the author hopes that the situation will be viewed leniently in the face of the difficulty of compressing into a limited number of pages a discussion of a subject of such complexity, and one on which such a tremendous mass of literature has been produced. Many have supplied information of value in personal letters and such assistance is gratefully acknowledged.

5 Before we can attempt to outline the status of industrial relations it is necessary to define what we mean by the term. To that end this statement is offered:

> *Industrial relations comprises that body of principle, practice and law growing out of the interacting human rights, needs and aspirations of all who are engaged in or dependent upon productive industry.*

6 It will be observed that this definition does not include directly the feelings of uneasiness and unrest among industrial workers, but views them as the expression of real, or fancied, rights or needs, and considers strikes and lockouts as but the assertion of the same or similar claims. On the other hand, the definition does include not

only the interests of employers and employees who are actually engaged in industry, but likewise of others who are dependent upon productive industry for the satisfaction of some of their needs, or for the safeguarding of some of their rights.

7 From the viewpoint that industrial relations comprises a body of principle, practice and law operating in productive industry it is a major subject for examination by this Society, for the members of The American Society of Mechanical Engineers are drawn largely from the great group of responsible executives in industry. It is clearly a function of this Society to enlighten its membership in regard to matters that so vitally concern manufacturing and production as the one under consideration.

8 Furthermore, now is a particularly appropriate time for such a study as we are in a period that promises to yield developments of the highest importance. In this respect the present is similar to the decade 1880 to 1889, during which time several of the important methods and practices that have been worked out in industrial relations had their beginnings.

9 To extend this comparison of times, the Civil War period was one of prosperity in America and Europe and of growth of labor organization. The years from 1865 to 1869, the half decade immediately following the close of the Civil War, are referred to in our history as the period of reconstruction. After 1870 there set in a long period of business depression which culminated in the financial panic of 1873 and continued until about 1879. During this time there was a decline in trade unionism. Beginning with about 1880 there was a recovery which brought business prosperity, high wages, a decrease in unemployment and an enlargement of trade unionism. During the 80's there were many hard-fought strikes, widespread labor unrest, and frequent and insistent demands upon the part of workers in industry.

10 At the present time we are witnessing the payment of higher wages than were ever before known in this country, there is a general feeling of uneasiness and unrest throughout our entire industry, labor is making many demands, and strikes are so frequent and widespread that it is doubtful if a single member of this Society has not at least been inconvenienced during the present year by the temporary cessation of some function of industry upon which he is in some degree dependent.

11 But this comparison carries further. During the 80's there were developed and put into use two practices expected to mitigate

or help solve the labor difficulties of that time, namely, profit sharing and methods of wage payment intended to give the worker a direct share in the benefits of increased production. At the present time we are seeing another heightening of interest in profit sharing and the widespread installation of the so-called shop-committee system. Thirty years ago and today are similar periods of experimentation in industrial relations.

12 So we cannot avoid the conclusion that times of labor unrest are productive of new plans and methods that attempt to satisfy the needs and harmonize the rights of those who are engaged in industry. And it is reasonable to expect that the period we are now in will mark the rise of new methods and practices that will properly belong in the classification of industrial relations.

13 Although the beginnings of the development of industrial relations took place during the 80's of the last century the situation in industry did not become acute in the United States until about 1905. That date fixes approximately the time when the evils of absentee directorate of large corporations came into prominence. Many of our great industrial consolidations had taken place before 1905 and the owners had put into effect a system of control and management arbitrarily determined in a few of our large cities, principally New York City, while the plants of these corporations were spread throughout the country.

14 It is frequently stated that the necessity for establishing industrial relations today is the growth of the factory system wherein all personal contact is lost between owner or manager and the worker. While this is true, it does not sum up the entire loss. In the system of absentee directorate there are other evils as well, and these taken together have set up situations where there have been clashes over the rights, needs and aspirations of those who belong to the class of employers and those who form the great group of employees. These losses in fitness for control may be stated in this wise:

a The loss of personal contact and relationship that formerly existed between the master and his skilled workmen and apprentices

b The loss due to the lack of personal knowledge of the work being done on the part of present-day directors and managers

c The loss due to the lack of personal knowledge of the tools and equipment used in production on the part of present-day managers *d* The loss of the direct oversight of saving and conserving materials and human effort on the part of present-day managers

e The withdrawal from productive work of the families of the directors and managers

f The loss of equality of living conditions between the families of the directors and managers and the workers.

15 The effect of these losses in creating a situation where there may be a clash of interests, and failure on each side to understand and appreciate the other, is brought home when we contrast the human relationships in the days of craftsmanship with those of the factory system. In former times the employer or master knew how to do all parts of the work himself, had in fact done so with his own hands, was in personal contact with all of the tools, equipment and materials used in his shop, had complete personal oversight of everything that was done, and held a relationship of almost father to son with his apprentices who, in many cases, lodged and boarded in the master's home and were served by the master's wife and daughters. Not only did the master instruct his apprentices in the requirements and skill of the trade, but he likewise set them the example of right living. All of the activities of the master opened to the apprentice as he became able to exercise them. There was a complete community of interest between the master, his family, his workmen, their families and his apprentices.

16 By contrast, in American industry today far too often the owners and directors live hundreds and thousands of miles away from the workers in their plants, under entirely different conditions, as varying as New York City and a Massachusetts mill town or a Middle-West factory community. All exact knowledge one of the other is lacking, there is no community of interest or purpose, and no assurance that a policy determined upon in the directors' room will meet the needs and rights of the workers in some far distant locality.

17 It is the creation of this situation of absentee directorate that has done much to focus attention upon the necessity of developing a body of principles, practice and law to satisfy the needs and safeguard the rights of all who are engaged in or dependent upon American industry.

18 Examination reveals six major lines of development amid the various methods, plans and systems that have been tried in seeking to work out better industrial relations. These are:

a Profit-sharing plans
b Methods of wage payment
c Methods and laws to reduce the hazards in industry and mitigate the effects of injuries and occupational diseases
d Employment management
e Declaration and enforcement during the period of war of three rights of workers, namely, collective bargaining, restricted hours of labor and the living wage. Declaration of these same rights and others in the Treaty of Peace.
f Systems for mutual or joint control by employers and employees.

19 A controlling reason for considering these six lines of development is their actual or promised permanence and widespread acceptance and application. For this same reason it does not seem pertinent to the purpose of this paper to devote space to welfare methods, industrial betterment, suggestion boxes or systems, shop gardens, factory bands, dances, ministrels, glee and athletic clubs, employees' loans, benefit associations, pensions and many other activities that properly classify under our definition of industrial relations. Without doubt successful applications of every one of these activities can be mentioned, and likewise, without doubt, experiences can be pointed out where they have had a beneficial effect in promoting satisfactory conditions and assisting to develop a "spirit of the organization." It is more than likely that many of these activities will always find a place in industry, but none of them seem to be a major line of development, and in fact all classify under welfare work or industrial betterment, which have fallen into disrepute because of the motive of charity or paternalism that has inspired them in many places.

20 The element of failure in all these agencies is the lack of removal of the fear of unemployment. For the fundamental cause of industrial unrest is the dread of losing the opportunity to work and thereby secure the necessities of life, or of being cut off from deserved promotion.

PROFIT SHARING

21 Profit sharing has been mentioned as one of the lines of development that become prominent during the period ending in 1889. Approximately 32 firms that put into effect some form of profit-sharing plan during this period.[1] A few of these were mercantile establishments, but all were large employers of labor.

[1] See Profit Sharing Between Employer and Employee, N. D. Gilman, 1889.

22 Not only is there reason to believe that these attempts were inspired by conditions of industrial unrest, but there is direct testimony to this effect in two cases. The Globe Tobacco Company, of Detroit, Mich., entered into an agreement in regard to its profit-sharing plan in 1886 with the district board of the Knights of Labor; and the plan of the H. O. Nelson Company, of St. Louis, Mo., which was announced on March 20 of that year, was an outgrowth of the great railroad strike, strikes in the building trades and the movement for the eight-hour working day.

23 Within the past six months a number of large industrial plants in this country have announced or put into effect profit-sharing plans, so we seem to be having a repetition of what took place during the 80's of the last century.

24 It is worth while to point out that although profit sharing was put into effect in a number of American industries a generation ago, it has never had widespread adoption. This fact may cause us to question its effectiveness as a promoter of good industrial relations. Several reasons are recognized for this situation: Payment of gains under profit-sharing plans are deferred and so lack an immediate appeal; the gains do not come directly nor entirely from the efforts of the workers, but are dependent upon the hazards of the enterprise; the amount distributed to any individual worker is equivalent to an increase of only a few cents an hour for the year; the amount shared by each person among the owners and managers is many times greater than that received by each workman.

25 Plans for stock participation have the same purpose and value as those for profit sharing.

METHODS OF WAGE PAYMENT

26 The pioneer work that has since yielded the science and practice of industrial management was performed during this same period of industrial unrest, that is, from 1880 to 1889. During this decade Mr. Henry R. Towne presented two papers before this Society that have been frequently referred to as the beginnings of the literature of industrial management. The first was read in 1886. In it Mr. Towne called the attention of his fellow-engineers to the need of a study of the financial and profit-making aspects of shop management. He urged his associates to become "economists" because the engineer is one who essentially effects economy. His second paper was presented in 1889 and in it was described a gain-sharing plan that Mr. Towne had applied in his own shop. The

object was to enable his employees to share the profits of the business, depending upon gains in efficiency as shown by careful accounting.

27 Influenced by his study of the epidemics of strikes during the same decade, Mr. F. A. Halsey originated his premium plan of wage payment that was disclosed to this Society in a professional paper read at the Providence meeting in 1891. In a recent letter written by Mr. Halsey to the author the activities of the Knights of Labor and the great street-car strike in New York City are mentioned as two of the events that influenced him to study the possibility of some form of wage payment that would reward the worker for increased effort and production.

28 This same decade was the period of the work of Dr. Frederick W. Taylor, which later on gave to this Society several papers outlining his system of shop management. It will be recalled that a part of his methods was a system of wage payment known as the "differential system."

29 Since that time several other methods of wage payment have been originated and developed, so that today we recognize some half a dozen that have more or less extensive application. Each one has been developed with the purpose of improving the relationships between employer and employee in regard to the division of the earnings and profits of industry, or to provide an extra reward for extra productive effort.

THE "SAFETY-FIRST" MOVEMENT AND WORKMEN'S COMPENSATION

30 About 1910 American juries began to award large sums in suits for personal damages, where the plaintiff had been injured by machinery or otherwise in industry. Employers turned to liability-insurance companies to defend these suits and settle with their employees, thus bringing an outside party between them and their workers. Appreciation of the hazards in industry and the hardships endured by incapacitated workmen and their families, not only during the period of recovery from the injury itself but perhaps throughout the life of the wage earner due to decreased earning power, gave rise to the "safety-first" movement and the enactment of workmen's compensation laws.

31 It is interesting to note that the safety-first movement was fostered largely by men residing west of Pittsburgh, many of whom were in the employ of large corporations where directorates were in

New York City. Furthermore it was from the outset a young man's movement. Here and there a few progressive firms paid generous compensation voluntarily before laws were passed making this practice compulsory. Mr. John L. Henning instituted such a plan in 1904 in a mining operation in Louisiana of which he was chief engineer. This is the earliest attempt of this kind in the United States that has reached the author's attention.

32 In the enactment of compensation laws the rights and needs of the community were recognized. An entirely different view of the legal relationship of employer and employee was taken than that which had existed before, and which was summed up in the expression "master and servant." The responsibility for the injuries was placed squarely upon the industry through the employer, and in boards and commissions, machinery was set up to make sure that employees would receive the compensation to which they were entitled. There are now 39 states which have compensation laws. The first was New Jersey and the effective date was July 4, 1911.

33 But the safety-first movement has yielded much more than this group of legal enactments. There has grown up a large body of practice in regard to safeguarding machinery and working spaces, establishing and operating first-aid rooms and factory hospitals, toward improving the heating, ventilation and sanitary conditions in factories, removing or mitigating conditions tending toward occupational disease, providing medical, dental and nursing service for the families of employees, establishing medical examinations before employment and at stated intervals thereafter, teaching personal hygiene to employees and their families, and, in fact, a complete new outlook in regard to the health and physical welfare of all who are engaged in industry. Through it all there has been a moral motive.

34 The movement has also brought into being two strong organizations, the National Safety Council and the Workmen's Compensation Bureau. In addition there is an association of physicians who are engaged in industrial practice.

35 The American Society of Mechanical Engineers has contributed to this development through taking the initiative in preparing several safety codes or standards. It is estimated that the safety-first movement of the past ten years has reduced the number of annual fatal accidents in the United States from 35,000 to 22,000 with a corresponding lessening of maiming and disabling accidents. This line of development is the most significant of all from the viewpoint of the interests of the community.

EMPLOYMENT MANAGEMENT

36 Since 1916 there has come into prominence what is now recognized as a new profession in industry, that of employment management. It comprehends in its broadest interpretation the establishment of all policies and direction of all of the functions having to do with personnel. It has taken over, expanded and developed the former work of hiring and discharging, has sought to reduce labor turnover and has fostered and directed those activities which are usually comprehended under the term welfare.

37 The oldest organized employment department of which the author has knowledge has been in operation about nineteen years in the plant of the B. F. Goodrich Company, Akron, Ohio. In 1907 Mr. H. F. J. Porter in a discussion of a paper presented before this Society called attention to the evils of labor turnover and outlined some methods that he had taken for its reduction. In 1914 Mr. Magnus W. Alexander presented a striking address before a convention of the National Machine Tool Builders' Association in which he gave statistics gathered from some twelve metal-working plants revealing the tremendous amount of the turnover of labor and its excessive cost. A small group of men in and around Boston, Mass., worked on this problem of employment for a number of years and as early as 1910 organized the Boston Employment Managers' Association.

38 But the movement did not gain headway until about 1916 when it was ready for the truly marvelous expansion that has taken place during the period of the war. The April 1919 issue of *Personnel,* the official Bulletin of the National Association of Employment Managers lists 27 employment managers' associations. In addition, a National Employment Managers Association was formed in May 1918. No other line of development in industrial relations has had the rapidity of growth of employment management. But the impelling motive has not been entirely that of fostering good industrial relations, although that result has come in many cases where the work has been well done. The major reason in the minds of most industrial executives in establishing employment departments has been to secure employees during the period of labor scarcity and to find out why men leave. Another impetus to the movement came through the action of the United States Government in insisting that such departments should be installed in plants manufacturing munitions, war supplies and ships. To

meet the demand for trained managers a number of colleges and universities established six-week courses in employment management under the direction of the War Department. Since the signing of the armistice several of these have been modified and put on what will probably prove to be a permanent basis.

DECLARED RIGHTS IN INDUSTRIAL RELATIONS

39 As a war measure President Wilson by proclamation created a National War Labor Board to establish principles and policies in regard to the employment and utilization of labor during the period of war, more particularly in war industries, and to set up machinery for considering and adjusting grievances. At the outset this Board declared three rights of labor: The right to organize and bargain collectively; the right to a limited number of hours of labor; the right to a living wage.

40 The Peace Treaty written at Versailles recognizes these three rights and several other principles that are of "special and urgent importance." Appendix No. 1 gives the text.

41 The recognition of these rights is a great step in the development of industrial relations, and they can never be abrogated in American industry. The same situation is reflected in British conditions as shown by the summary of the report of the Employer's Industrial Commission of the United States Department of Labor.

MUTUAL OR JOINT CONTROL

42 The sixth major line of development has to do with control in industry. Students of the present condition of unrest have pointed out that the fundamental is a "struggle for control," the opposing forces being the owners and the workers. The expression "industrial democracy" is frequently used as describing a state that is about to come, or is now being ushered in. The parallel between our political democracy and the expected industrial form is sometimes put in this wise: The slogan that gave birth to this nation and brought our political democracy was, "No taxation without representation." The parallel slogan expressive of the movement to bring industrial democracy is "No control without representation."

43 The method that is being followed to put this ideal into practice is the shop committee. It is a new development. Approximately 105 firms in this country have some form of representative

shop committee. The oldest was put into effect in 1903 and most of them were started in 1918. Bridgeport alone has 44 plants where this system is in force. The plan installed by Mr. H. F. J. Porter in 1903 in the plant of the Nernst Lamp Company in Pittsburgh was described by him in an article published in *The Engineering Magazine* in August 1905. The shop committee was made up of secretly-elected employees under the chairmanship of the shop superintendent. Conditions in the plant that might be bettered for the employees were discussed, and the committee evidently functioned as a safety-first committee as well as a representative shop committee.

44 Without doubt the success of the safety committees with which there has been some six or seven years of experience has paved the way in many plants for the representative shop committee. In fact, a prediction of this development is found in a paper presented before this Society in 1915 written by Mr. W. H. Cameron.[1]

45 A general classification of these shop committee plans yields three types:

46 The first sets up an organization roughly paralleling the Cabinet, Senate and House of Representatives of the United States Government. The Cabinet may consist of some or all of the directors or higher executives of the plant. The Senate may consist of all or a portion of the foremen, while the House of Representatives is a body secretly elected by the employees.

47 A second type divides the workers in the plants into divisions, each having a definitely determined and equal number of employees, not necessarily defined according to craft or occupation. Any one division may include employees from several trades and doing varying kinds of work. Each division secretly elects its own representative and these representatives coming together form the shop committee. This plan is simple, but is not adapted to industries where there is a high degree of organization or where the workers naturally divide into a series of recognized trades.

48 The third plan is the one adopted by the National War Labor Board and installed under its direction in a number of plants making war materials. Its essential features are outlined in the following official statement of procedure for election:

Shop committees shall be selected to meet with an equal or lesser number of representatives to be selected by the employer. Each department or section of

[1] The Attitude of the Employer Towards Accident Prevention and Workmen's Compensation, W. H. Cameron, Trans. Am. Soc. M. E., vol. 37, p. 905.

the shop shall be entitled to one committeeman for each one hundred employees employed in the department or section. If in any department or section there shall be employees in excess of any even hundred, then an additional committeeman may be elected, providing the additional employees beyond the even hundred shall be fifty or more; if less than fifty no additional representation shall be allowed. As an example: In a department or section employing 330 men, three committeemen will be elected; in a department employing 375 men, four committeemen will be elected.

49 In plants where shop committees have been in operation for some time a wide variety of topics has come up for discussion and determination. One grouping lists some forty-two different kinds of matters, of which only one was wages. The plans are too new, however, to estimate the extent of the effect they may have in developing good industrial relations. But a sufficient amount of experience has already been accumulated to indicate that the application of the ideas of joint consideration and control on the part of employers and employees will produce favorable results. To reach a workable basis the employer must voluntarily limit his own authority and agree to conduct his business by the rule of reason and even-handed justice as interpreted by the representative shop committee that he may set up. The plan seems to restore, so far as possible in a large-scale business, the simple and effective relationship that used to exist between the master and skilled workman, and which largely exists even today between the small employer and his half-dozen employees.

50 A rather more formal mechanism for establishing representation in industry is the protocol system worked out in the coat and suit trade in New York City, and which has been applied in a few other instances in similar trades.

51 From one viewpoint the rapidly developing movement to put shop committees into effect in the United States is a confession on the part of employers and those who have the responsibility for industrial enterprises that they have already lost some of their control. Under such condition there is a readiness to experiment. From another viewpoint it is an earnest attempt to find a basis of democratic coöperation in the control of industry.

52 A parallel movement has taken place in Great Britain leading to the proposal to establish Industrial Councils, through which the British Government may promote effective coöperation between the organized employers and workers, believing that representative government in industry will foster good relations.

DEVELOPMENT OF MOTIVES

53 After having sketched these six lines of development in improving industrial relations and in building up a great body of principle, practice and law, it is wise to examine some of the predominating motives.

54 One that came into play early and brought to the front those practices and activities summed up under the headings welfare work and industrial betterment was the motive of altruism. The successful manufacturer, taking the part of the autocratic benefactor, enjoyed the swelling of the heart and feeling of personal gratification that came to him when he arranged for the spending of money to provide conveniences and benefits for his employees, as bathrooms, restaurants, flower gardens, reading rooms, rest rooms, and the like. But workers are quick to resent favors if they are substituted for justice, and the welfare movement as such has very properly been discredited and has practically disappeared, although many of its activities have been retained but inspired by a different and proper motive.

55 The safety-first movement from its inception was influenced by engineers who saw the essential economy in preserving the life and limb of the workers. The employers translated this engineering economic viewpoint into the commercial motive "it pays." Selfish though this was and is, nevertheless it was neither charitable nor paternalistic, and because the movement had a firm basis in sound industrial economics and in morals it has met with deserved and widespread success.

56 But even this engineering-commercial motive is inadequate to provide the impetus for the developing of the industrial relations that we all hope for. It does not comprehend the interests of all who are dependent upon and must be served by productive industry; so many of the leaders of thought on industrial matters have declared and emphasized another motive, saying that it must predominate and prevail else we will never have the development that we need. This is the one of *service*. Mr. H. L. Gantt, in an article in the May, 1919, issue of *Industrial Management*, presents the motive of service as applied in his own consulting work. Once it becomes active we can hope for the working out of a new body of principle and practice in regard to industrial relations that will bring far happier conditions than any we have yet experienced.

INTERESTS IN INDUSTRY

57 With this development of motive has been a broadening of the recognition of those who are interested in the proper carrying on of industry itself. In the early days the expression "master and man" or the legal "master and servant" summed up these interests and relationships, but as industry developed and the clash of rights, needs and aspirations became apparent the familiar term became "capital and labor," used more particularly to denominate antagonistic forces. In fact, this term is used very generally at the present time, although it has been pointed out that there can be no conflict between capital as such and labor, for we cannot conceive of a man fighting with accumulated wealth represented in the physical means of production. So, to clarify thinking some authors have insisted that we should use the term "capitalists and laborers" instead of capital and labor. Point is given to this contention by a quotation from a speech delivered by President Lincoln before the Wisconsin Agricultural Society in 1859: "Labor is prior to, and independent of, capital — in fact, capital is the fruit of labor, and could never have existed if labor had not first existed. Labor can exist without capital, but capital could never have existed without labor."

58 The next development of interests brought consideration of the needs and rights of the public, which depends upon the proper carrying on of industry for much of its well-being. This thought brought a new grouping — "capitalists, laborers and the public." Quite recently in studies of industrial relations some writers have included five parties, namely: The capitalists who supply the materials and means for production; the laborers who supply productive capacity; the managers who provide direction and control; the community in which the industry is located and upon whose operation its welfare to a certain extent depends; and the public that purchases the articles and goods produced.

59 There is more than ample reason for including the interests of the community and public in any such classification. The laws regulating the hours of labor for women and children, for the elimination of hazards in industry, and for compensation for industrial accidents are but one expression of the rights of the community as well as the worker in the operation of industry.

60 Before justice can be framed in the form of law there must have been developed a body of general principles. It is evident that the principles underlying industrial relations are now in a process of rapid formulation. It is probable that before long our

courts will have to pass upon an increasing number of industrial-relations controversies. Such matters are justiciable today; they were not twenty years ago. We may look forward to a time when controversies in regard to such rights will become just as justiciable as any controversy in regard to property. It is possible that when this time comes it will be properly referred to as the era of industrial democracy.

61 Three tendencies in this development of industrial relations seem to be new though they are not novel.

62 The first is the acceptance of the motive of service, which on moral grounds declares for recognition of the rights, needs and aspirations of every one engaged in or dependent upon industry. It is the engineering viewpoint rendered unselfish.

63 The second is the willingness to consider workers in groups. By training and experience the engineer has only been willing to look upon workers as individuals. Two of the governing principles that have brought modern industry to its present heights are the division of labor which minutely subdivides the job, and the selection and adaptation of the worker which individualizes him and attempts to fit him to some particular task, tool or machine. This is the viewpoint of specialization which deals only with units.

64 But in industrial relations the workers must be considered in groups or in the mass. This is the viewpoint of the industrial psychologist as contrasted with that of the technical engineer, and the latter has been slow to understand that his methods of subdivision and specialization cannot be used successfully in dealing with the problems of industrial relations. Much has been said about fostering coöperation, but little progress has been made. A reason for this situation is found in the lack of understanding on the part of industrial executives that to build *morale* or the *spirit of the organization* their working people must be appealed to in the mass and not as individuals.

65 The development of the safety committees began to open the eyes of industrial executives as to what might be accomplished once employees as a body had a chance to express their desires and opinions. The present movement to establish shop committees will carry this experience further and into new aspects of the problems of industrial relations. The experience so gained will show the possibilities and advantages of discarding the individualistic viewpoint of the engineer. It will also bring the passing of arbitrary and autocratic decisions.

66 The third tendency is toward mutual or joint control, toward mutuality and the working out of representation. It is an expression of democratic ideals.

67 From the experience of the past and in the face of the tendencies and forces now operating we may confidently expect a greater development in industrial relations during the immediately forthcoming years than in any preceding equal period of time. May engineers accept their entire responsibility and perform fully their duty in working out proper solutions of the problems presented!

APPENDIX NO. 1

The general principles in regard to labor incorporated in the German Peace Treaty signed at Versailles and found in Part XIII, Section 11, Article 427, are as follows:

The High Contracting Parties, recognizing that the well-being, physical, moral, and intellectual, of industrial wage earners is of supreme international importance, have framed, in order to further this great end, the permanent machinery provided for in Section 1 and associated with that of the League of Nations.

They recognize that the differences of climate, habits, and customs of economic opportunity and industrial tradition, make strict uniformity in the conditions of labor difficult of immediate attainment. But, holding as they do, that labor should not be regarded merely as an article of commerce, they think that there are methods and principles for regulating labor conditions which all industrial communities should endeavor to apply, so far as their special circumstances will permit.

Among these methods and principles, the following seem to the High Contracting Parties to be of special and urgent importance.

First: The guiding principle above enunciated that labor should not be regarded merely as a commodity or article of commerce.

Second: The right of association for all lawful purposes by the employed as well as by the employers.

Third: The payment to the employed of a wage adequate to maintain a reasonable standard of life as this is understood in their time and country.

Fourth: The adoption of an eight-hour day or a forty-eight hour week as the standard to be aimed at where it has not already been attained.

Fifth: The adoption of a weekly rest of at least twenty-four hours, which should include Sunday wherever practicable.

Sixth: The abolition of child labor and the imposition of such limitations on the labor of young persons as shall permit the continuation of their education and assure their proper physical development.

Seventh: The principle that men and women should receive equal remuneration for work of equal value.

courts will have to pass upon an increasing number of industrial-relations controversies. Such matters are justiciable today; they were not twenty years ago. We may look forward to a time when controversies in regard to such rights will become just as justiciable as any controversy in regard to property. It is possible that when this time comes it will be properly referred to as the era of industrial democracy.

61 Three tendencies in this development of industrial relations seem to be new though they are not novel.

62 The first is the acceptance of the motive of service, which on moral grounds declares for recognition of the rights, needs and aspirations of every one engaged in or dependent upon industry. It is the engineering viewpoint rendered unselfish.

63 The second is the willingness to consider workers in groups. By training and experience the engineer has only been willing to look upon workers as individuals. Two of the governing principles that have brought modern industry to its present heights are the division of labor which minutely subdivides the job, and the selection and adaptation of the worker which individualizes him and attempts to fit him to some particular task, tool or machine. This is the viewpoint of specialization which deals only with units.

64 But in industrial relations the workers must be considered in groups or in the mass. This is the viewpoint of the industrial psychologist as contrasted with that of the technical engineer, and the latter has been slow to understand that his methods of subdivision and specialization cannot be used successfully in dealing with the problems of industrial relations. Much has been said about fostering coöperation, but little progress has been made. A reason for this situation is found in the lack of understanding on the part of industrial executives that to build *morale* or the *spirit of the organization* their working people must be appealed to in the mass and not as individuals.

65 The development of the safety committees began to open the eyes of industrial executives as to what might be accomplished once employees as a body had a chance to express their desires and opinions. The present movement to establish shop committees will carry this experience further and into new aspects of the problems of industrial relations. The experience so gained will show the possibilities and advantages of discarding the individualistic viewpoint of the engineer. It will also bring the passing of arbitrary and autocratic decisions.

Important Phases of the Labor Problem. 1918. (In Iron Age, v. 102, p. 1259–61, 1322–5. Nov. 21–28.)

Babson Conference on Coöperation. 1918. (In Amer. Mach., v. 49, p. 749–52. Oct. 24.)

Brophy, J. P.

Bringing Capital and Labor Together. 1918. (In Ind. Management, v. 55, p. 40–1. Jan.)

Causes of Labor Unrest Pointed Out by National Mediation Board. 1918. (In Elec. Ry. Jnl., v. 51, p. 333. Feb. 16.)

Channing, J. P.

Man Power. 1918. In Amer. Inst. Min. Engrs. Bull. 137, p. 963–9, May. Iron Trade Rev., v. 62, p. 1611–3. June 27. Jnl. Amer. Soc. Mech. Engrs., v. 40, p. 464–5. June.

Coöperation of Employers and Employed in Industry; Safety Devices and Measures: Coöperation the National Ideal: Series of Posters Issued by the National Industrial Conservation Movement. 1918. In Amer. Ind., v. 18, p. 26–7. May.

Fraboni, C.

Plan of Democracy in Business. 1918. In Iron Age, v. 101, p. 1402–4. May 30.

Good, E. T.

Wages and Profits in British Industry. 1918. (In Engr., v. 125, p. 47–8. Jan. 18.)

Government War-Labor Policy. 1918. (In Amer. Mach., v. 48, p. 489–90. Mar. 21.)

Haynes, G. H.

Small Industry in a Democracy: Abstracts. 1918. In Jnl. Amer. Soc. Mech. Engrs., v. 40, pt. 1, p. 533–8. July. Iron Tr. Rev., v. 62, p. 1607–10. June 27. Excerpts. Iron Age, v. 102, p. 77. July 11.

Horton, J.

Compulsory Arbitration is Opposed by Investigators of British Industrial Unrest. 1918. In Iron Trade Rev., v. 63, p. 255–7. Aug. 1.

Important Developments Relating to Labor and Capital. 1918. In Met. & Chem. Engng., v. 18, p. 395–5. Apr. 15.

Inspiring Harmony Between Capital and Labor. Y. M. C. A. conference at Lake Geneva, Wis. 1918. In Amer. Lumberman, 2252, p. 41–2. July 13.

Jennings, S. J.

Address of President to Local A. I. M. E. Sections. 1918. In Amer. Inst. Min. Engrs. Bull., v. 137, Sup. 11–13. May.

Kent, S.

Industrial Peace — First Essential in Winning War. 1918. In Automotive Ind., v. 37, p. 907–9. Nov. 22.

Lang, C. G.

Social Reconstruction Bound to Follow Victory. 1918. In Engng. News, v. 80, p. 596–7. Mar. 21.

Meeker, E. J.

What of The Labor Situation? 1918. In Engng. News, v. 80, p. 241–2, Feb. 7. Power, v. 47, p. 298, Feb. 26. **Discussion, Engng. News, v. 80, p. 472–3, Mar. 7.**

Stoney, G.
President's Address to the Engineering Section of the British Association. 1916. (In Engr., v. 122, p. 205–7, Sept. 8. Excerpts: Elec. Wld., v. 68, p. 870–1, Oct. 28.)

Weiss, G.
Bridging the Chasm Between Capital and Labor. 1916. (In Amer. Ind., v. 17, p. 22–3, Nov.)

Weston, G.
Securing Industrial Democracy. 1916. (In Elec. Ry. Jnl., v. 48, p. 1205–6, Dec. 9.)

Weston, J.
Industrial Democracy with Particular Reference to the Relations between Capital and Labor. 1916. (In W. Soc. Engrs. Jnl., v. 22, p. 125–30, Mar., 1917. Abstract, Elec. Rev. & Western Electrician, v. 69, p. 1011, Dec. 9, 1916. Discussion, W. Soc. Engrs. Jnl., v. 22, p. 131–53, Mar., 1917.)

1917 — Barnes, C.
Industrial Peace. 1917. (In Engr., v. 123, p. 35–6, Jan. 12.)

Berry, G. L.
Sharing the Responsibility of Industry. 1917. (In Inland Ptr., v. 58, p. 670–1 Feb.)

Changing Existing Standards; Statement of the National Industrial Conference Board Respecting National Labor Situation. 1917. (In Amer. Ind., v. 18, p. 33–5, Nov.)

Hodge, J.
How Great Britain is Meeting the Labor Problem. 1917. (In Factory, v. 19, p. 21–4, July. Ill. Plans.)

Industrial Councils; Report of Sub-committee of Government Reconstruction Committee. 1917. (In Engr., v. 124, p. 10, 13, July 6.)

Industrial Reconstruction. 1917. (In Illum. Engr., v. 10, p. 243, Sept.)

Industry and the Whitley report. 1917. (In Engr., v. 124, p. 169, Aug. 24.)

Kent, S.
War Problems. 1917. (In Engng. News, v. 79, p. 957–8, Nov. 22.)

Longridge, M.
President's Address, Institution of Mechanical Engineers. 1917. (In Engr., v. 123, p. 373–4, Apr. 27.)

Noxon, F. W.
Paradox of Unrest. 1917. (In Amer. Ind., v. 18, p. 15–8, Aug.)
Secret of Development. 1917. (In Engr., v. 123, p. 85, Jan. 26.)

Walker, P. F.
Ethical Tendencies in Modern Industrialism. 1917. (In Ind. Management, v. 53, p. 877–85, Sept.)

Wheeler, H. A.
Organized Labor and Business Must Accept Equal Obligations. 1917. (In Eng. Rec., v. 75, p. 219–21, Feb. 10.)

1918 — Alexander, M. W.
Elements of the Labor Problem; with Discussion. 1918. (In Textile Wld., v. 53, p. 5143–4a, May 4.)

Important Phases of the Labor Problem. 1918. (In Iron Age, v. 102, p. 1259–61, 1322–5, Nov. 21–28.)

Babson Conference on Coöperation. 1918. (In Amer. Mach., v. 49, p. 749–52, Oct. 24.)

Brophy, J. P.

Bringing Capital and Labor Together. 1918. (In Ind. Management, v. 55, p. 40–1, Jan.)

Causes of Labor Unrest Pointed Out by National Mediation Board. 1918. (In Elec. Ry. Jnl., v. 51, p. 333, Feb. 16.)

Channing, J. P.

Man Power. 1918. (In Amer. Inst. Min. Engrs. Bull. 137, p. 963–9, May. Iron Trade Rev., v. 62, p. 1611–3, June 27. Jnl. Amer. Soc. Mech. Engrs., v. 40, p. 464–5, June.)

Coöperation of Employers and Employed in Industry; Safety Devices and Measures; Coöperation the National Ideal; Series of Posters Issued by the National Industrial Conservation Movement. 1918. (In Amer. Ind., v. 18, p. 26–7, May.)

Fraboni, C.

Plan of Democracy in Business. 1918. (In Iron Age, v. 101, p. 1402–4, May 30.)

Good, E. T.

Wages and Profits in British Industry. 1918. (In Engr., v. 125, p. 47–8, Jan. 18.)

Government War-Labor Policy. 1918. (In Amer. Mach., v. 48, p. 489–90, Mar. 21.)

Haynes, G. H.

Small Industry in a Democracy; Abstracts. 1918. (In Jnl. Amer. Soc. Mech. Engrs., v. 40, pt. 1, p. 533–8, July. Iron Tr. Rev., v. 62, p. 1607–10, June 27. Excerpts. Iron Age, v. 102, p. 77, July 11.)

Horton, J.

Compulsory Arbitration is Opposed by Investigators of British Industrial Unrest. 1918. (In Iron Trade Rev., v. 63, p. 255–7, Aug. 1.)

Important Developments Relating to Labor and Capital. 1918. (In Met. & Chem. Engng., v. 18, p. 393–5, Apr. 15.)

Inspiring Harmony Between Capital and Labor. Y. M. C. A. conference at Lake Geneva, Wis. 1918. (In Amer. Lumberman, 2252, p. 41–2, July 13.)

Jennings, S. J.

Address of President to Local A. I. M. E. Sections. 1918. (In Amer. Inst. Min. Engrs. Bull., v. 137, Sup. 11–13, May.)

Kent, S.

Industrial Peace — First Essential in Winning War. 1918. (In Automotive Ind., v. 37, p. 907–9, Nov. 22.)

Lang, C. G.

Social Reconstruction Bound to Follow Victory. 1918. (In Engng. News, v. 80, p. 566–7, Mar. 21.)

Mehren, E. J.

What of The Labor Situation? 1918. (In Engng. News, v. 80, p. 241–2, Feb. 7. Power, v. 47, p. 298, Feb. 26. Discussion, Engng. News, v. 80, p. 472–3, Mar. 7.)

New Industrial Dispensation Must Be Made Labor Insurance. 1918. (In Engng. & Min. Jnl., v. 106, p. 318–9, Aug. 17.)
Of What Use is the Millionaire? 1918. (In Engng. & Min. Jnl., v. 105, p. 470, Mar. 9.)
Philadelphia Rapid Transit System Coöperative Plan Extended. 1918. (In Elec. Ry. Jnl., v. 52, p. 459–62, Sept. 14.)
Relations Between Capital and Labor in India. 1918. (In Amer. Ind., v. 18, p. 44, Mar.)
ROCKEFELLER, J. D. JR.
Relation of Capital to Labor. 1918. (In Amer. Arch., v. 114, p. 262, Aug. 28.)
Shop Stewards and the Coventry Strike. 1918. (In Engr., v. 125, p. 8, Jan. 4.)
Significant Changes in Business Management. 1918. (In Amer. Mach., v. 49, p. 191–3, Aug. 1.)
TIPPER, H.
Agreement vs. Bargaining. 1918. (In Automotive Ind., v. 39, p. 784–5, Nov. 7.)
Labor Representatives Lack Sufficient Power. 1918. (In Automotive Ind., v. 39, p. 835, Nov. 14.)
Lack of Confidence Between Employer and Employee; the White Plague of Labor Efficiency. Diags. 1918. (In Automotive Ind., v. 39, p. 625–6 +, Oct. 10.)
Which Side Are You On? 1918. (In Automotive Ind., v. 39, p. 669–71, Oct. 17.)
WARREN, L. L.
Propaganda in Industrial Relations. 1918. (In Ind. Management, v. 55, p. 197–8, Mar.)
Whom Are We Fighting? 1918. (In Amer. Mach., v. 49, p. 211, Aug. 1.)
1919 — ALLEN, R.
Industrial Unrest. 1919. (In Power, v. 49, p. 296, Feb. 25.)
ALWYN-SCHMIDT, L. W.
Organizing the Nation for Peace. 1919. (In Indus. Management, v. 57, p. 45–8, Jan.)
GILLESPIE, J. J.
Indifferentism: The Present Danger in Manufacturing. 1919. (In Indus. Management, v. 57, p. 38–9, June.)
TIPPER, H.
Social Surroundings Have Important Bearing on All Labor Questions. 1919. (In Automotive Industry, v. 40, p. 366–7, Feb. 13.)

BOOKS [1]

1912 — TALBOT, W.
Select Bibliography of Recent Publications on the Helpful Relations of Employers and Employed. W. Talbot, Cleveland, Ohio. 1912.
1913 — COMMONS, J. R.
Labor and Administration. New York. MacMillan. 1913. Relation of labor to scientific management.
NEARING, SCOTT.

[1] These books are not available in the U. E. S. Library.

Financing the Wage Earner's Family. 1913. New York. Huebsch. Discussion of wages and living conditions from radical standpoint.

SPARGO, JOHN.

Sindicalism, Industrial Unionism and Socialism. New York. Huebsch. 1913. An exposition of the radical attitude towards industrial relations, by a Socialist.

TRIDON, A.

New Unionism. New York. Huebsch. 1913. The attitude of the extreme industrial unionists (I. W. W.) toward industrial relations.

WILSON, WOODROW.

The New Freedom. Garden City, N. Y. 1913. Deals with the relations between capital and labor.

New York (State). Factory Investigating Commission.

Preliminary report. New York. State Factory Investigating Commission. 1912. 3 v.

Reports 2–4, published 1913–1915. 8 vols.

1914 — U. S. — Industrial Relations Commission.

First annual report for year ending Oct. 23, 1914. U. S. Govt. Printing Office. 1914.

U. S. — Mediation and Conciliation Board.

First annual report of the Commissioner of Mediation and Conciliation. Wash. U. S. Supt. of Documents, 1913–1914.

1915 — GRANT, LUKE.

National Erectors Association and the International Association of Bridge and Structural Iron Workers. U. S. Com. on Industrial Relations. F. P. Walsh, Chairman, Kansas City, Mo. 1915.

Rockefeller Foundation, New York City.

Information furnished by the Rockefeller Foundation in response to questionnaires submitted by United States Commission on Industrial Relations. New York. Rockefeller Foundation, 1915.

U. S. — Industrial Relations Commission.

Report on the Colorado strike. U. S. Commission on Industrial Relations, F. P. Walsh, Chairman, Kansas City, Mo.

U. S. — Industrial Relations Commission.

Report of the investigation of the wages and conditions of telephone operating. U. S. Commission on Industrial Relations. F. P. Walsh, Chairman, Kansas City, Mo. 1915.

1916 — American Labor Year Book. (Annual.) v. 1–. 1916–. New York.

Rand School of Social Science, 1916 +. A review of labor conditions in various countries, giving statistics, industrial relations, etc.

COHEN, J. H.

Law and Order in Industry. New York. MacMillan. 1916. Collective bargaining or the working out of trade agreements between employers and employees.

NEARING, SCOTT.

Poverty and Riches. Phila. Winston. 1916. A criticism of the wage system by a radical, recommending sweeping changes in the relation of labor and capital.

U. S. — Industrial Relations Commission.

Final report, including the report of Basil M. Manly, Director of Research and Investigation, and the individual reports and statements of the several commissioners. (Repr. from Senate Doc. No. 415, 64th Congress. Washington, Govt. Printing Office, 1916.)

U. S. — Industrial Relations Commission.

Industrial relations; final report and testimony. (64th Congress, 1st Sess., Senate Doc. 415, 1916.) 11 volumes. Full report of hearings of the Commission on wages, hours of labor, management, strikes and industrial relations.

1917 — American Academy of Political and Social Science.

Present labor situation. Phila. Publ. by Academy. 1917. On industrial arbitration of labor disputes.

1918 — HENDERSON, ARTHUR.

Aims of Labor. New York. Huebsch. 1918. Sets forth the policies of the British Labor Party and the British labor unions on industrial relations, reconstruction, etc.

1919 — American Academy of Political and Social Science.

Reconstruction Labor Policy. Phila. Pub. by Academy, 1919. Reconstruction and industrial relations.

DISCUSSION

The following excerpts from the general discussion on Industrial Relations apply to the preceding papers, Nos. 1692 and 1693. An extended account of the discussion of these papers was given in MECHANICAL ENGINEERING for July 1919.

FRED J. MILLER was of the opinion that the industrial problem resolved itself mainly into the establishment of a business relation between employer and employee of such a character as usually exists between any other classes of people who do business together — relations which must be mutually and permanently satisfactory to each side involved.

The employee must feel that he is doing at least as well as he could do with another employer in the same line, and the employer must feel that he is doing about as well as he could with any other body of employees.

A business relation implies, also, the right of either party to present a proposition at any time for a change in the terms of that relation. It should be agreed that in any negotiations regarding the terms of employment, either side may have the right to be represented by a representative of its own choosing. In these days of large corporations, the owners of a business have chosen, under the name of officers, superintendents, foremen, and so on, men who represent them in all negotiations concerning the business relation that should exist between employer and employee. Nobody disputes the rights of the owners to select such representatives of their own choosing, and if a business relation between employer and employee is to exist, the employees should have exactly the same right to choose their representatives. These may constitute a committee of their own number or they may be union representatives; but so long as they are chosen by the employees they should be treated as their representatives and allowed to state their case.

When an employer has sat down for a discussion across the table with representatives of that sort, whether they be employees of his own or union representatives, or what, and views have been expressed candidly and freely as man to man, it will often be found that many of the difficulties which may have been anticipated will disappear; in fact, that thereafter there is no difficulty whatsoever.

FORREST E. CARDULLO. We have at the present time two conflicting ideas of what constitutes industry. One of these is based on the old idea of competition between all men for the good things of life. It falls back on the proposition that the primary purpose of industry is to make money for the capitalist or owner. Incidently it furnishes a means of livelihood to the worker and in doing so affords the capitalist or owner an opportunity to secure a commodity which he desires, namely, service in exchange for wages. The owner has the primary rights and privileges and the entire control, while the worker is put in the position of competing with his fellow workers in order to dispose of a perishable commodity, service, susceptible only to the operation of the laws of supply and demand.

Now we are coming to a point of view which regards the primary purpose of industry as service to the community and contributory to the general welfare of the nation; while the general welfare of the capitalists, owners, managers and employees who are engaged in the industry becomes secondary.

Our present-day attitude of mind on questions of capital and labor is such that we regard industry as passing from the competitive to the coöperative basis; from the idea of the entire and exclusive control by the owner to the idea of joint control, primarily in the interests of the community. In this we base our ideas on the theory of a democratic government which demands equal rights and privileges for all men.

The defects of the competitive system we have all seen and will continue to see as long as it remains such, which it is liable to do for some time to come. They are (1) that labor is under no moral obligation either to the community or to the employer to give its best, or to coöperate for a joint benefit; and (2) labor believes that it is compelled to adopt forms of organization and methods of procedure which emphasize the fighting power of the union and which develop inefficiency rather than efficiency and a spirit of antagonism rather than a spirit of coöperation. The men in charge of the activities of the union are dependent for their support upon fostering, at least, a reasonable amount of friction, so that they may have something to do and may point with pride to their achievements. Before we can do away with this spirit we must substitute something which will assure to the workmen a larger measure of the good things of life and larger measure of control in the things in which he is interested.

This can only come about gradually and as the result of economic education of the workingman who must understand what he is to strive for, what kind of men he must elect to take care of his interests, and that efficiency must be the basis for any increase in his material welfare.

Otto P. Geier.[1] There is great need today for trained industrial physicians who will not be satisfied simply to be called in as specialists, but who will actually make a contribution to the solution of some of the very difficult problems that are facing industry today. When we take into account that there is, perhaps, a billion dollars' worth of loss in this country per year, at least half of which is preventable by proper medical supervision in industry, then the problem that looms up for solution by the industrial physician, and by engineers and managers, is worth talking about.

Industrial Relations means to me that feeling of responsibility which the farseeing management has; that in having large units of society, such as industry, in its charge it has all the responsibilities that a community has toward large groups; that it must exercise all the care in regard to the matters of health and sanitation and safety, and all the things that naturally come up, and perhaps the thing that has not been stressed today sufficiently is the matter of housing — the matter of living conditions.

Many of you know a man out in the field today, who is a per-

[1] Director Employees' Service Department, Cincinnati Milling Machine Co., Cincinnati, Ohio.

sonal service man, who is in overalls, and going from plant to plant in this country trying to find out what workmen are thinking about. I recently spent about five or six hours with him to find out, if possible, what his reaction had been as a result of about four months in overalls, working in coal mines, in steel mills, and doing the things that other men do who earn their money by the sweat of their brow.

Perhaps the thing that he stressed more than anything else was the fact that in spite of all the talk that we have had about sanitation, safety and medical examinations, we haven't even scraped the surface. We are not getting this message across to the workmen, and we haven't begun to realize how much we can do for them by actually caring for their health. Living conditions are the things that are making men start this unrest, and they are coming to the factory in the morning tired and miserable and ill at ease with everybody and the world at large.

Boyd Fisher[1] said that he was interested in presenting the social and economic justification for the type of plan which Mr. Alford had described and Mr. Young enlarged upon, whereby the struggle for control in industry might to some extent be compromised.

The problem of the maladjustment of capital and labor is of interest because such maladjustment reduces production. The public is interested because lessened production increases the cost of commodities; and the manager, engineer and capitalist alike are interested because profits depend on rate of output.

As to labor, the speaker said that it had occurred to him that one of the reasons why labor willfully restricts and limits output, although it is one of the partners of production, is because it is the one holding the least advantageous position in the apportionment of the production, and in formulating the rules by which that apportionment is made.

This is due to the fact that those who undertake industry pay off all of the other factors of industry. They pay rent and interest, they pay the makers of the tools and suppliers of material, they pay the managers and they pay labor, and then they keep what is left.

These factors, other than labor, are mostly beyond the control of those who undertake industry. The manufacturer cannot control rent, for example, and even his paid managers have a bargaining power fixed in part by market conditions and in part by their

[1] Consulting Engineer in Management, 362 Burlingame Ave., Detroit, Mich.

ability to stay out of the market if the price offered does not suit them. Labor, on the other hand, has no such advantage and has to come into the market without sufficient resources and reserves and take what the market conditions provide. So it has been, at least, in the past.

Inasmuch, therefore, as capital has to pay off all of the other factors at prices over which it has no full control, at figures that must be satisfactory to the other factors before the bargain is made, and does not have to pay off labor in accordance with a bargain that is satisfactory to labor, capital is often able to derive an advantage at the expense of the workers in industry.

Mr. Young has proposed a method whereby labor can advantage itself equally with the other factors, so that, in a sense, it can say whether it will or will not stay out of the market. It can protect itself and get a fundamentally right bargain, which is the first step toward the basic need of the public in relation to industry, namely, to increase production. For with such protection labor has no just grounds for attempting to restrict output.

L. W. WALLACE [1] said that labor problems have always existed and are likely to continue to arise as long as humanity is constituted as it is. He, therefore, wanted to sound this warning:

"I do not anticipate that at this time or at any future period there will be evolved a panacea that will forever solve any and all problems that may arise between employer and employee. This is no more possible than that a plan can be evolved whereby there will be no more wars between nations. Some form of industrial democracy on the one hand and a league of nations on the other will unquestionably be an agency of great value and influence, but these agencies within themselves will not eliminate labor troubles nor make impossible future wars. In fact, no instrumentality or agency will accomplish much unless there be behind them and disseminated throughout every fiber and thread the spirit of fairness, honesty and justice.

"In all sincerity the principle of the 'Golden Rule' must obtain: do unto your employee as you would have him do unto you. I believe it is the duty of the employer first to demonstrate that he is operating on this principle. It is his responsibility to engender into the minds of the employees perfect confidence, absolute warmheartedness and cordial respect for him.

[1] Director Red Cross Institute for the Blind, Baltimore, Md.

"It is also my conviction that no man will ever succeed as a leader of men and solve the industrial problems, unless he has a large store of human sympathy in his heart. Unless we are sympathetic, we are cold and indifferent to those matters that are nearest and dearest to men. We are apt to be impatient with human weaknesses, we are apt to make demands that are unfair and unreasonable, and we are apt to be cruel in our decisions and rash in our actions. Unless the principles enunciated are carried out, no satisfactory results will be obtained, whether the plan is a committee system, an industrial democracy system, or a House and Senate plan. The plan or the machinery whereby you are to operate is not nearly so important as the motive that prompts, the ideals that prevail and the sincerity that obtains."

Sam H. Libby stated that they had had a coöperative committee in operation at the Sprague Electric Works since last December. Ninety per cent of the 2000 employees voted to try the plan, which is much like that described by Mr. Young. In the Sprague committee, however, the elected members number about two to the hundred, with chairman, vice-chairman and secretary. Three members were elected to each of six sub-committees and on the sub-committees the management appointed three other members, making six members each. These sub-committees each selected a chairman, which in every case proved to be an appointed representative, showing the confidence of the men in the proposition. Mr. Libby described at length some of the practical workings of the committee.

Richard A. Feiss, as the result of a discussion introduced by Mr. Young as to the extent to which the question of management is an engineering problem, contended strongly that it is strictly such and that one's views should be big enough to make engineering stand for the things with which the engineer has to deal today rather than for the mere word "engineering" as defined in the old dictionaries. He contended that primarily mechanical engineering is production. The purpose of a machine is to produce and the machine itself must be produced in the process of manufacture. Many times an engineer has seen the best theoretically designed machine go to pieces in the hands of the average workman, because it had been forgotten that a man had to control the machine and its production, and that its successful operation could not be determined solely by the working drawings on which it is based. How, then, can one say that a problem affecting the man whose labor is assisting in production, or perhaps being replaced by a machine which is designed, does not present what is distinctly and directly an engineering problem? Again, we note that the running of two or three machines in a group is much more complicated than the running of one. It must be evident that the questions of the man and the machine cannot be separated and that their relationship is so essential in securing production that the question of labor and its management is distinctly an engineering problem.

All works councils and other plans to develop the relationship between the management and men are merely part of the general scheme needed to develop and improve the conditions of the worker in order to make him a more contented and better workman with a view to enhancing production. None of these plans in themselves will do.

The solution to th problem, which is variable from time to time, depends upon taking into consideration every factor with a clear vision as to one's own objects and the ultimate relationship to be established. It has been well said that there are no panaceas. It should be said however, from one point of view that there is one panacea, *viz.* truth. It is necessary to study and realize the truth on the part of the management and to give full publicity in every possible way to the truth in the plant and outside of the plant in order to educate workmen and public at large and bring them closer to the realization of the mutual problem evolved in industry. And the ultimate solution of the industrial problem depends upon our ability to see and to make others see the truth.

H. F. J. PORTER (written). In his admirable paper Mr. Alford makes mention of my article in the *Engineering Magazine* for August, 1905, which presents a fairly clear idea of my installation of a factory committee in the plant of the Nernst Lamp Company, in Pittsburgh, in the winter of 1903–4, which I understand was the pioneer installation of its kind in this country. In 1905 I presented a paper before The American Society of Mechanical Engineers entitled The Realization of Ideals in Industrial Engineering,[1] in which I referred to the merits inherent in enlisting the interest of the human element regarding matters affecting it in the management of an industrial plant. The basis for this paper was the

[1] Trans. Am. Soc. M. E., Vol. 27, page 343.

experience obtained with my shop committee in the Nernst Lamp Company.

When I took charge at the Nernst plant I decided to install the suggestion system used successfully by the National Cash Register Company, but disliking the method of handling all such features for the management, I determined to have the suggestions passed on first by a committee of the employees themselves and, therefore requested the latter to elect by secret ballot a representative to it from each department. The committee so formed elected its own chairman and secretary and all suggestions were collected from the boxes by the secretary and prepared by him for the committee.

Among the questions considered by the committee were the following:

Wage Payment, the premium system being finally adopted.

Permanency of Employment. The committee considered the taking on and laying off of all employees and succeeded in stabilizing the working organization, lowering the labor turnover developing functions now accorded to an employment manager.

Sickness and Accident Prevention, resulting in suggestions for safety and the appointment of a nurse in charge of an emergency hospital and a visiting physician.

Fire Drill. There had been a panic due to a false alarm of fire in the factory before I took charge, which led to the development of a factory fire drill. I found to my surprise after a canvass of representative factories of the country that further than developing a fire-fighting brigade for handling the hose, etc., no such thing as a fire drill existed anywhere. The fire drill at the Nernst plant was the first established anywhere, which at a given signal would take the employees out of a factory building in quick time.

Mr. Alford refers to the fact that my shop committee was composed of workmen and foremen and that the superintendent presided. The first committee was formed wholly of working men and women but as they were advanced to positions in the management and the employees reëlected some of them to office, some foremen were on the committee and the superintendent was made chairman. There was, however, a second committee representing the management composed of heads of departments and of which I was the chairman.

The second shop committee which I installed was in the Nelson Valve Company's plant in Philadelphia in 1907. Here we had a shop committee of workers only and an advisory board composed of foremen and the superintendent. These committees met weekly, first separately and then together. I think this arrangement will be found to give the greatest satisfaction.

Mr. Alford mentioned in his paper that it does not seem pertinent to devote space to certain activities that classify under the definition of industrial relations. I feel that he minimizes the importance of some of these features, particularly the suggestion system, benefit associations and pensions, which seem to me to be of as vital importance as profit sharing, methods of wage payment, the safety first movement, employment management, mutual or joint control, etc.

Scientific Management, always difficult to install in a factory under autocratic management, goes in as a matter of course under committee management.

Cyrus McCormick, Jr.[1] (written) I disagree profoundly with Mr. Alford's statement that "students of the present condition of unrest have pointed out that the fundamental is a struggle for control, the opposite forces being the owners and the workers." The idea underlying employee representation is not a question of class struggle, but rather of removing class distinction. It is a constructive endeavor to secure added benefits which are not granted by our present system, rather than a restricted effort to prevent the spread of ills incident to industrial warfare. It may, it is true, prevent the spread of these ills, but if it cannot at the same time ameliorate conditions, it must philosophically be regarded as a failure. The first two principles of employee representation hereinafter described prove these points.

Employee representation, if it is to succeed, must include a practical application of the following fundamental principles:

a There must be full representation for employees concerning working conditions, protection of health, safety, wages, hours of labor, recreation, education, and the like. This statement, of course, involves a recognition of the right of collective bargaining. It assumes that the workers, individually and as a group, are intellectually capable of maintaining their share in the joint discussion with chosen representatives of the management.

b Joint conference between men and management. It is not

[1] Works Manager, International Harvester Co., Chicago, Ill.

sufficient to grant the employees merely the right to discuss their own affairs. This discussion must take place with the management or its representatives. If any subject can be brought out into the open where it can be tested by the clash of men's minds and where honest opinions are exchanged in free and frank discussion, a happy solution of any debated point cannot be long delayed. The fundamental point is to get together; to talk things over; to debate them; and so to understand each other's opinions and points of view.

c Shop committees differ only in the basis of representation. The great point is to secure a sufficient representation to build the plan upon an essentially democratic foundation.

d Employee representation must include an easy channel of approach from the lowest workmen to the highest official in the company whereby the former can appeal to the justice of big minds. In this way it is to be hoped that personality can be reintroduced into industry without depriving industry of the efficiency of broad organization.

e There must be no discrimination on account of race, sex, political or religious affiliation, or membership in any labor or other organization. This is an essential tenet of democracy, and if it is not included in the groundwork of any plan of committee representation, that committee cannot succeed.

f Finally, there must be executive supervision for the plan. In this way the experience gained from day to day can be collected and coded into a working principle for the future. The handling of labor problems is becoming the work of specialists, and as the years progress the specialist in labor matters will be more and more important even than he is at present. Certainly this is not an engineering problem in the narrow use of the word "engineering," nor is it a problem for psychologists or sociologists. It is rather an attempt to translate humanity into the language of modern industry. The whole problem must be looked at, not merely on ethical, but also on economic grounds, and the future development of Industrial Relations will, it is believed, depend upon a satisfactory solution of this dual view of the problem.

MACK GORDON [1] (written). Although the shop committee and collective bargaining outside of trade-union control are comparatively new and untried methods in management, some conclusions can be drawn from the limited experience at hand.

If the worker does not understand clearly what are the functions of the shop committee, and what his part is to be, it will not have his confidence, which is essential.

How is this confidence to be built up? The management must make up its mind absolutely to turn over to the workers the right of bargaining with it on any matter whatsoever that pertains to the workers' interests, such as:

a Wages. The quantity to be produced for those wages. The method of determining what the quantity to be produced shall be, such as time study. Individual requests for increase in pay

b Hours of work

c Any rules or regulations affecting the conduct of the worker

d His right to a hearing in case of discharge.

More than five years' experience has proved that the workers are reasonable. As long as there is confidence in the committee and in the honesty of the management, they will remain so. They do not ask for impossible things and when they do it is only because of lack of knowledge as to conditions. A frank and honest discussion will soon dissipate the misunderstanding.

However, the mere existence of a shop committee that can present grievances, ask for increases in pay, changes in hours, review of discharges, etc., is not enough. When the committee finds something wrong and asks that it be remedied, in the most important cases, the conditions can only be remedied by a change of manufacturing policy, or of methods of management. The management must be willing and able to make these improvements. If it cannot, the mutual confidence that is so necessary will be dissipated, due to the lack of ability to satisfy the workers. It is in this phase of the situation that scientific management can help to solve the problem.

WILLARD G. ABORN [2] (written). In the first place the employer must be entirely convinced that the employee committees can and shall work out to the mutual benefit of all concerned. Fair-mindedness on the part of the employer and a willingness at the beginning

[1] Consultant, 226 Marion Bldg., Cleveland, Ohio.

[2] 619 West 113th St., New York City.

to go more than half way to convince his employees that he is earnestly endeavoring to bring about a mutually beneficial condition is a necessary preliminary to the successful inauguration of such a scheme.

There have been and always will be dissatisfied radical workers in every plant, whose influence increases or decreases in proportion as the fairness of the employer decreases or increases. Therefore, by all means, in instituting these committees, reduce the influence of these radicals by giving the rationals, who are about 80 per cent, entire freedom in the selection of their representatives; in other words, remove all possible suspicion of undue influence at the elections by arranging that they shall be directly under the supervision of the workers themselves or their representatives. Do not permit any official or foreman to in any way dominate, interfere, or even suggest, except as they may be called upon by the workers for advice.

Elections without nominations seem desirable and have worked out extremely successfully where used, in that conservative employees of long service in the plant have almost invariably been elected. Also the time required is much less and the result is really more democratic.

Department committees of three are also recommended for the reasons that three are more constructive than one and when representing their own department only have intimate personal knowledge on any subject affecting their constituents.

Welcome joint discussion on any matter pertaining to the plant — in fact, if necessary or desired, originate matters for discussion so that the committees may function and the workers' interest in the plant welfare be stimulated. Those who have had dealings with committees know that there is nothing much more dead than a non-working committee. Joint meetings of employer and employees' committee should be held frequently enough to keep up a live interest. Allowing the committeemen to honestly feel that they are originating and inaugurating matters for common good will tend to keep their enthusiasm alive and working.

Guy P. Miller[1] (written). Although the plan of Industrial Relations adopted by the Bridgeport Brass Co. has been in operation only nine months the results have been far greater than anticipated. About 100 meetings of committees have been held during

[1] General Manager, Bridgeport Brass Co., Bridgeport, Conn.

this time, at 25 per cent of which hours and wages have been discussed, and in no case has the conclusion reached been other than unanimous and entirely satisfactory to the employees and the management. Every employee has an opportunity to be heard whenever he has a grievance, which enables the company to adjust little things which cause annoyance and to explain other things to the satisfaction of the men. In order to get the greatest benefit out of these committees, the safety and sanitation work, as well as the work of the sick benefit association with the insurance features, have been turned over to them. Joint committees are also handling recreational and athletic activities in all plants of the company.

Every three or four months an evening meeting of all the committees is held with a dinner in our cafeteria, and I explain the general condition of the business and ask for expressions from the men, which has helped cement the spirit of confidence which has been established. No plan will be successful which fails to instill confidence, but as soon as the men are confident that the management will give them a square deal they will go half way.

Many feel that agitators may gain control of the committees and organize them as union representatives, but I am convinced that this danger is not to be feared as long as the management has the confidence of the employees, and when this is gone no plan can be successful.

P. J. Reilly [1] (written). The work of industrial engineers and employment managers must be supplemented by a new type of foreman — one who has been trained for his foremanship. Information that will assist foremen to administer their departments more effectively should be organized and presented to them in a form that will make the material readily available. The philosophy behind any rules of organization which foremen are expected to enforce should be explained so that they can enforce such rules without appearing arbitrary. Regular meetings for foremen should be held for the discussion of problems affecting the planning of work, the quality, quantity, and economy of production, the handling of new workers and the promotion of deserving older workers. Such meetings are effective in the enlightenment of foremen so that they are in entire accord with the management's policies in each of these fields. Much can be done in the development of foremen by

[1] Head of Personnel Division of the Retail Research Association, 225 Fifth Ave., New York.

relieving them from duty in their department for short periods so that they can work under the direction of the employment manager, the industrial engineer, or the master mechanic.

Opportunity should be given to foremen to attend special courses on industrial management, or to visit other factories that they may get the broadening which so many of our foremen badly need.

If the industrial leaders will give them the chance, the foremen will learn to treat help with a degree of human sympathy that will result in better team work. They will develop more patience, judgment and tact, and will eventually realize that men under them will produce best when their heads and hearts are in their work as well as their hands.

D. G. Stanbrough (written). With reference to the development of the personal relations, this is a matter of organization. The foreman, in the minds of the workmen, represents the management, and consequently, if we are to be successful in maintaining personal relations, it becomes necessary that we develop a high class of foremanship, and this instruction work must be done by executives familiar with the broad policies of the business and who have had the necessary practical experience to enable them to talk to the foreman from the foreman's viewpoint.

The foreman must be a man of the proper amount of personal kindness and should be able to understand the psychology of management. Such foremen will be found among every class of workers, or at least potential foremen of this caliber, and to develop such men it goes without saying that the factory executive must have the proper viewpoint.

I want to take particular issue with the closing paragraphs in the paper. I do not think that we can ever hope for any real measure of success if we are to consider workers in groups or masses. You cannot build morale by appeal to masses. I believe that the appeal must be made to the individual, through the organization line. The spirit of organization can be strongly built up if each individual worker feels that the management is conscious of his efforts and that the appeal is to him personally. Even in a large organization much can be done by the executive in knowing his men. At least he can know a few men in each department, and through them the spirit can be communicated to the organization.

John L. Henning (written). As Mr. Alford states, the loss of personal contact is one of the contributing causes of present unrest;

but if employers will use the engineer to the fullest extent and permit him to take some responsibility in handling labor questions he will, I believe, demonstrate that his so-called fault of dealing with labor from an individualistic viewpoint may be turned to a very good account. He will use a little more human sympathy in dealing with labor, which after all is simply a collection of human beings who inherently have the same "rights, needs and aspirations" as the employer and those dependent upon their product of industry.

Referring to Mr. Alford's conclusions as to the reasons for the apparent failure of remedies heretofore applied, I have proved to my own satisfaction, in a small way, that reasonable assurance of a permanent job is the greatest single weapon against unrest and Bolshevism. The moment uncertainty as to tenure of job and rates of pay is introduced into a workman's mind he is ready to believe almost anything, and you cannot expect a worker in any class to save for a home when he has no or only small assurance of the regularity and permanency of his work.

William M. Leiserson [1] wrote saying that the tendency of the engineers is to consider workers only as individuals, while the "industrial psychologist" holds that they must be dealt with in the mass. On the other hand, the economist, who has been studying labor problems and industrial relations for more than a hundred years, takes a broader view based on a study of the labor problem, its history, and its causes and manifestations under different systems of industry.

The economist, he continued, analyzes industrial relations and finds not one problem requiring individualistic or collective handling, but two sets of problems, one requiring dealing with each worker as an individual, the other necessitating dealing with wage-earners as a class. The first of these may be called the personal relations in industry, the second, the economic or governmental relations. The personal relations include such subjects as recruiting, selecting, hiring, training, placing and promoting workers, looking after their health, safety, comfort and welfare. These problems are individual problems; they are technical questions that must be decided by technical experts like the engineer or the physician. The second set of relations, however, has to do with bargaining, wages, hours, shop government and discipline. These questions are essentially controversial. They cannot be decided by technical experts.

[1] Working Conditions Service, U. S. Dept. of Labor, Washington, D. C.

They are matters of opinion requiring decision by a democratic majority.

Failing to catch the economist's fundamental analysis of the problem, engineers, employment experts and industrial managers are wont to group all industrial relations together and to include the fixing of just rates of pay in the managers' function. Accurate records of individual production and fairness and square dealing with employees, they think, will assure justice to the workers. This may be true as far as injustices between individuals within a plant are concerned under a general scale of remuneration that is already fixed, but it overlooks entirely what President Wilson has called the progressive improvement in the condition of the wage earner, the raising of the scale of all so that labor may receive a very much larger share of the product of industry than it gets today.

H. L. GARDNER [1] wrote outlining in detail the personal qualifications of the employment manager, his duties and the functions of the employment department. He said that the manager should be a broad-gage man, perferably a high executive or an officer of the concern, for which he preferred the title "service manager." He should be of sufficiently large caliber to "sit in" with the officers or a committee responsible for all relations between employer and employee; and being thus constantly apprised of general problems and accepted policies he can then organize and direct his department to coördinate with all other service departments and activities.

Interviewing is perhaps the keynote of his work, and with the right viewpoint and with practical knowledge of plant operations and conditions he can be of tremendous influence in building the right kind of working force. Continuing, Mr. Gardner wrote:

"One of the more abstract functions of Employment Management deserves detailed attention. If we are to offer a satisfactory substitute for the decreased personal contact in industry, the foremen are the channels through which we must work. To the workmen, the foremen are the personification of the manager and the company; it seems vital that such representation should harmonize with the general policies of the concern, yet too little has been done to develop this contact and to insure the desired results. To my mind, one of the most important services which employment management can render, and one which has a most important influence

[1] Manager Personal Relations Sec., E. I. DuPont de Nemours Co., Wilmington, Del.

on industrial relations, lies in securing real coöperation of foremen through understanding of, and sympathy with, employment methods of attack on the common problems of personnel and production. This may sound too theoretical, but it is, in fact, a tangible problem, and quite possible of surprisingly satisfactory solution.

"At the risk of repeating certain thoughts which I emphasized at the National Association of Employment Managers Convention in Cleveland, I would criticize the average employment department as too selfish. We too frequently devise theoretical solutions for the existing problems and impose them on the plant without sufficient sympathy for the other fellow's opinions and troubles. Complete success demands that the employment department thoroughly acquire the general plant viewpoint and merge its individual activities into the broader service program."

MARK M. JONES[1] (written). Functions in industry can be arranged according to many very interesting classifications. The five M's — money, methods, materials, machines and men — have appealed to me as a simple classification. Engineers will recognize that in our progress on the fifth M — men, we have from an administrative standpoint reached a point of about 30 in case we consider progress on the other four M's — money, methods, materials, and machines — as being at 70, with 100 the ideal.

Employment management as a part of Industrial Relations has as its object the administration of recruiting, selecting, placing, transferring, promoting, and releasing workers on an engineering basis. It seeks to apply the labor policy of the enterprise so far as it may affect those functions named. Its effectiveness is only limited by the strength of the belief of the management in the value and possibilities of such a service.

Employment management is distinctly a "service" function. It is an aid to production and a department for the purpose must always be operated with the proper understanding of the important part it plays in turning out the finished product.

A well-organized employment department surrounds the initial contact of the worker with those activities which influence him favorably and contribute definitely toward the final object of production. It provides a proper reception place and courteous treatment while the applicant is being studied. It studies him from mental, moral, physical, financial and social standpoints with the object of

[1] Director of Personnel, Thomas A. Edison Industries, Orange, N. J.

placing him where he can work to his own best interests. The employment manager knows definitely that an individual cannot work to the best interests of the enterprise unless he is working to his own best interests. The interests of worker and business are identical. From the standpoint of proper placement there is no divergence. From the standpoint of effectiveness, however, men must be weighed more carefully and a more exact method of so doing remains to be developed.

If during the coming months engineers will have in mind just one need of this field, namely, the same recognition and study of the fifth M — men — as of the other factors in production, they can do much in assisting American industry toward the great ideal of industrial nations, which is that of "increasing individual production."

Mr. Alford's Status of Industrial Relations is a timely summary of past and present, full of valuable data and epitomized in a warning to be heeded.

HARRINGTON EMERSON (written). I have been requested to contribute to the discussion by notes on Wage-Payment Plans as one of the problems of modern industry.

Wages are but a phase of a much deeper problem. On the one side are the necessities of life, to live morally, to have health, time for study and industrial improvement, and to have opportunity; on the other, what the earnings will afford, and again also the market rate for similar services. An empirical equilibrium is struck. Not as high wages as an ideal life would require, but often more than the earnings justify.

But what I recognize is that three conditions enter into work and wages: (1) The basic hourly wage, however determined; (2) the equivalent in output for the hourly wage; and (3) the special excellence of the worker. Of these the third, the individual excellence of the worker, is the most important.

The great truth, as yet only partially recognized, is that the superior worker is worth so much more than the average or inferior worker that any amount of care and supervision and all the extra pay required to secure him is a good and paying investment.

From this conviction it is evident that I am wholly out of sympathy with so-called profit-sharing, which repudiates individuality and makes of business a kind of providence that rains on the just and the unjust alike.

There is no connection between profit and skill and effort. The product of the highest skill may be sold at a loss, the product of malingerers may be sold at high profit.

The extra wage paid for individual competence is not a dole, a gratuity, a present. It should be a measured and full equitable compensation for a measured delivery.

There are current wages so low as to be dishonest. There are also current wages so high as to be dishonest.

Starting with a basic hourly rate, increased yield should command higher pay as long as unit costs drop. An increase in wages that increases unit costs (unless money is falling in value) is robbing the three other divisions of the community.

The problem always is so to lower costs as to increase output, to lower unit costs yet to pay more per hour, to give greater security and volume of investment yet to lower interest and dividend rates, so to compensate executives as to stimulate them to secure the best combinations of men, materials and machines, thus again lowering unit costs and adding to the common fund.

Fundamentals, not expedients, underlie all real solutions of the wage problem.

R. G. A. Phillips[1] (written). To my mind industrial relations problems came into being along with the "Big Business" idea and I think it well and timely that we give these various problems serious thought and study.

In our industrial relations work at the works of the American Multigraph Company the biggest topic of the day is our industrial democracy. Therefore, although it comes last in point of things done, it ranks first in order of importance.

In our case Industrial Democracy was no "spur-of-the-moment idea." It is a subject we have been studying almost as long as we have been in business. Our active interest dates back about five years — we have been all that time progressing toward our goal. It took about a year and a half of constant study before the final plan with its constitution became a fact. It took definite shape toward the end of February 1919, and finally on March 1 the plan was put up to the employees.

Since then we have had many incidents that have made us wish we had started sooner. For instance, I will refer to some of the minutes of the congress meetings and pick out suggestions that were approved by the senate as being wise and necessary.

[1] Vice-President, The American Multigraph Co., Cleveland, Ohio.

Educational Committee: Suggestion that certain employees be taught what the Multigraph does so they might do their work with greater understanding. (Assemblers and final inspectors).

Production Control Committee: Suggested rearrangements of departments that will produce greater results.

Suggestion Committee: Presented about fifty new ideas, the principal one of which will result in considerable saving in handling of tools, etc.

Sales Coöperation Committee: Working all the time with Sales Department. Pushing manufacturing and keeping up standards. Latest job the reduction of time taken to fill and ship foreign orders about 80 per cent.

Sanitation and Safety Committee: Always at work. Producing great results. Total suggestions adopted to date, about 65.

Recreation Committee: Managed several dances, an indoor baseball league, bowling league and all gymnasium classes.

Spoiled-Work Committee: Very active. Reported last week on correcting a condition that reduced scrap and increased production on a certain part about 150 per cent.

Shop Training: Managed all class work during season just finished. Promoted mathematics classes (two each week after hours) and a big general shop efficiency class that met every Friday night having an average attendance of 150.

Industry has got to begin to get all the effort it is buying. It should no longer be satisfied with the services of the hands of its workers — the brains, too, must be induced to participate. This matter of brain and hand cannot be commanded either — it must be more of a pull-together effort. We think we are headed that way through our Industrial Democracy.

C. B. Auel (written). It seems almost certain that the six lines of development listed by Mr. Alford will go a long way, if fairly universally adopted, toward lessening industrial unrest; but, they will hardly eliminate it since the fundamental cause for this unrest as admitted by the author is fear of unemployment, and methods of overcoming it have not been included. Some persons may point out in opposition to this statement that during the recent period of tremendous industrial activity, labor unrest was perhaps at a maximum; but, in answer to this it may be said this unrest was quite abnormal, due very largely to war conditions, and was moreover aggravated by employers practically bidding against one another in the labor market, with the very natural consequence that labor tended to oscillate to and fro, wherever wages were highest, or in the contrary event to insist on wages being brought to the high level.

One very important item the author has omitted from his list is "Americanization," which has been carried on by many corporations for a number of years past, principally through the medium of schools for their employees; but, fine as is the work already done by them, the task is so huge that it needs to be supplemented by greater efforts on the part of the various states if real headway is to be made. In Pennsylvania, for example, one authority has announced that with a population of 8,000,000, there are 1,500,000 foreigners over the age of 10 years and of these half a million cannot read or write English, while a third of a million cannot read or write any language, but what is worse than these statements is the further fact that the half-million that cannot read or write English increased to this figure from a quarter-million in the short space of 10 years. Illiteracy is a fertile field for the propagation of every kind of "ism" and it seems astonishing that a condition like this should exist in any state in our Union, and doubtless a similar situation exists in certain other states.

Dwight T. Farnham (written). At the close of the paper Mr. Alford opens a question whose just solution will, I believe, within the next ten years tax to the utmost the ability and resourcefulness of our financial, economic, ethical and engineering minds. The fact that the International Harvester Company had hardly installed their "industrial democracy" plan before they were deluged with recommendations for rate raises illustrates this trend. When their committeemen, however, were taken to the company's books and shown that the business could not continue if their demands were granted, unreasonable demands for the most part ceased. From the standpoint of human relations, then, the first general principle necessary in order to establish a sincere industrial democracy would seem to be the possession of profits which will survive the light of public opinion.

The fair division of the rewards is the question which we face, and it will call for the development of the general principles under which controversies may be adjudicated. The first step is perhaps the establishment of a fair return upon capital which is so invested as to be reasonably secure. The next step in logical order to insure stability would then be the determination of adequate sinking funds to make reasonably certain steady dividends from each sort of business. Setting up such reserves would take investment in industrials out of the gambling class and would make them attractive at a 6 or 7 per cent return instead of having to offer chances of 15 to 20 per cent in order to sell stock.

One of the rocks upon which profit sharing has in the past gone to pieces has been its call upon the workers to share in the sacrifice during dividendless periods. The other has been the determination of the share to which labor is entitled. The solution of such questions requires first the scientific analysis which the engineering mind is best qualified to make.

What Mr. Alford describes as the second great tendency in the development of industrial relations — the willingness to consider the workers in groups — I interpret as not so much a plea for an understanding of mass psychology as a plea for that knowledge which brought the ancient Greeks to the belief that he who understood life was all-powerful. Every age has had its interpreter of motives which actuate humanity. Today the human psychologist is an industrial psychologist. We applied the mind of the engineer first to materials, then to machines. Now we have reached labor.

C. E. Knoeppel (written). Through organization and specialization in industry, relationships have each year become more and more complex, so that today this great problem of human contact is by far the most important confronting us, and my prediction is that from now on Industrial Relations will be considered the keystone of the industrial structure.

I am, therefore, pleased indeed to see that The American Society of Mechanical Engineers is giving this matter of Industrial Relations the prominent place it occupies. If the engineering world does not give the subject attention, where will the initiative come from? From the workers? No. From the employers? No. I say "no" in both cases advisedly, because in the last analysis each side is generally suspicious of the other's motives when suggesting improvements having to do with human relationships.

The world-wide tendency toward socialism and revolution in ideas, if not in acts, is due not so much to desire for political changes as to a demand for economic changes. What the great masses of people want are homes, food, farms, clothes, jobs, wages which balance cost of living, participation, representation in affairs, opportunity for self-expression and development.

The gigantic convulsion the world is now going through seems to me to be a protest against the way modern society is organized, against much in the present plan of man-to-man contact. The purpose of the coming era, as I see it, is to find the right basis, and this the people the world over will do, regardless of the strife and bloodshed necessary to its accomplishment.

JOHN YOUNGER (written). I would heartily endorse the thought expressed in Par. 63 and 64 — that the workers should be considered in groups; and while it is true that the tendency is for their operations to be sub-divided to a point of intense specialization, yet a process similar to that of establishing workshop limits can be used and limits placed on such specialization so that the men inside these particular limits fall into groups.

The trade unions of England feel that the problem of employment is not solely for the employer to solve, but that they also have a share in the proper solution. With this object in view, Ruskin Hall was founded many years ago at Oxford, being practically a branch of Oxford University but reserved exclusively for students of all ages sent by the different trade unions.

This college has specialized on economic and sociological problems and questions of employment, capital and labor. These are studied, of course, as disinterestedly as possible with a slight natural bias towards the viewpoint of labor.

It is my belief that the training of men in this work is distinctly beneficial and cannot help but give more intelligent coöperation in the human problems of manufacturing interests, and I would suggest that the workings of Ruskin College and its results be studied very carefully by capitalists and laborers.

No. 1694

CENTRAL-STATION HEATING IN DETROIT

By J. H. Walker,[1] Detroit, Mich.
Non-Member

This paper discusses the general problem of the utilization of the heat ordinarily discharged to the condensing water in a central electric generating station. The impossibility of its complete utilization for the purpose of heating buildings and the difficulties in the way of even its partial utilization are pointed out, with particular reference to conditions existing in Detroit.

The development of the central heating system of The Detroit Edison Company is traced, showing how the use of exhaust steam for heating was abandoned in favor of live steam. The reasons why it is more commercially expedient under the existing local conditions to supply live steam to the heating system and to generate all electric current in the condensing stations are also fully brought out.

The latter part of the paper describes some interesting features of the central heating system in Detroit, such as the boiler plants, distributing system, underground pipe and tunnel construction, consumers' installations and meters. Special mention is also made of distribution losses, condensation return lines, and the method of transmitting steam through feeders at high velocities and with large pressure drops.

The paper concludes with a discussion of the advantages of central heating service and of the obstacles to its wider use and points out the possibility of operating individual plants in combination with the central plant.

ONE of the natural results of the grouping together of human beings in civilized communities is the existence, in our cities, of central plants for the generation and distribution of heat to surrounding buildings. The advantages of central heating service to the user, over the alternative of operating a heating plant in his own building, are comparable to those accompanying any other public service. To the community also a properly operated central heating plant is a distinct asset, commercially and economically.

2 Started in a limited way about forty years ago, the central heating industry has grown steadily, though not with the rapidity of some other utilities, until at the present time there are between two and three hundred enterprises operating as public utilities in cities of all sizes in most of the northern states and doing an annual

[1] The Detroit Edison Company.

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

gross business estimated at from ten to fifteen million dollars. The capital invested is perhaps thirty to forty million dollars. As media for distributing the heat, both hot water and steam are used, but because of the impracticability of metering hot water service and because of the better adaptability of steam to the requirements of the average building the trend of progress is toward the latter. Further development of the business now hinges mainly upon the possibility of the establishment of the proper relationship between the selling price and the cost of the service so as to insure adequate return upon invested capital. In many instances this relationship is not satisfactory at present, chiefly because the actual cost and the value of the service are not fully appreciated. The engineering practicability has been amply demonstrated; and that there is a pronounced economic demand for the service is beyond question.

3 The central heating business is closely allied with the electricity supply business and in most cities is carried on directly or indirectly by the electricity supply companies. The task of supplying the demand for heating service falls quite naturally to the electric company because of the partial similarity in the methods of production and distribution of the two commodities; and furthermore the ability to offer to its prospective customers both electric and heating service, with the consequent entire elimination of any sort of power plant from the customer's building, is of great advantage to an electric company doing business in a large city.

4 Other reasons for the combining of the two utilities are the economies in the consumption of fuel, and to some extent in the investment costs, which are sometimes made possible by the physical combination of the electric and heating plants.

THE UTILIZATION OF EXHAUST HEAT

5 The heat carried away by the condensing water in the central electric stations of the United States, equivalent to about 60 per cent of the total fuel burned by them, is one of the more obvious (although not by any means the greatest) sources of waste of the country's fuel resources. Like many other similar losses this one exists, not because its reduction is theoretically impossible but because it is seldom commercially practicable. But while commercial considerations have always dictated certain practices in the utilization of fuel in the past and will continue to do so in the future, the increasing cost of coal and the present impulse towards its conservation now direct attention to some of the fundamental problems.

6 This great quantity of heat, rejected at low temperature from the generating units, may be considered as a by-product of electricity supply and as such its rate of production will depend upon the rate of production of electricity, the primary product. Since neither electrical energy nor heat can be stored to any great extent, it is necessary for the complete recovery of the by-product heat that the demand for it be equivalent, hour by hour, and day by day, to the rate of electricity supply. The warming of the interior of buildings is a natural means of making use of this heat, but the great

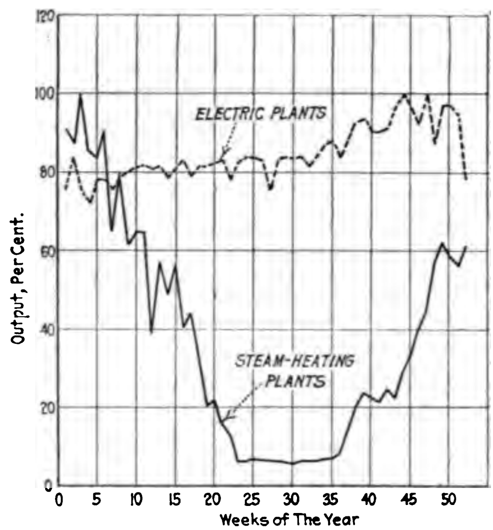


FIG. 1 LOAD CURVES OF ELECTRIC AND STEAM-HEATING PLANTS

diversity in the rates of use of the two commodities renders impossible even an approximately complete utilization of the by-product heat. The lack of agreement in the rates of use of electricity and heat from week to week throughout the year is illustrated in Fig. 1, in which the 1918 load curves for electrical and heating service in Detroit are plotted to a percentage scale with the maximum point on each curve taken as 100 per cent.

7 The possibility of establishing a better relation between the rates of use is rather slight. The rate of heat supply is largely fixed by unalterable climatic conditions. The use of electricity for lighting is also governed by natural elements. The demand for electricity for industrial use, which now constitutes a major fraction

of the output of most central stations, is not governed by these factors and is the only kind of load whose characteristics could conceivably be adjusted to suit the requirements for exhaust heat, but even this could probably not be done to any practicable extent.

8 Another important obstacle to the full use of this by-product heat through the warming of buildings is due to the great development of the central electric stations which in many industrial centers in the United States have so increased in size that the amount of exhaust heat which would be available as a by-product is greatly in excess of that which it would be commercially feasible to distribute for the heating of buildings. For example, in 1918 the central electric generating stations in Detroit produced approximately 774,000,000 kw-hr. of electricity. The exhaust steam which would have been available if discharged at pressures above atmosphere and which could have been utilized for heating, considering the winter months only, would have amounted to over 9,000,000,000 lb. This is five times the quantity actually distributed in the existing central heating system of the city, which entirely covers the only portion of the city in which the heating load is sufficiently dense to render the laying of distribution mains commercially justifiable. For because of the great investment costs the distribution of heat by the medium of steam is feasible only in the districts of relatively great density of load, which, in most of our cities, comprise only the business district and the very best residence districts. Nor are the economies to be gained by using by-product heat sufficient to enlarge this area appreciably. The central heating business in the average American city could keep pace with the electricity supply business only if the density of population were far greater than is compatible with present standards of living.

9 Thus it is apparent that the exhaust heat from central electric stations can at best be utilized for heating only during the times when heat is required and then only in so far as it can be commercially distributed. But there are certain obstacles which stand in the way of its utilization even to this extent — obstacles which arise primarily from the difficulty of transmitting steam over long distances.

SYSTEMS OF CENTRAL HEATING

10 Assuming that a central heating load exists and is to be supplied by the electric company, there are in general three methods by which this can be done:

a The heating load can be served from a condensing generating station so designed that steam is available for heating at pressures above atmospheric after partial expansion in the electric generating units, the remainder of the steam used for current generation being fully expanded and condensed at high vacuum

b Separate heating plants may be built in locations near the heating load and equipped with non-condensing generating units which will generate current only to the extent of the requirements for exhaust steam for heating, the remainder of the electricity being produced in a condensing station

c The heating system may be supplied entirely with live steam from boiler plants located near the center of the heating load.

11 It may so happen that the natural location for the main generating station serving a city is near the heating load, and if this is the case, the first method is preferable. In such a plant the use of bleeder turbines, designed so that steam can be extracted from the intermediate stages after partial expansion, offers great advantages. This arrangement has been successfully carried out in some instances and is probably the nearest possible approach to ideal conditions, since the duplication of equipment is reduced to a minimum.

12 Often, however, a consideration of land values or of railroad connections requires that the main condensing station be located at such a distance from the heating load as to preclude the possibility of transmitting steam from it. This is the case in Detroit as will be seen from Fig. 2. The Delray plant and the Connors Creek plant, the two main generating stations operated by The Detroit Edison Company, are respectively $3\frac{1}{2}$ and $4\frac{1}{2}$ miles from the heating district. A condensing plant, located on high-priced land near the heating district and with inconvenient railroad connections, or none, would be necessary if this first method were to be used. To such a plant, built for electricity supply and consequently, for reasons previously stated, burning more coal than a plant built solely for heating, this matter of proper railroad facilities is particularly important. Here again enters the matter of the transmission of steam, for seldom could the bleeder-turbine plant be located as favorably with reference to the heating load as could live steam plants, and the additional investment in transmission lines must therefore be charged against this plan.

13 The size of the pipes required to transmit a given quantity of steam over a given distance decreases as the density of the steam and the amount of pressure drop along the pipe increase. The most economical method from the standpoint of investment costs would be to extract steam from the high-pressure stages of the turbine; but the amount of electricity which could be generated per pound of steam would then be reduced. The relative values of these factors for an assumed river-front bleeder-turbine plant in Detroit are illustrated by the curves in Fig. 3. Curve *A* shows the total credit for the saving in coal and boiler capacity which could be allowed such a bleeder-turbine plant located on the river front and

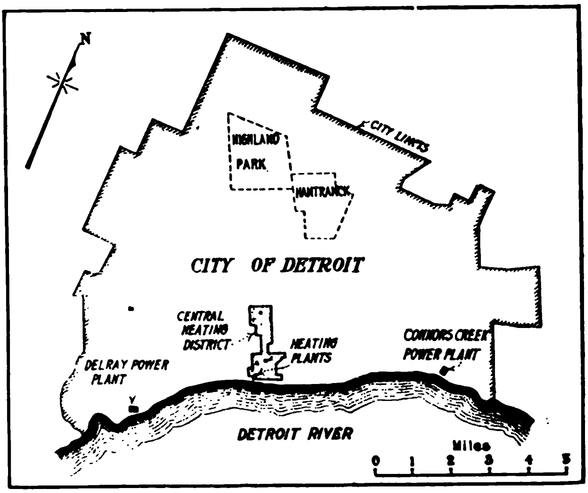


Fig. 2 Map Showing Location of Detroit's Heating District and Electric Generating Stations

serving the existing Detroit heating system. Curve *B* shows the total of the additional investment charges due to the higher-priced land on which the plant would be built, plus the investment charges and line losses involved in transmitting the steam from this river-front plant to the centers of the heating load. Even at higher extraction pressures the bleeder plant would not be justified under Detroit conditions because of its unfavorable location with respect to the heating load.

14 The second plan above mentioned, namely, the building of plants located near the heating load and generating current only to the extent of the exhaust-steam requirements, involves somewhat

different factors. In this case the capacity of the generating units will be determined by the heating requirements and the electricity generated by them will be the by-product. An essential requirement for the success of this plan is that the relation of these generating units to the electrical system as a whole be such as to allow their loads to be adjusted without restriction according to the momentary requirements for exhaust steam.

15 The value of such current as the heating plant will produce will be determined largely by the cost of producing an equal amount

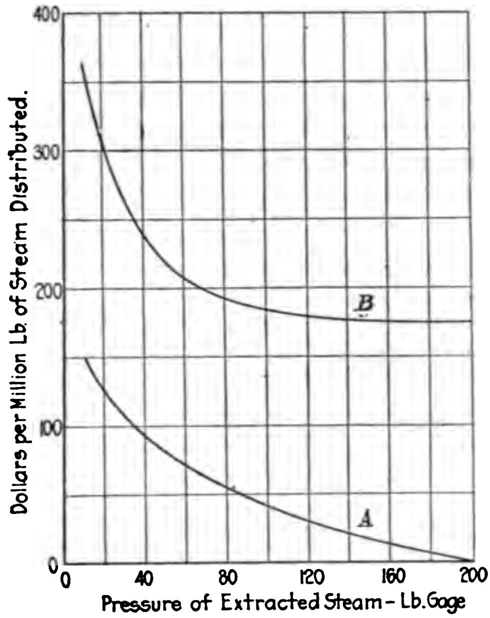


FIG. 3 CURVES SHOWING CREDIT (*A*) AND DEBIT (*B*) TO BLEEDER-TURBINE PLANT FOR VARIOUS EXTRACTED-STEAM PRESSURES

in the condensing stations. If the condensing stations do not produce current at an extremely low cost and, if other conditions are not unfavorable, this plan may prove very attractive and is in fact the most widely used method of combining the electrical and steam-heating plant. But unfavorable conditions surrounding the production of this relatively small amount of current may render its cost unattractive. The size of such generating units as might be installed in the heating plant may be so insignificant compared to the large size of the units in the condensing plant that the investment in the main station is not measurably reduced and the smaller units are therefore *additional* investment. The investment in steam-distribution mains is also much greater than when steam at higher pressures is used.

16 Thus it may in some cases prove in the end more commercially expedient to omit entirely the generation of current in the heating plant.

17 In Detroit many of these conditions conspire to make the use of exhaust steam for heating unattractive. Generating units exhausting into the heating mains were operated in one of the heating plants for several years but this practice has since been definitely abandoned and the heating system is now being supplied with live steam. Current generation is limited to the output of small house-service turbo-generators whose exhaust is utilized to heat the feed-water. The history of the central heating industry in Detroit is one of steady progress toward this method of live-steam operation.

DEVELOPMENT OF CENTRAL HEATING IN DETROIT

18 The immediate motive which, in 1903, led to the establishment of the central heating industry in Detroit was the possibility of obtaining a high thermal efficiency in the generation of electricity through the utilization of the exhaust steam. The generating plant from which steam was first supplied for heating had previously dfscharged its exhaust to the atmosphere. Owing to the building, at this time, by The Detroit Edison Company, of a large condensing plant, the smaller plant, with several others, would have been shut down, but the possibility of selling the exhaust steam made it appear desirable to continue the generation of current there to the extent of the exhaust steam requirements. Also, since the plant was located in a district served with direct current, the loss involved in converting an equivalent amount of alternating current received from the main plant to direct current would be saved for such current as might be generated there. The plant was in a high-class residence district and the heating service proved very popular, but the actual overall economy of the plant was not as great as had been anticipated.

19 A year later the construction of a central heating plant was begun in the business district of the city. This second project was undertaken, not as a means of disposing of exhaust steam, but for the express purpose of supplying the demand for central heating service among the owners of downtown buildings, who were con-

sidering shutting down their plants and purchasing electric service. The downtown plant was built primarily as a heating plant, and though provision was made for electric generating units they were never installed and the heating mains were supplied with steam from the boilers through reducing valves.

20 With the development of the large condensing generating stations the generation of current in the heating plants grew less and less attractive and when a third heating plant became necessary, because of increasing demand for steam heat, no provision was made for electric generators. The same practice was followed when, in 1916, the original exhaust-steam plant was rebuilt; and when the steam-distribution system of another company which had been engaged in the generation of electricity and the distribution of exhaust steam in the business district was purchased by The Detroit Edison Company, the new plant which was built to supply this district was also designed as a heating plant only. The entire combined distribution system is now being supplied with live steam from the boilers through reducing valves.

21 Since there are no generating units exhausting into the heating mains the pressure carried in them is not limited by considerations of back pressure. On those sections of the system formerly supplied with exhaust steam at from 2 to 5 lb. pressure, the pressure now maintained ranges from 5 to 15 lb. and is being increased from year to year, toward the upper limit permitted by the strength of the pipes. The pressure on the section originally designed as a live-steam system is about 30 lb. Because of the increased capacity of the distribution system at higher pressures, due to the greater density of the steam and the greater allowable pressure drop, these higher distribution pressures are desirable.

22 The steam is delivered to the network of mains through connections made at the plants and also through feeders radiating from the plants and delivering steam to certain "feeding points" in the distribution network, the method being similar to the feeder and main method employed in electricity distribution systems.

REASONS FOR USING LIVE STEAM IN DETROIT

23 The supplying of live steam to the heating system and the abandonment of the generation of current in the heating plants is commercially justifiable, even though such current would be generated at a high thermal efficiency. The underlying reason

for this is that there exist certain unfavorable conditions which outweigh the thermal advantage and make the total cost of such current higher than the generating cost at the large and efficient main generating stations.

24 In considering the cost of generating current, when the heat in the exhaust is recovered, it should be borne in mind at the outset that the amount of coal consumed, though small, is not by any means negligible as compared to that consumed in a condensing station. For each kilowatt-hour so generated there would be extracted from the steam, if the conversion were 100 per cent efficient, its heat equivalent, 3415 B.t.u. Taking into account mechanical and electrical losses in the generating unit and the losses involved in generating steam from the coal, the actual number of heat units devoted to the generation of electricity is not less than 5700 B.t.u. per kw-hr. This is 27 per cent of the corresponding figure for the Connors Creek plant (a condensing station) which in 1918 generated its total output at an average of 20,900 B.t.u. in the coal per kw-hr. of output. So that at the outset, the additional fuel which would have to be burned in the heating plants if current were generated, would be over one-quarter of the amount required to produce the equivalent amount of current at the Connors Creek plant.

25 One of the principal elements of cost which militates against current generation in the heating plants is the attendance labor, which, because of the low load factor and small size of any generating units which might be installed there, is much higher per kw-hr. generated than in the large stations where the size of the units and the load factor are much greater.

26 But the really deciding factors are the investment charges. The change of policy involving the final abandonment of exhaust-steam heating was made in 1916, with the rebuilding of the Willis Avenue heating plant, the original exhaust-steam plant. At this time the electrical load in Detroit was increasing rapidly and several new turbo-generator units were being purchased. At the Willis Avenue heating plant the existing and future exhaust-steam requirements would have called for a generating unit of about 2000 kw. capacity. The unit purchased at that time for the Connors Creek plant was the 45,000-kw. machine which has since been put into service. Would the existence of small generating units in the heating plants, aggregating altogether possibly 4000 kw., actually reduce the number of machines in the Connors Creek plant at that time or at any future time, if the latter were to increase in steps of this magnitude?

Regardless of theoretical considerations, as a matter of fact it actually would not have done so and it became clearly evident, therefore, that any investment in generating units in the heating plants must be reckoned as *additional* investment and the cost of any electricity generated by them must include the fixed charges on that investment and could not be credited with having saved any investment elsewhere. The relatively insignificant proportion of the total system

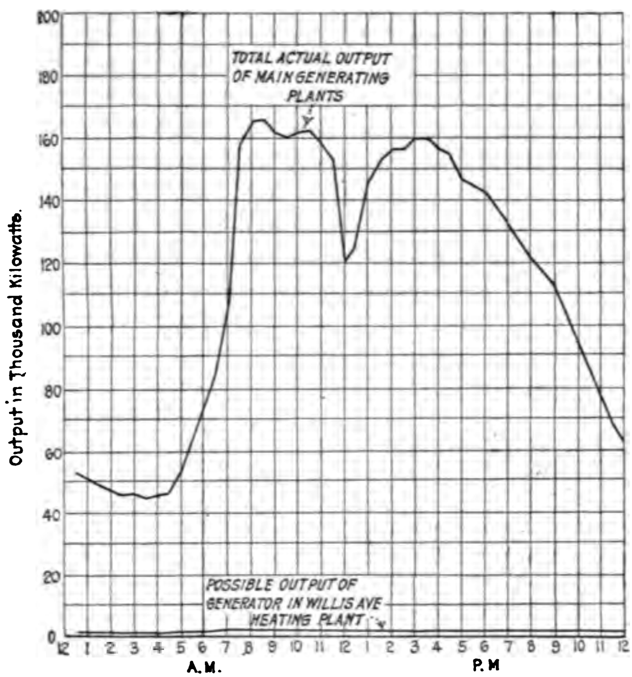


FIG. 4 LOAD CURVES SHOWING POSSIBLE ELECTRIC OUTPUT OF HEATING PLANT AS COMPARED WITH OUTPUT OF MAIN STATION

load which could be borne by a unit in the Willis Avenue plant is strikingly illustrated in Fig. 4.

27 The small size of the unit and its low load factor would make these investment charges relatively high per kilowatt-hour generated. Furthermore, while the chief advantage of the heating-plant generating units would lie in their ability to avoid conversion losses by generating direct current required for the downtown district, this advantage could not be completely made use of because of the lack of coördination between the heating and electrical loads of that district, which would make it necessary to convert some of the

direct current back to alternating current for transmission to some other district or else would reduce the load factor of the unit.

28 For a quantitative economic study of the subject the Willis Avenue heating plant offers a favorable example, for it had been operated as a generating plant and the distribution mains had been designed for exhaust steam pressures. Based on the performance of the old engine-driven units, there would have been generated by the 2000-kw. turbo-generator which it was proposed to install in the new plant about 6,000,000 kw-hr. per year.

29 In accordance with the considerations which have been mentioned, the proposed turbo-generator installation should be charged with its operating and maintenance costs including labor, supplies, and the fuel equivalent of the energy produced. It should also be charged with the fixed charges on the unit itself and on the building space occupied. Based on the fairly stable pre-war prices existing in 1916, these items might be conservatively estimated as follows, neglecting the disadvantage due to the lack of coördination between steam and electrical loads.

Charges Against Generating Unit — Operation and Maintenance:		
Labor — 3 oilers		$ 3,360
Supplies, etc.		200
Fuel cost — 1221 tons at $2.66		3,247
Maintenance		200
		$7,007
Fixed Charges (Additional Items Only):		
2000-kw. turbo-generator at $15 per kw.	$30,000	
Installation cost, wiring and piping	4,000	
Building and foundations	10,000	
	$44,000	
Depreciation at 4 per cent		1,760
Return on investment at 6½ per cent		2,860
Total charge against turbine		$11,627

30 The turbo-generator must of course be credited with the cost of generating an equivalent amount of energy at the main plant, of transmitting it to the heating-plant district, and of converting it to direct current, since these costs would be incurred if the turbo-generator were not installed.

31 The production cost of electric current generated by a central station consists of two parts, the demand or "readiness to serve" component and the "energy" component. The former may be defined as that portion of the total production cost which would be incurred if no electricity were actually delivered, but if the plant

were merely held in readiness to deliver the loads actually sustained —held, in other words, with steam pressure up, turbines and auxiliaries in motion, a sufficient number of boilers banked and the operating crew on duty. The energy component may be considered as that part of the total cost which is directly proportional to the amount of energy delivered. Although the exact separation of these components is impossible, the distinction between them is none the less real and is generally recognized.

32 In the present case the heating-plant generating unit can be credited only with energy component, since it is not to be considered as having reduced the size or number of units in the main generating plant. The energy component may fairly be considered as including 75 per cent of the fuel cost, 50 per cent of the cost of maintenance of plant equipment and that part of the labor, such as coal handling, which may be considered as depending upon the amount of electricity generated, and amounting in this case to 9 per cent.

33 The actual combined efficiency of transmission to the heating-plant district and of conversion to direct current is approximately 80 per cent, so that to deliver 6,000,000 kw-hr. of direct current to the heating-plant district there would be generated at Connors Creek 7,500,000 kw-hr.

34 The credit which can be allowed to the heating plant for the current which it would generate would then be as follows:

Fuel — 4112 tons at $2.38	$9,786
Wages	285
Maintenance	617
Total credit allowable	$10,688

35 This credit of $10,688 compared with the larger figure of $11,627 which is to be charged against the heating plant unit, thus indicating that the generating of current in the heating plant would not be justified under pre-war price conditions.

36 The foregoing study is based only upon conditions in the plant itself and no mention has been made of the effect upon the distribution system when exhaust steam is distributed. It is common practice in exhaust-steam systems to carry a back pressure of from 2 to 15 lb. At these pressures the specific volume of the steam is high and the allowable pressure gradient throughout the system is very limited, making it necessary to install much larger pipes than is the case when live steam at high pressures is used, and consequently increasing the investment. While it is true that

turbines have been built for exhaust pressures up to 30 lb., the electrical energy which can be extracted per pound of steam under such conditions is reduced and the cost per kilowatt of turbine capacity is increased; furthermore, in the method of feeder operation which is actually employed and which has proved of inestimable value, the pressure of delivery to the feeders is much higher than this.

37 The method of live-steam operation was adopted in Detroit before the recent great advance in the price of coal. The present high price of coal makes exhaust-steam operation appear somewhat more favorable and it is of course conceivable that at some future date a very high coal price may compel a change of policy. But with the cost of underground lines also increasing, the saving in distribution investment will probably continue to be sufficient to justify a continuation of the present methods.

38 Most of the foregoing facts apply only in cases where the generating capacity in the heating plant is negligible in comparison with the main generating stations. If this is not the case, if the discrepancy in size is not great so that investment in the main plants is actually saved, or if the additional current required from the condensing station is actually not produced at a relatively low cost, then the situation may be entirely changed, and there are many instances of this in the United States. The consideration of the steam-transmission investment, when low-pressure steam is used is often controlling, however, and is being increasingly well recognized.

DISTRIBUTION SYSTEM

39 The popularity of the heating service in Detroit has led to its development on an extensive scale. The present distribution system covers an area about 2 miles long and half a mile wide which includes the entire central shopping, business, and financial districts and a small portion of the residence district. About 2,700,000 sq. ft. of radiation, besides numerous water heaters and cooking fixtures, are served. The distribution system contains about 20 miles of underground mains and 2 miles of tunnels. The four boiler plants which supply steam to the system contain 17,470 rated boiler horsepower, and they delivered in 1918 nearly two billion pounds of steam to the system. Over 1700 consumers are served.

40 The distribution mains and the buildings served are shown in Fig. 5. Though originally built in three distinct sections, the

Fig. 5 Map of Central Heating System in Detroit

system is now a practically continuous network, and the plants are so much interconnected that the load can readily be shifted from one to another. In the spring and early autumn two of the four plants serve the entire area.

BOILER PLANTS

41 The four boiler plants which supply steam to the heating system are shown in Fig. 6. They are equipped with water-tube boilers and underfeed stokers and are of modern design throughout. Their location in the central district of the city necessarily restricts the amount of land which they may occupy and demands a suitable type of architecture, absolutely smokeless operation, and extreme cleanliness in the handling of coal and ashes.

42 A cross-section of the Congress Street plant, the newest of the four, is shown in Fig 7. It is designed to contain eventually four 1300-hp. boilers and two 2600-hp. boilers of the "W" type. In the effort to reduce the amount of attendance labor at this plant the auxiliaries are located, for the most part, on the boiler-room floor so as to be within convenient reach of the few men constituting the operating crew. Coal is hauled from bunkers at the railroad sidings to the plant in drop-bottom buckets of 5 tons capacity, which are lifted by a crane and emptied into overhead hoppers from which the coal is distributed by belt conveyors to the boiler bunkers. Ash-handling equipment is practically nil, the boilers being set at a sufficient elevation to allow wagons to be driven beneath the hoppers.

43 Because of the fact that only a relatively small amount of condensation is returned to the plants, careful treatment of the raw water is necessary. The feedwater flows through live-steam purifiers, operating at boiler pressure, in which the scale-forming materials are precipitated. In addition, sodium carbonate is fed in automatically graduated amounts to reduce the slight amount of hard scale-forming material which finds its way into the boiler. Although Detroit water is not a bad boiler water, these precautions are necessary because of the large percentage of make-up water and the rather high rates of steaming at which the boilers are sometimes driven.

44 The auxiliaries are almost entirely motor driven. The current is supplied by a 750-kw. turbo-generator unit exhausting into an open feedwater heater. The load carried on this generator is

Farmer Street Plant

Willis Avenue Plant

Congress Street Plant

Park Place Plant

FIG. 6 HEATING PLANTS OF THE DETROIT EDISON COMPANY adjusted according to the requirements for exhaust steam for heating the feedwater and any excess current generated is delivered to the outside electric-distribution system. Conversely, if the electricity requirements of the plant exceed the output of the generator, current is drawn from the outside supply. Exhaust steam is thus made available for heating the feedwater and the advantages of motor-driven auxiliaries are also secured. Moreover the turbo-generator constitutes a source of electricity supply for the plant in case of local interruption of the outside service.

45 Steam is generated at a pressure of 130 lb. gage. This pressure is chosen in order to provide for considerable pressure drops in the outgoing feeders as calculated for present conditions. It may be raised to 160 lb. at some future date. The outgoing steam lines leave the plant through a tunnel shaft.

FEEDERS

46 Because of the great increase in connected load the transmission capacity of the distribution network, most of which was installed several years ago, is now quite inadequate. To have raised the pressure throughout the system would have increased its capacity; but the system pressure was permanently limited by the fact that most of the underground fittings are of a low-pressure pattern, and temporarily by the fact that in those sections of the system formerly supplied with exhaust steam the customers' installations are not provided with reducing valves. Instead of attempting to change these conditions, which would have involved the reconstruction of much of the distribution network, the less expensive plan was adopted of running feeders from the plants to various centers of distribution. In selecting the pipe sizes for feeders, advantage is taken of the difference between boiler pressure and distribution pressure, and the size of the pipe is so chosen that at times of maximum load much or even all of this pressure drop will take place in the pipe itself. The diameter and the cost of the pipe line are thus materially reduced. In operation, the pressure of the steam delivered to the feeder is raised or lowered as required by the adjustment of a reducing valve at the plant, in order to maintain a constant pressure at the center of distribution which the feeder supplies. The pressure existing at this center of distribution is recorded at the plant by a long-distance gage, electrically operated.

47 The velocity of the steam flow in the feeders at times of heavy

load is very high. Velocities as great as 75,000 ft. per min. have been measured. This high velocity does not appear to be at all objectionable, however, there being no apparent erosion of the pipe and no hammer, or objectionable vibration. The fact that the steam is in a superheated state because of the pressure drop is an

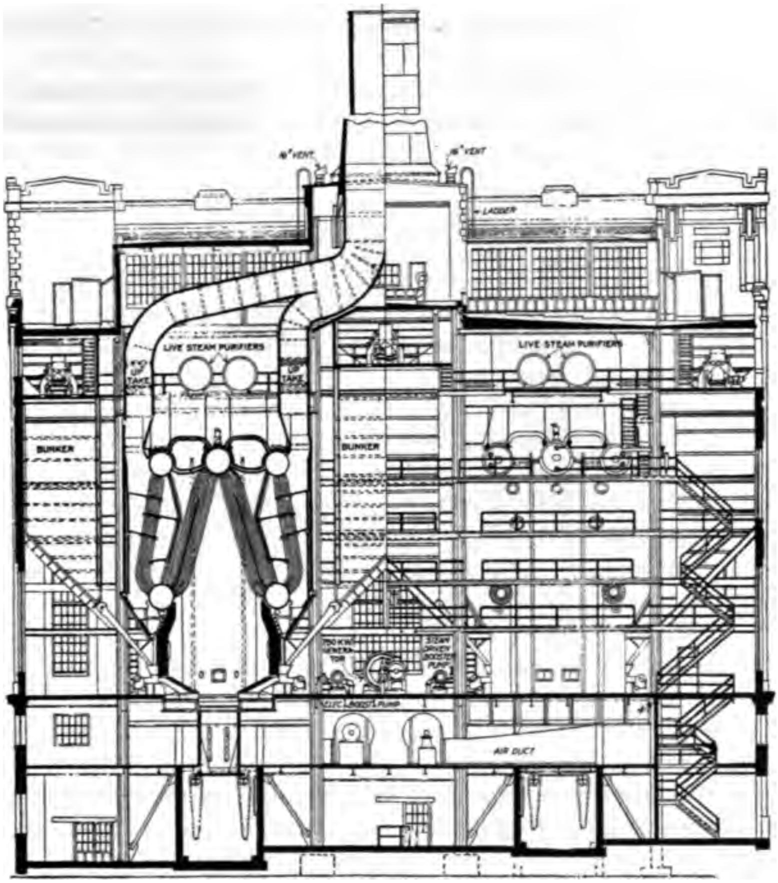


FIG. 7 CROSS-SECTION OF THE CONGRESS STREET PLANT

advantage in these respects. Feeders are constructed with long radius bends and where the connection is made to the distribution mains the diameter of the pipe is gradually increased by special taper fittings so as to reconvert a portion of the velocity head to static pressure.

48 A considerable pressure gradient is also allowed to take place in the distribution mains as well as in the feeders; but this is

relatively small, since the upper limit is fixed by the allowable working pressure of the older mains. The more recently laid mains are capable of withstanding a working pressure of 125 lb. and the gradient in the mains can therefore be increased at some future date. From a standpoint of safety, however, the desirability of carrying pressures in excess of about 50 lb. on the service connections to buildings is questionable until further development in pressure-reducing apparatus is made.

49 This method of steam distribution is obviously an adaptation to previously existing conditions and would doubtless be modified if an entirely new system were being laid out.

UNDERGROUND CONSTRUCTION

50 The distribution mains range in size from 20 in. near the stations to 4 in. at the outskirts of the system. The original underground pipes were laid in a segmental wood casing bound with wire. This construction has been fairly successful under favorable soil conditions, but the concrete conduit shown in Fig. 8 has been found to be superior in many respects and has been used exclusively for several years in all new construction. In this construction the pipe is insulated with a standard thickness of pipe covering and surrounded with an envelope of concrete poured over a wooden form, leaving an air space around the pipe. Proper underdrainage is of course essential.

51 The longitudinal expansion and contraction of the pipe, due to changes in its temperature, are absorbed, in the earlier construction, by means of expansion fittings of the copper-diaphragm type. In recent construction a slip joint, consisting of a brass sleeve, sliding in a packed gland, has been used. The use of slip joints decreases somewhat the cost of the pipe line as they will absorb more travel and can be placed at wider intervals than the diaphragm fitting.

52 The underground mains have not been laid sufficiently long to permit of an accurate estimate of their life. The oldest lines laid in wood casing have now been in service 15 years and, while the casing in many places has deteriorated considerably, in other places it is in fairly good condition. The concrete construction, dating back 10 years, seems to have deteriorated very little. The only repairs or replacements which have been made to date have been made necessary by external corrosion of the pipe arising from

some outside and local cause such as a water-pipe leak. An average life of 20 years for the wood log construction and 30 years for the concrete construction would seem to be a very conservative estimate. Soil conditions in that part of the city where the heating mains are laid are particularly favorable, the soil being very well drained. Manholes are located at intervals of about one city block (300 to 400 ft.), to house the slip joints, valves and bleeder traps.

53 In the heart of the business district the pipes are in tunnels, of which there are about 2 miles, lying from 25 to 40 ft. below the street level. In cross-section they are similar to a horseshoe and are built of brick, with concrete floors. (Fig. 9.) For the most

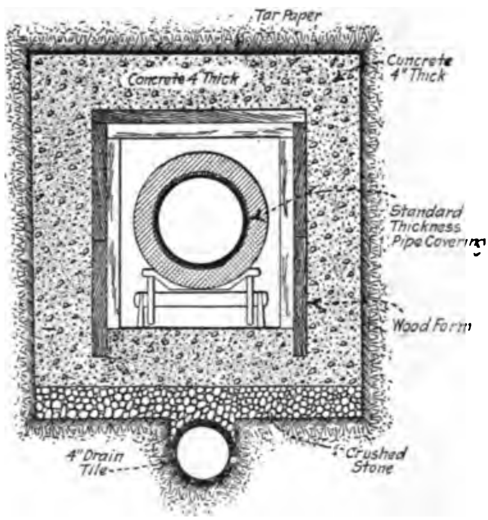


Fig. 8 Underground Steam-Line Construction

part they are about 6 ft. in height and 6 ft. wide. The tunnels are ventilated by suction fans which draw a small amount of air through them continuously and much larger amounts when it becomes necessary for men to work in them. Their temperature ranges between 90 and 130 deg. fahr. When two or more pipes are to be laid under a street in a congested district it has proved desirable to build a tunnel to avoid the inconvenience to the public attendant upon tearing up the street, either for the original construction work or for subsequent changes. The tunnel permits of ready access to the pipes at any time, and since it is far below any other structures, no obstructions are met with. Tunnel construction is exceptionally simple in Detroit where the subsoil is a blue clay nearly impervious to water and well adapted to tunneling operations. The cost of tunnel construction is high but the advantages gained are compensatory.

FIG. 9 TUNNEL CONSTRUCTION FOR STEAM LINES

DISTRIBUTION LOSSES

54 Distribution losses from various sources are considerable. Heat losses from the mains are the principal item. This loss can be

controlled within limits in designing the system by the application of the proper amount of insulation, so as to produce the most economical relation between heat loss and investment costs. Besides the condensation losses in the mains and service connections, there are losses due to the leakage of steam from consumers' piping and air values, and loss due to the escape of unmetered condensation, and the slip of meters. About 80 per cent of the steam delivered by the plants is metered as condensation in the consumers' buildings. This figure compares favorably, considering the nature of the business, with the efficiency of electrical distribution. In 1918, for example, the ratio between the electrical plant output and the consumers' meter readings for the Detroit district was 83 per cent, the large items of loss being transmission, transformation and conversion.

CONDENSATION RETURN LINES

55 The condensation from the buildings heated is returned to the plants only to a very limited extent. In the tunnels it is necessary to install a return line to receive the discharge from the traps on the steam lines, and wherever possible the condensation is drained from the adjacent buildings to this line. It is difficult and costly, however, in many cases, to arrange a gravity discharge from the building basements to the tunnel, and the cost of installing and operating pumps to handle the condensation would more than offset the value of the heat and the feedwater which would be salvaged.

56 In the districts not served from tunnels, all of the condensation is wasted to the sewers. Even with the present high cost of coal, return lines would scarcely be a profitable investment. Furthermore, they are short-lived, and any leakage from them is disastrous to adjacent steam lines. The proper method of salvaging the heat in the condensation is by means of an economizing coil in the consumer's building.

CONSUMERS' INSTALLATIONS

57 The consumer's equipment includes, besides the usual radiators and piping, a pressure-reducing valve, which reduces the service pressure to the lower pressure required for heating, and a trap, whose function is to discharge the water of condensation from the system.

58 Any existing steam-heating system can be adapted for

service by making a few minor changes. A hot-water system can be served with steam by substituting for the fuel-burning water heater a surface heater in which the water in the system is heated by the steam. A hot-air system can be adapted to the use of steam by substituting steam coils for the furnace and using the same duct system, though this is rarely done.

59 Economizing coils, utilizing the heat in the condensation, are recommended by the Company but not required. They are seldom installed in any but the largest buildings as few consumers care to make the necessary investment even though a demonstrated saving can be made. In some buildings the condensation is passed through a surface heater which preheats the water used for lavatory purposes, and this seems to be the most satisfactory form of economizer for large buildings.

60 The consumer's installation is the least reliable factor in the maintenance of uninterrupted and satisfactory service. Boiler plants can be and are designed and operated so as to be extremely reliable. A distribution system, if properly laid out, with ample capacity and duplicate feeding routes, can be operated so that the steam supply to a building is practically never interrupted. But the consumer's piping system, and the special appliances attached to it, operated by persons unfamiliar with mechanical apparatus are a frequent source of trouble. Tagging of the valves to indicate their proper operation, distribution of printed instructions and other educational measures are employed with varying success, and the service is on the whole much more reliable than that obtained from the ordinary individual plant. A "trouble service" is maintained day and night to insure immediate attention to minor repairs and adjustments of consumers' equipment.

61 The regular heating season covers 8 months of the year, though summer service for cooking and water heating is supplied to a few consumers conveniently located. Cool weather in September usually makes necessary the commencement of service before the contract date of October 1. It has, in fact, been started in August. Steam was originally sold to some extent for power purposes but this service is now practically discontinued.

METERS

62 With the exception of a number of calibrated steam jets used in cooking fixtures, the steam is sold entirely on a condensation

basis, the condensation flowing from the system through a meter and thence to the Company's return line or to the sewer.

63 The art of metering condensation was comparatively new when the Company commenced operations and although many advances have been made, it has not yet reached, and probably never will reach, the standards of reliability and accuracy of electrical metering. The troubles experienced are largely mechanical; but there are certain fundamental obstacles in the way of their entire

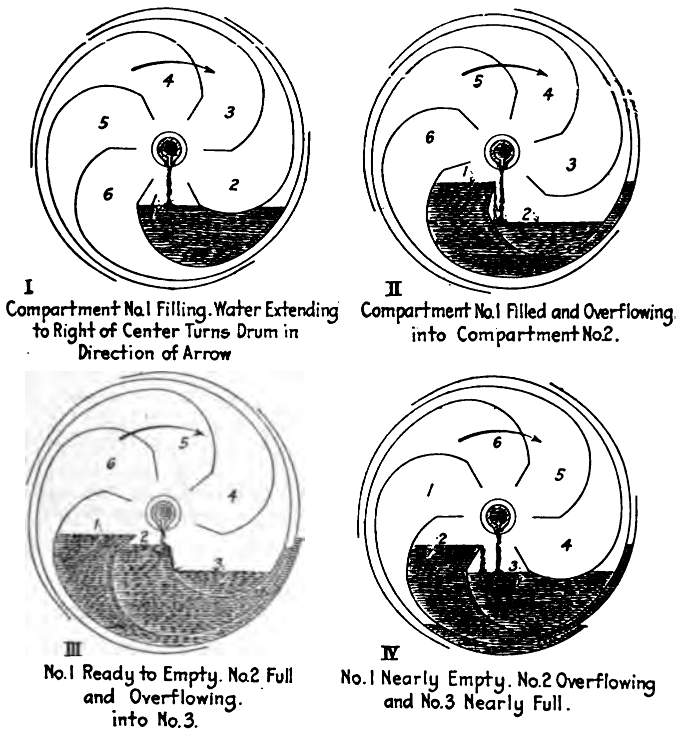


FIG. 10 CONDENSATION METER AND ITS PRINCIPLE OF OPERATION

elimination. The operating conditions, because of excessive temperature, moisture, and dirt are severe, and the allowable size and cost of the meter are limited.

64 The meter first used in Detroit was of the tilting type, consisting of a rectangular pan of two compartments hung on knife-edge or ball bearings, the compartments filling alternately and, when full, tipping the pan so as to bring the opposite compartment to the filling position. In 1907 a revolving meter was devised,

consisting of a drum of four compartments which filled and dumped successively, the drum being revolved by the weight of the water. This meter was fairly successful and a few of that design are still in use, but a greatly improved design of revolving meter, operating on the same general principle (Fig. 10), was originated in 1909 by Hans Resert and is being used at present, with slight modifications.

LOAD CHARACTERISTICS

65 The daily boiler-plant load curves vary considerably both in magnitude and in shape with different outdoor temperatures. In

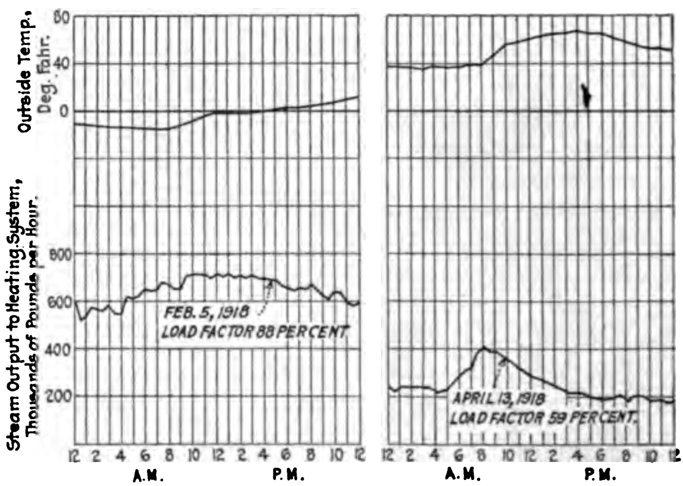


FIG. 11 TYPICAL LOAD CURVES OF HEATING SYSTEM

moderate weather steam is used in many buildings only for a few hours in the morning, resulting in a decided peak at that time with a steady decrease in load throughout the remainder of the day. On the very coldest days, however, the heat is used continuously throughout the 24 hrs., in most buildings, giving a daily load factor on such days of between 85 and 95 per cent. This is illustrated in Fig. 11. The monthly sales of steam are very closely proportional to the difference between 70 deg. fahr. and the average outside temperature. The annual load factor is about 35 per cent.

66 The load curves of the individual buildings show these same general characteristics although different types of buildings have markedly different daily load curves. In Fig. 12 are shown the

load curves of three buildings of different types. The high peak, in the case of the theater, was caused by the throwing on of the fan system. The steam in the office building was shut off at night, while in the hotel it was used continuously.

67 Detroit rates at the present time take no account of varying load factor though this is done in some other cities in an approximate way. Possibly future progress in the art of metering will lead to the general establishment of rate schedules having a demand component as well as a consumption charge.

ADVANTAGES OF CENTRAL HEATING SERVICE

68 The existence of a central heating system in a city holds certain distinct commercial and economic benefits to the consumer, the electric company, and the community in general.

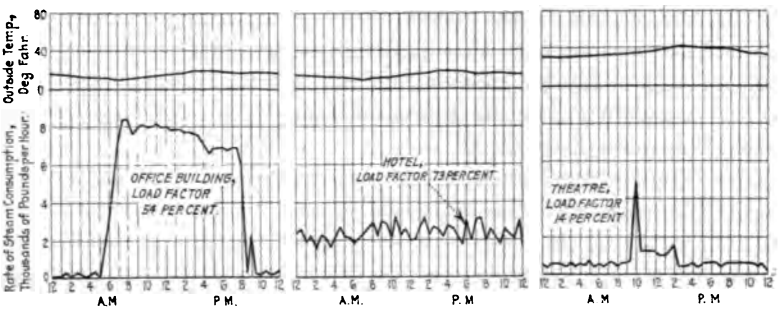


Fig. 12 Typical Load Curves of Individual Consumers

69 In a residence district the simplification of domestic arrangements, the elimination of smoke and of the dirt and annoyance resulting from the handling of coal and ashes, and the availability at all times of any desired amount of heat, when coal is high priced and scarce, are factors of such real merit that they are reflected in higher land and rental values. The scarcity of domestic labor in American homes and the steady trend toward more luxurious standards of living are yearly increasing the value of these advantages to the consumer. The high cost of underground mains compared to the amount of steam which can be sold in an area of detached residences, however, makes this the least desirable class of service.

70 The owner of the medium-sized commercial building is particularly appreciative of the service since by so arranging the piping

that the heat used by his various tenants is metered separately and paid for directly by the tenants, he is entirely relieved of this burden. In the large downtown store or office building, the elimination of coal and ash handling is a very tangible advantage and the space saved by the exclusion of boilers has a measurable rental value.

71 To the community at large the elimination of smoke and soot, the better handling of coal and ashes, and the reduction in manual labor, are distinct economic benefits.

72 The ability of the service to displace the individual plant in residences and business buildings rests upon the value of these factors and upon the savings which are possible through the more economical purchase, handling and utilization of coal in the large central stations as compared with the wastefulness of the small individual plant.

73 The obstacles to the wider use of the service are the burden of the investment in the distribution system and the losses which necessarily accompany the distribution of heat. Both of these factors are of greater moment in an area of detached residences, and consequently, the most favorable field is in the more densely loaded business districts.

74 That central heating service in general can compete at profitable rates with the individual plant is sometimes questioned. Its ability to do so must rest, in the last analysis, upon a proper appreciation, by the consumer, of the intangible advantages which have been mentioned. The improvement in standards of living and the general progress toward cleaner cities are undoubtedly serving to increase the value of these advantages.

COMBINATION WITH INDIVIDUAL PLANTS

75 In some of the larger cities operating companies have been formed which take over and operate existing boiler plants in downtown buildings and supply steam for heating the buildings in which they are located and often adjoining buildings. This plan owes its existence to the fact that the gain in economy of the central station over the individual plant is less in the case of large buildings where the boiler plants are often of considerable size and can be operated with fair efficiency, and also to the fact that the cost of installing of distribution mains in such districts would be very high. By relieving the building owner of the burden of operating the plant, the operating company performs a service of measurable value.

76 Some of the chief advantages of the central heating service are defeated by this plan, however, since the dirt, smoke, and labor nuisances still exist. By combining such existing plants with an efficient central station and operating them only in the coldest weather, advantage can be taken of their capacity to reduce the size of the main boiler plant, and of the feeders and distribution mains. Possibly this plan will be the ultimate solution in some of the larger cities. It may be described as a relatively cheap investment, with relatively high operating costs, to take care of occasional peak loads, while the more costly and efficient central plant and street pipes take care of the body of the annual load curves with obvious economic possibilities.

DISCUSSION

August H. Kruesi emphasized the importance of the author's statement of the reasons for using live steam in Detroit which showed that the additional fuel necessary, if current were generated as a by-product of heating, would be one-quarter of that which would be used in a condensing steam plant. The reason was the great improvement in the performance of large turbo-generators. In the plants with which he was associated, he said, they charged 13.5 per cent of the steam passing through the engines and turbines to power and the remainder to heating, and he thought that 10 to 13 per cent a more representative figure of ordinary industrial practice than the 27 per cent which prevailed under the conditions at Detroit. In such a case the by-product plant should receive a greater credit, that is, the author's item of $10,688 should be doubled, and the return on the investment of an exhaust-steam heating plant would be in the neighborhood of 25 per cent.

The Author, in answer to a question, stated that the use of exhaust steam for district heating had been definitely abandoned in Detroit four years ago and repeated that the underlying reason for giving it up was that current could not be generated in the heating plants as cheaply as in the main generating station, this being due to the fact that the small generating units installed in the heating plants must necessarily be reckoned as *additional* investment since they certainly would not reduce the number of generating units in the larger plants.

In his closure, the author emphasized the fact that the paper was intended to describe the conditions existing in Detroit and did not imply that the use of exhaust steam for central heating was not generally feasible. It was true, however, that these conditions might ultimately be reached in many other cities, for with the increasing size and efficiency of condensing generating plants it will become more difficult for central heating stations to produce electricity which can compete in cost with that produced by the condensing station.

No. 1695

THE PRODUCTION OF LIBERTY MOTOR PARTS AT THE FORD PLANT

By W. F. Verner, Detroit, Mich.
Member of the Society

This paper deals with the production of Liberty motor cylinders and connecting-rod crankshaft bearings as carried on at the Ford Motor Company's plant at Detroit. The contract made with the United States Government called for 5000 motors and these were to be produced at the rate of 50 per day of eight hours. To do this, important developments in the methods of manufacture were brought about by the Production Department of the Ford Motor Company.

One of these was the method of producing cylinders from tubing. Six operations were necessary and the author describes them in detail. The methods employed to produce connecting-rod crankshaft bearings likewise resulted in a great saving of time. Twenty-one operations were found necessary for this work and a complete description of each is given. The paper concludes with an explanation of the method of installing bearings in the upper and lower halves of the Liberty motor crankcase.

On November 22, 1917, the Ford Motor Company entered into a contract with the United States Government to build 5000 Liberty motors. The contract was accepted at a time when Ford cars were being manufactured at the rate of 3500 per day and to change over from their production to that of Liberty motors was by no means a minor undertaking. The manufacture of Liberty motors differs in many essentials from the manufacture of the ordinary type of motor used for automobiles, and of the 14,000 tools used on Ford car production only 987 were adaptable to the production of Liberty motors.

2 It was estimated that 350,000 sq. ft. of floor space would be required to produce 50 Liberty motors per day of eight hours. The ultimate space required was 550,000 sq. ft. As no floor space was available, this necessitated the dismantling and removing of several thousand machines and the rearrangement of over 50 per cent of the regular plant equipment. The new arrangement was so made

---

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

as to allow an entire building for the production of Liberty motors. This permitted the concentration of production and therefore maximum efficiency and production in the shortest time, and since production in the shortest time possible was the great issue, no expense was spared in this shop rearrangement.

3 The shifting of labor from standard-car production to Liberty motor production was gradual. No labor trouble was experienced as the curtailment in regular products released enough labor for the production of Liberty motors. Table 1 shows the growth in the number of men employed in the Liberty Motor Department.

TABLE 1 SHOWING GROWTH OF FORCE BUILDING LIBERTY MOTORS AT FORD MOTOR CO. WORKS

Date	No. of Men Employed	Date	No. of Men Employed
Nov. 22, 1917	0	Aug. 1, 1918	7,976
Feb. 2, 1918	675	Sept. 5, 1918	9,390
March 1, 1918	779	Oct. 1, 1918	10,653
April 1, 1918	1,550	Nov. 1, 1918	11,288
May 1, 1918	2,450	Dec. 1, 1918	826
June 1, 1918	3,412	Jan. 2, 1919	543
July 1, 1918	5,141		

4 In accordance with the terms of the contract deliveries were planned as follows:

April 1918	200
May 1918	800
June 1918	1000
July 1918	1000
August 1918	1000
September 1918	1000
Total	5000

5 This schedule was subsequently revised with instruction from the Government to attain a maximum production of 100 motors per day in anticipation of an order for 7000 additional motors. The schedule was revised to obtain a production of 2500 motors per month by December 1918.

6 Production was fast approaching the goal when the armistice was signed, as evidenced by Fig. 1, which shows the production of Liberty motors at the works of the Ford Motor Company for the period from November 1917, to December, 1918. As is invariably the case in any new undertaking, the production of these motors

was retarded by many factors which it was the effort of the engineers to eliminate. Among the predominating ones were the following:

a Orders for raw material could not be placed immediately on account of incomplete detailed specifications

b There were many changes. For a period of 14 months, 1013 changes were authorized: March 1918 showing 167, April 109 and May 115; then tapering off to December 1918 which showed only eight changes. These changes

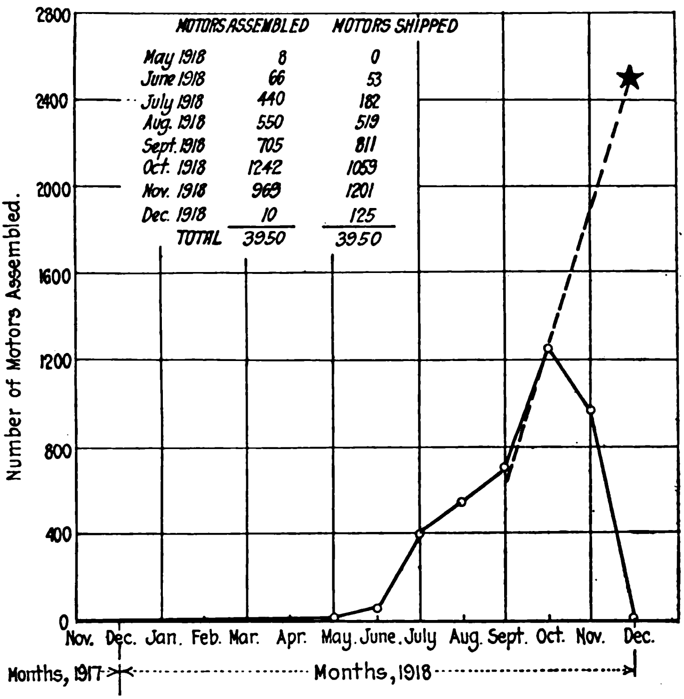


FIG. 1 PRODUCTION OF LIBERTY MOTORS AT THE FORD MOTOR COMPANY'S PLANT

represent Government changes and do not include those made by Ford Motor Company in dies, jigs and fixtures which averaged from three to five for each of the above

c Material specifications were constantly revised

d Shortage of fuel, especially with sub-contractors

e Railroad embargoes delayed shipment on shop machines and raw materials

f A-4 priority on shop machinery instead of A-1 *g* Lack of acceptable thread gages

h Lack of actual shop experience by Government inspectors.
The falling off in production in November was of course due to the signing of the armistice. On the day the armistice was signed, the highest mark was reached in the number of motors assembled in one day, namely, seventy-five.

METHOD OF PRODUCING MOTOR CYLINDERS

7 Several major and important developments were brought about by the Production Department of the Ford Motor Company. First among these was the cylinder forging made from tubing. This method resulted in an enormous saving when considering the cost of machining a cylinder from a solid forging and also the cost of making a solid forging. The Liberty motor cylinder was forged from a high-carbon steel tube $5\frac{3}{4}$ in. outside diameter, $4\frac{3}{4}$ in. inside diameter by $39\frac{3}{4}$ in. mill length, and completed in six operations as follows:

Operation A — Cut-off
Operation B — Close head
Operation C — Form the head
Operation D — Rough-drill bosses for inlet- and exhaust-valve ports
Operation E — Upset and form flange
Operation F — Heat-treat

8 *Operation A — Cut-Off.* The tube was heated, at the point at which it was to be cut to about 1200 deg. fahr., in a specially designed rectangular gas furnace having a series of circular openings along two sides, and through which the tubes were inserted. The capacity of the furnace (Fig. 2) was such that the successive tubes were heated sufficiently for cutting within the time required for the cutting operation. Once started, the operation was continuous with a production rate of 150 tubes per hour per machine.

9 Upon removing the tube from the furnace it was placed in the shearing die of a press equipped with a special punch and die. The tube was then fitted with an arbor so constructed that, as the punch of the press sheared the outer wall of the tube, the arbor transmitted the shearing power to the lower wall, thus shearing the whole without distorting the tube. The punch and die were set on the press so that the end of the tube was cut at an angle of 19 deg. with the center line of the tube, to the required length of

$20\frac{5}{8}$ in. at one side of the angle and $19\frac{1}{8}$ in. at the other side. This angular cut was essential to Operation B.

10 *Operation B — Close Head.* It was considered for a long time next to impossible to forge a Liberty motor cylinder from a tube on account of the manufacturing difficulties encountered in closing the head. When the end of the tube to be closed was cut at right angles to the center line, it was found unsatisfactory due to cold

FIG. 2 GAS FURNACE FOR HEATING TUBES — OPERATION A

shuts or unfused sections in the metal occurring in the center of the dome.

11 However, by cutting the tube at an angle of 19 deg. with the center line of the tube it was found that the forming dies could be so constructed to cup or draw inward the tube wall, the high or extended portion of the wall causing the converging or closing of the metal to one side of the center line of the tube until, in the final forming of the head, the metal was joined at right angles to the 19 deg. cut (see Fig. 3). After this operation the appearance of the closed end resembled the common type of explosive shell with the nose portion at an angle of 19 deg. The central portion of the

dome is thus formed without a weld and retains to the fullest extent its fibrous strength.

12 The angular head of the tube was then heated in a furnace, similar to the one previously described, to about 1900 deg. fahr. preparatory to forming the head so that the point could be used as a part of the boss which later was drilled for an intake or exhaust post.

13 The die used in this operation was of the double-action type

Fig. 3 Preliminary Forming of the Cylinder Head — Operation B

and comprised two semi-circular steel jaws, tapered on the upper outside diameter and pivoted in the rear to swing horizontally. A cast-iron locking plate was attached to the blanking foot of the press and tapered to correspond to the taper of the jaws, so that when the blanking foot was in down position the tapered surfaces of the jaws served to lock the tube in position. The interior upper parts of the jaws were bored and fitted with split bushings or bronze bearings to fit the punch. Resting flush, normally with the upper surface of the jaw, was a semicircular steel supporting band equipped with three guide pins supported on springs.

14 When the jaws were swung in position around the tube, they formed a steel ring which gripped the tube around the top or heated portion and as the punch descended, the ring slipped down the tube, and the supporting springs depressing under the pressure of the punch thus prevented the tube from bulging when the punch closed in the head. The punch was designed so that the dome was

Fig. 4 Press Used in Final Forming of Cylinder Head — Operation C

drawn to a point 19 deg. to one side of the center line of the tube, as previously described.

15 *Operation C — Form the Head.* This operation was performed on a press (Fig. 4) provided with a specially designed punch and die. To serve as a die, a bolster or base plate was mounted on the bed of the press. The base plate was bored in the center to receive the shank end of an upright cylindrical locating arbor and counterbored to receive a thrust plate. Two sections made up the arbor, the lower of which was made of soft steel bored to receive the hardened-steel tip or top section. The top section tapered slightly

inward at the extreme end and the top surface was curved to properly form the dome or head of the cylinder.

16 Horizontal sliding jaws around the locating arbor were held open by springs and operated by cams attached to the ram of the press. The upper interior portion of the jaws was shaped to form the expanded area for the combustion chamber. When the jaws were closed by the action of the cams, they fitted snugly around the punch.

17 As the ram descended, the cams attached thereto forced the jaws together so that as the punch pressed down the head of the tube on the arbor it formed the valve-port and spark-plug bosses and the jaws formed the expanded area for the combustion chamber. On the back stroke, the jaws were forced apart by the springs as soon as the pressure of the cams relaxed. In case of adhesion to the cylinder, a wedge attached to the ram and operating between the two ends of the jaws breaks the adhesion on the back stroke. A knock-out sleeve located about the base of the arbor and operated by an arm beneath the bed of the press on the back stroke was carried upward and loosened the cylinder on the arbor. Production on one machine totaled 150 per hour.

18 *Operation D — Rough-Drill Bosses for Inlet- and Exhaust-Valve Ports.* The cylinder was held in a trunnion fixture attached to a drill press so that the center line of the valve port swung in line with the spindle. The cylinder was located by using the valve-port bosses.

19 *Operation E — Upset and Form Flange.* These operations were performed on a 5-in. forging machine in one heat-treat on two separate dies. The die used for the first portion of the operation (upset) comprised two horizontal sliding steel jaws operated by cams and bored at both ends to fit tighly about the body of the cylinder. At the middle the jaws were undercut or recessed so that in the upsetting the metal would so flow as to form a heavy ring of metal at the section at which the flange was to be located. The punch was in the form of a mandrel with a shoulder which fitted the closed jaws and which on striking the bottom of the skirt or open end of the cylinder forced the heated metal to the proper upset dimensions about the mandrel and into the recess of the jaws.

20 The die employed in the second portion of the operation (form flange) was of the same type above described, with the exception that the jaws were counterbored at the entrance end to the forged flange dimensions, instead of being recessed in the middle.

The punch consisted of a mandrel with a shoulder surrounded by a sleeve which extended beyond the shoulder. The sleeve fitted the closed jaw of the die and the inside extended portion upset the metal by pressing it against the die, thus forming the flange. The sleeve was provided with two vent holes permitting gases that might be formed during the forging, to escape.

21 In operation the skirt of the cylinder was first heated to about 1900 deg. fahr. in a furnace, then dipped in water to a depth of about 1½ in. This cooling was done to form a ring of hard metal at the bottom of the skirt for the punch to act upon. The cylinder was then placed in the forging machine and the flange made. Production totaled 85 per hour on one machine.

22 *Operation F — Heat-Treat.* The completed forged cylinder was placed in a large rectangular heat-treating furnace and heated to a temperature of 1525 deg. fahr. and then quenched in a brine solution. After quenching it was heated in an annealing furnace to a temperature of 1125 deg. fahr. and cooled in air. Brinell test was 217–255. This heat treatment left the cylinder ready for the machine operations.

METHOD OF MANUFACTURING CONNECTING-ROD CRANKSHAFT BEARINGS

23 Next in importance to the method just described of producing Liberty motor cylinders was the development of a special process of making bronze babbitt-lined bearings for the crankshaft end of the connecting rods which would stand up under the Government's 50-hour test. The method comprised 21 operations, as follows:

1 Rough-drill closed end
2 Rough- and finish-bore bronze and face one end
3 Turn outside diameter to fit babbitting fixture and face one end
4 Babbitt
5 Cut-off gate, rough- and finish-bore babbitt
6 Finish-turn outside diameter to 3.095 in. and face ends to length
7 Press in broaching ring
8 Broach hole to 2.4275 in. in diameter
9 Press out of broaching ring
10 Grind outside diameter to 3.075 in.

11 Cut in halves, using $\frac{3}{64}$-in. saw (2 on)
12 Close in
13 Swage
14 Finish-mill the parting line
15 Face ends to length
16 Fillet both ends
17 Cut two grooves for forked-end rod
18 Drill and ream dowel holes in lower half-bearing only
19 Cut two $\frac{1}{8}$-in. semicircular oil grooves on the parting line
20 Cut twelve oil pockets in both halves
21 Burr.

24 *Operation* 1 — *Rough-Drill Closed End.* A 21-in. drilling machine was used in this operation, which was necessary only on castings where a thin web of metal entirely closed one end.

25 *Operation* 2 — *Rough- and Finish-Bore Bronze and Face One End.* This operation was performed in a 12-spindle 14-in. multiple machine. The bushings were gripped in a chuck and bored at the rate of about 80 per hour.

26 *Operation* 3 — *Turn Outside Diameter to Fit Babbitting Fixture and Face One End.* A 12-spindle 14-in. multiple machine was also used for this operation. The bushings after boring were clamped on special arbors and the outside diameter turned to fit babbitting fixture.

27 *Operation* 4 — *Babbitt.* The equipment required for this operation consisted of acid vats, tinning furnaces, die-casting machines with water-circulating systems, a furnace large enough to supply a unit of four die-casting machines with molten babbitt and a compressed-air outfit to use with the gas furnaces.

28 The bushing was first dipped into a flux made proportionately of 11 lb. sal ammoniac, 9 lb. zinc chloride, 6 qt. muriatic acid and 13 qt. water. The hydrometer reading was 23 to 25 deg. B.

29 Before placing the bushing in the specially designed die-casting machine to be babbitted it was immersed in the molten tin. The die-casting machine, Fig. 5, consisted of a rectangular plate mounted on suitable legs or base, having a cored hole for receiving a crucible containing the metal; a crucible with suitable rim having two bosses on which were fitted bearings for a pump-lever shaft; a pump fastened to two bosses of the rectangular plate for forcing the metal through a nozzle into the die proper; and a fixture mounted on the plate straddling the pot for casting the bearing.

30 The fixture for casting the bearing was made up of a circulating-water-cooled mandrel or arbor sliding vertically in a housing. This housing was secured to two side-support brackets which were bolted to the plate. The lower die holder carrying the die spans the crucible and slides vertically on guide pins. Springs received the weight of the holder in suspensio 1. Just inside of the guide-pin bearings and screwed into the holder were two suitable rods

FIG. 5 BABBITTING MACHINE — OPERATION 4

which extended through the housing. The housing was drilled and counterbored to receive rods and springs. The lower die holder in its normal or loading position was suspended on the springs so that the opening or gate of the die was slightly above the nozzle of the pump.

31 For babbitting, a tinned bushing was inserted in the lower die and the water-cooled arbor was moved downward until stopped

by the arbor stripper ring, clamping on the upper end of the bushing to be babbitted and the gate end of the lower die on the pump nozzle. The movement of the arbor was controlled by an upward movement of the hand lever. While holding the arbor firmly in position with one hand, the operator with the other hand pulls upward on another hand lever attached to the pump lever, thereby forcing the molten babbitt from the pump cylinder through the nozzles and into the bushing.

32 The metal was allowed to set for about 30 sec., when the pump-control lever was pushed downward. The piston of the pump upon its return uncovers ports permitting molten metal to flow into the cylinder preparatory to another casting. Simultaneously the arbor-control lever was thrown downward, assisted by the weighted end, causing the arbor stripper ring to strike violently against the lower face of the arbor housing and thereby stripping the babbitted bushing from the arbor. This upper movement of the arbor allowed the lower die holder to regain its normal position and severed the connection between the lower die and pump nozzle, which assured the bushing sticking to the water-cooled arbor and not becoming gate-anchored to the lower die. The thickness of the babbitt wall at the top for best results in babbitting was $\frac{3}{32}$ in. and for the bottom $\frac{3}{16}$ in.

33 *Operation* 5 — *Cut-Off Gate, Rough- and Finish-Bore Babbitt.* Geared-head screw machines were used for this operation. The bushing was held by a three-jaw clutch; an ordinary cut-off tool held in the tool block of the cross-slide was used for cutting off the gate, and a boring bar with two cutters (one for roughing and one for finishing) was mounted in the turret for boring hole to proper diameter, with an allowance for broaching.

34 *Operation* 6 — *Finish; Turn Outside Diameter to* 3.085-*in. and Face Ends to Length.* In this operation 14-in. x 4-ft. lathes with back-arm attachments were used. The bushings were held on an expanded arbor, which was held in the spindle of the machine. A cutter mounted in the tool block of the lathe cross-slide turned the outside diameter to fit the broaching ring. The back arm carried a tool block with two cutters spaced to face the bushing to its proper length.

35 *Operation* 7 — *Press in Broaching Ring.* For this operation (see Fig. 6) a bolster plate with two holes bored large enough to allow the bushing to drop through and counterbored to suit outside diameter of the broaching ring was strapped to the bed of a press.

In the ram of the machine were carried two cylindrical punches slightly smaller in diameter than the outside diameter of the bushing but differing in length.

36 In operation a ring into which had been pressed a broached bushing was placed in the counterbored hole under the long punch, and a bushing that was not broached was slightly entered into an

FIG. 6 PRESSING IN BROACHING RING — OPERATION 7

empty ring and placed under the short punch. When the press was tripped, the long punch forced the broached bushing out of the ring and the short punch forced the bushing, which had not been broached into the ring.

37 *Operation* 8 — *Broach Hole to* 2.4275-*in. Diameter.* A special broaching machine, Fig. 7, was designed for this operation. The

legs were removed from a No. 1 Knowles keyseater and the body complete with gears, rack, slide, etc., was bolted in a vertical position to a special base casting. An upper guide of cast iron, with a hardened and ground steel bushing for the broach-holder quill was

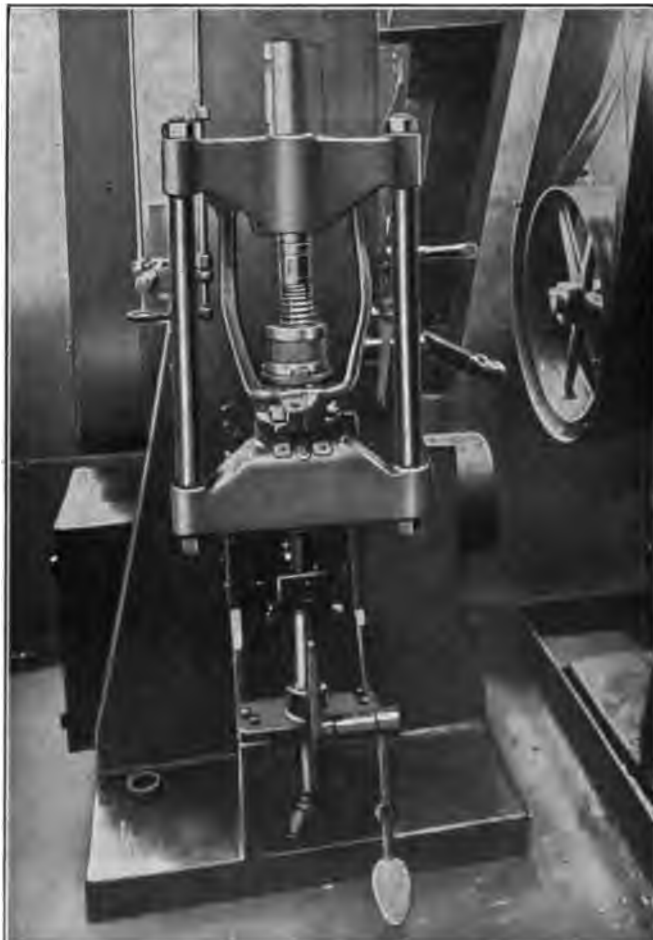


FIG. 7 SPECIAL BROACHING MACHINE — OPERATION 8

dovetailed to the ways of the machine. Below this upper guide was the extended work-holding bracket (a part of the bed of the Knowles keyseater) which carried the work holder. This work holder was a hardened steel bushing ground on its outside diameter to fit the hole in the bracket and counterbored to fit the broaching ring. On the special base casting below the work-holding support was bolted the lower guide, a gray iron casting with a hardened bushing ground to fit the broach holder. The broach holder was a long tool-steel bar hardened and ground on one end to fit the guide bushings and on the other end to fit the holes in broach and broach-holder quill.

38 The foot-pedal bracket was bolted to the base of the machine and was bored to fit the broach holder, which it served to guide and keep in alignment. Between the lower guide casting and the foot-pedal bracket was disposed a collar firmly fastened to the broach holder, and attached to this collar was a yoked lever. This lever was so fulcrumed that the weight of the broach was just slightly more than counterbalanced by a cast-iron weight which insured the broach end of the holder being piloted in the quill when the broaching was being done. A hand lever with lift rods was attached to the yoked lever to control the loading.

39 The broach had eight cutting edges varying in size from 2.422 in. diam. to 2.4273 in. diam. In addition to the cutting edges, the upper end of the broach had three burnishing surfaces, the diameter of which were 2.4274 in. The broach was 5 in. long and had a hole ground to 1.625 in. diam. to fit the end of the broach holder.

40 In operation, the keyseater functions normally, with the exception that the down (or what would be the return) stroke of the ram is utilized for pushing the broach through the work. The hand lever controlling the broach holder is pulled down until the latch on the foot lever engages the top of the collar on the broach holder. This operation holds the pilot end of the holder and the broach holder quill apart, allowing the broaching ring containing the work to be mounted in place. The broach holder was then allowed to ascend until the end of the holder protruded enough to allow slipping on the broach, and then further until its end was piloted in the hole of the quill. The machine was then tripped and the broaching was done on the downward stroke of the slide carrying the quill, the broach being forced through the work by the pressure exerted on descending quill. This downward stroke was carried far enough to allow the foot-lever latch to engage the collar on the broach holder automatically. The machine was reversed, disengaging the quill and the broach holder. The broaching ring containing the work was then lifted out and the broach was removed from the holder; leaving the machine ready for the next operation.

41 *Operation* 9 — *Press Out of Broaching Ring.* This operation has been previously described under Operation 7, and is shown in Fig. 6.

42 *Operation* 10 — *Grind Outside Diameter to* 3.075 *in.* The bushing was ground between centers on a 6 x 18-in. plain grinder. A split hardened-steel ring, ground on its outside to a diameter of 2.4275 in., was inserted in the bushing and a hardened arbor ground with a slight taper was in turn inserted into the split ring, the hole of the latter being ground tapering to conform to the arbor. A light

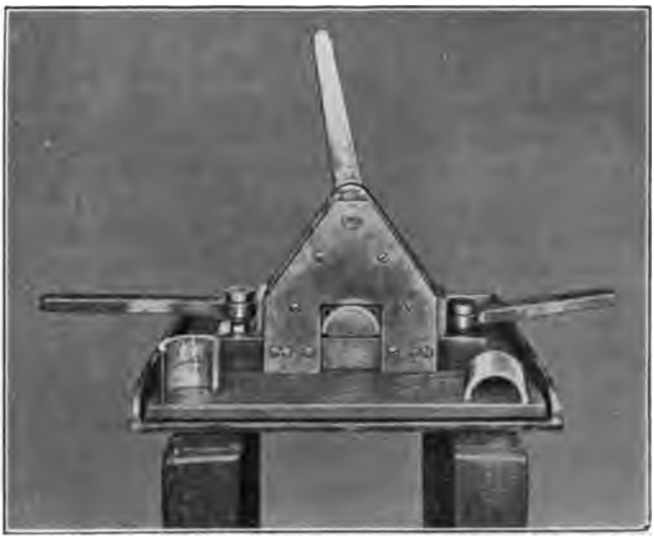


FIG. 8 MACHINE FOR BENDING BEARINGS — OPERATION 12

pressure applied to the end of the arbor expanded the ring sufficiently to firmly hold the bushing for grinding.

43 *Operation* 11 — *Cut in Halves using* $\frac{3}{64}$*-in. Saw* (2 *on*). This operation was performed on plain milling machine with an indexing type of fixture. The work-holding arbor of the fixture was made long enough to accommodate two bushings. The bushings were clamped on the arbor with a "C" washer and stud bolt. A common $\frac{3}{64}$-in.-wide milling cutter was used.

44 *Operation* 12 — *Close In.* After being cut in half, the bearings were bent to decrease their diameter so as to allow their fitting easily into the swaging fixture used in Operation 13. The bending fixture, Fig. 8, was made with a semicircular arbor bolted to a base

plate. A housing with three cam lever slides was fastened to the base plates. Openings in both sides of the housing allowed for the insertion of the bearing over the arbor. The top cam lever clamped the work in position and the side cam levers closed in the bearing the necessary amount.

45 *Operation* 13 — *Swage.* This operation, which was performed on a press, was the keystone operation in the successful production of accurate bearings, as the set given them insured their holding the shape of the master forms.

46 The fixture, Fig. 9, was comprised of a hardened and ground

Fig. 9 Upsetting of Bearings to Finished Dimensions — Operation 13

steel base plate fastened to a hardened-steel form. Two eyebolts were held by pins in each side of the form. These parts assembled formed the female section and were fastened to the bolster plate. The male section was made up of a half-round arbor of hardened steel on the clamping plate, the joining surfaces of which were also ground. Two slots in each end of the clamping plate allow the eyebolts to hold the two sections together. A tongue-and-groove construction on the joint surfaces of the sections kept them in alignment. A filler piece made of hardened steel and ground to the finished bearing dimensions was used to take the flow from the ram of the press.

A hardened-steel cylindrical punch with a flat-ground bottom surface was fitted into the ram of the machine.

47 In operation, a half-bearing with a filler piece on top was clamped between the male and female section of the fixture; the press was then tripped and the cylindrical punch on descending struck the filler piece, which projected slightly above the upper surface of the fixture. This upset the metal to the finished bearing dimensions with an allowance on the parting-line surfaces of about 0.003 in. for finish-milling.

48 *Operation* 14 — *Finish-Mill the Parting Line.* A plain milling machine on this operation was very satisfactory considering the close limits of plus or minus 0.00025 in. A fixture with a half-round seat or nest bored to fit the outside diameter of the half-bearing was bolted to the plate of the machine. The parting-line surfaces of the half-bearing were leveled by the hinge gage attached to the fixture. The work was clamped with a half-round hardened and ground steel clamp, the curved surface of which was made to fit the inside diameter of the half-bearing. Two plain milling cutters were mounted on the arbor and were so spaced that they straddled the clamping bolt and nut.

49 *Operation* 15 — *Face Ends to Length.* A 14-in. x 4-ft. lathe without a tailstock was used for this operation. A hinged clamping ring was used to clamp the work tight on the steel arbor, which was mounted in the spindle of the lathe. Two halves, or one complete bearing, were machined at one setting. A special tool block with two cutters straddling the clamping ring faced the bearing to the proper length.

50 *Operation* 16 — *Fillet Both Ends.* The fillet cuts in both ends of the bearing, to clear the radius of the crankshaft pin, were made on 21-in. drill presses. The fixture comprised a circular-shaped base bored in the center to receive a flanged hardened-steel pilot bushing. The outside diameter of this pilot bushing above the flange was made to fit the inside diameter of the bearings and the hole of the pilot bushing was a fit to the pilot of the filleting cutting holder. The half-bearings were clamped in pairs about the piloting bushing by means of two hinged clamps rotating on a pin located in the rear of the fixture. A single-formed filleting center was fastened in a slotted holder. The holder was held in the spindle of the drill press with a tapered shank.

51 *Operation* 17 — *Cut Two Grooves for Forked-End Rod.* A 14-in. x 5-ft. lathe, Fig. 10, with a special cross-slide arranged with

front and back tool blocks, was used for this operation. A stub arbor mounted in the spindle of the machine was made with a flange to serve as a stop for locating the work while clamping with a hinged ring, similar to the one used for Operation 15. The half-bearings were clamped on the arbor at one setting. The arbor was provided with a center so that the lathe tailstock could be utilized to stiffen the support of the work under the pressure of the cut. The grooving cutters were of the circular forming type, with six cutting edges. The adjustment of the cutting edges was controlled by the movement

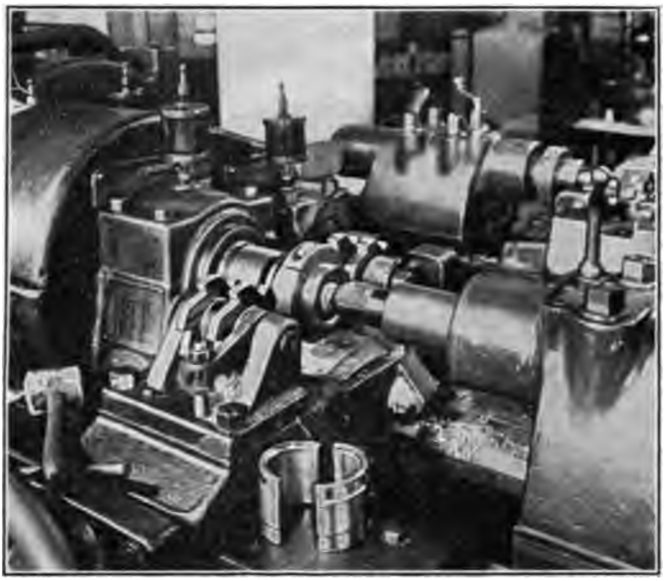


Fig. 10 Method of Cutting Grooves for Forked-End Rod — Operation 17

of a toothed lever. The teeth of this lever meshed with the teeth in the boss about the center of the grooving cutter. The cutter in the rear tool block was used for roughing, and the front cutter for finishing. Suitable stops were arranged on the cross-slide to control the depth of the cut.

52 *Operation* 18 — *Drill and Ream Dowel Holes in Lower Half-Bearings Only.* A 2-spindle 14-in. drill was used on this operation. The drill jig comprised a base with side supports, to which was fastened the drilling plate. On the under side of this plate was fastened a locating block formed to fit the contour of the work.

A slide disposed between the side supports and actuated by a cam lever served to clamp the work in position for drilling and reaming. Slip bushings were used in the drilling plate to insure accuracy, one for the drill and one for the reamer.

53 *Operation* 19 — *Cut Two $\frac{1}{8}$-in. Semicircular Oil Grooves on the Parting Line.* This operation was done on a hand miller. The cast-iron fixture supported the work in a semicircular nest and was clamped in place with a strap and thumbscrew.

54 *Operation* 20 — *Cut Twelve Oil Pockets in Both Halves.* This

FIG. 11 METHOD OF CUTTING OIL GROOVES — OPERATION 20

operation was also done on a hand miller. The fixture, Fig. 11, was made to so clamp the work that the parting surfaces were in a vertical position. The oil pockets were cut on a radius, the cut being $\frac{1}{32}$ in. deep at the parting line and running out to the inner surface of the bearing $\frac{1}{4}$ in. below the parting line. An arbor with six $\frac{1}{32}$-in. wide plain cutters properly spaced was mounted in the spindle of the machine and supported by the over arm.

55 *Operation* 21 — *Burr.* A No. 1 keyseater was used for removing the burrs on the inside or babbitted surface of the bearings.

The fixture was made in halves, the lower half having a cylindrical section which served to hold the fixture in the machine. Ways for the slide carrying the broach were machined in the lower half, and also recesses cut for the hardened and ground blocks on which the parting surfaces of the work rested. The upper half was made in the form of a clamp, this part being bored to fit the half-section of circular hardened and ground steel liner, the inner surface of which fits about the outside of the work. Bosses bored for guide pins extended from the fixture and served to keep the halves in

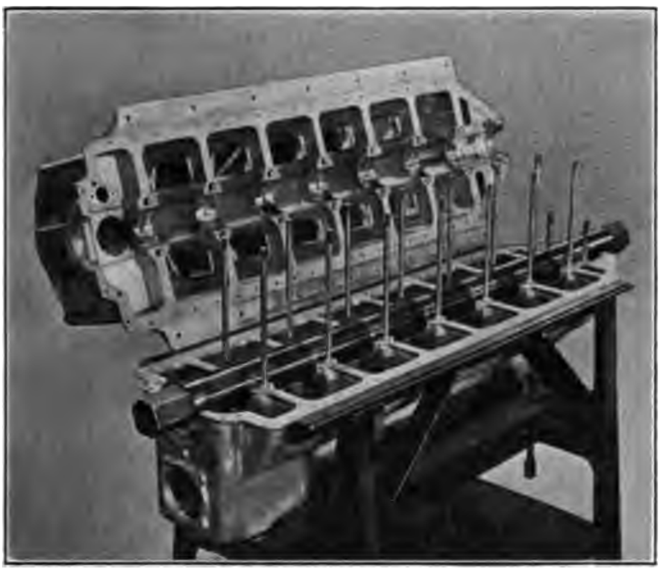


Fig. 12 Upper and Lower Halves of Liberty Motor Crankcase

alignment. The weight of the upper half was disposed on springs coiled about the guide pins. The broach was of a semicircular section and bolted to the slide. This slide was connected to the ram of the machine. A copious flow of oil was kept on the work, keeping the locating block and broach free from chips and dirt. The cutting was done on the pull or regular stroke of the machine.

METHOD OF FITTING BEARINGS TO CRANKCASE

56 Fig. 12 shows the upper and lower halves of the crankcase as it comes from the assembly department, and after being bolted

together the bearings line is reamed to 3.000 in. or slightly less. Under a tolerance of 0.0015 in. below 3.000 in. 50 crankcases can be reamed before it becomes necessary to regrind the reamer bars.

57 The most satisfactory reamer is of a solid-bar pattern having inserted cutter blades peened in place with two taper collars on reamer bar. Care must be taken in securing the main bearing bolts to the lower half of the crankcase before reaming. The nuts are drawn up until the shoulder (dowel portion) on the main bearing bolt comes metal to metal with the crankcase. The nut is then

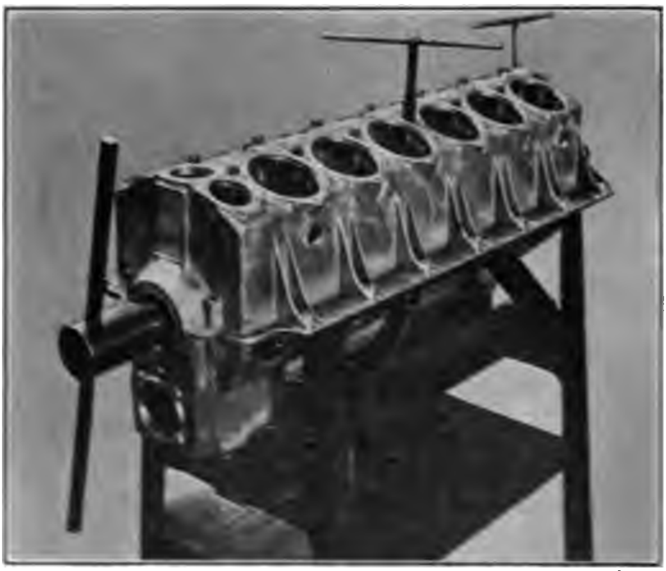


FIG. 13 LIBERTY MOTOR CRANKCASE ASSEMBLED

given an additional quarter turn. In assembling the upper half care should be taken in screwing down the nuts evenly. After the upper and lower half of the crankcase have been drawn down until they are metal to metal, the nuts are given a final turn.

58 Fig. 13 shows the crankcase assembled with a lining bar in place. The lining bar is 2.998 in. in diameter with a handle on [illegible] upper and [illegible] crankcase and [illegible] impressions [illegible] finished

show the amount of scraping to be done. This first scraping operation removes all slight distortions in the bearings and brings the holes in alignment.

59 Next the lining bar is again blued or blacked, placed as before the halves of the crankcase, bolted as previously described, and then the bar is rotated so as to again leave an impression. If the first scraping is properly done about 25 lb. pressure at the end of the bar handle will rotate the bar. The case is now ready for the final scraping, leaving the bearing holes absolutely to size and in perfect alignment. By this method, of lining and sizing the bearing holes in the crankcase, strict interchangeability of all bearings is obtained and a perfect backing is made for the babbitt-lined bronze-backed bushings.

60 The crankcase is now ready for the bearings. The backs of the bearings, both upper and lower, are blued or blacked, also the edges of the parting line of the upper halves, and a 2.6275-in. lining bar is then placed in position. The crankcase is again bolted together, using care as before in screwing down the nuts. The crankcase actually clamps the bar, and a wrench with heavy handle is used for rotating the bar. If conditions are ideal, the wrench when in a horizontal position will, when given a light tap, fall 6 in. to 8 in. to a position as shown.

61 The crankcase is again pulled apart. The lower half of the crankcase takes an impression around the main bolting from the upper half which has been blued or blacked. This indicates that the two halves of the crankcase came together and that the bearings did not hold them apart. Impressions are next looked for made by the parting line of the upper half of the bushing on the lower half of the bushing. An impression shows that the two halves of the bushings came together and that the bar did not hold them apart.

62 The bearings are numbered so that they can be identified and put back in the same place from which they were removed. The backs of the bearings having been previously blued or blacked leave impressions on both the upper and lower halves of the crankcase, and if the impressions show high spots, the cases are again scraped. This is the final scraping, after which the crankcase is ready for the crankshaft assembly.

63 The above described method may appear to be long and tedious. Two men, however, can scrape four complete crankcases in 8 hr. The bearings were furnished complete to a tolerance of

plus or minus 0.00025 in. in both inside and outside diameter. All halves of the bearings are interchangeable. No allowance was made for reaming the bearings in place. It was found that the babbitted surfaces, when left in their original broach-finished state gave better results than when reamed and scraped in place.

DISCUSSION

H. M. Crane [1] (written). Mr. Verner's description of the method of producing Liberty Engine cylinder forgings from steel tubing is extremely interesting, and I was closely enough connected with the Liberty Engine program to be able to appreciate the great aid that this system of production was to the rapid and economical manufacture of the Liberty Engine.

The method of manufacturing the bearing boxes also, as described by Mr. Verner, is extremely ingenious and advantageous, in view of the fact that while providing bearings held to extremely close limits of accuracy, it is still possible to do most of the work on a complete round bushing, which after splitting is brought to the correct shape for use without shims by what are practically punch-press operations.

To me, however, the most interesting thing in this paper is the description of seating the main bearing bushes in the crankcase. The method, of course, is not a new one, for it has long been known to be essential that the bearing bush must be properly seated to get a satisfactory job. Unfortunately, in the last few years, due to the pressure for quantity production and low cost the very essential operations described have been largely omitted in automobile engine practice. I am thoroughly convinced that this has not been economical from the automobile owners point of view, and that the slight saving in first cost has always resulted in considerable increased maintenance expense or in the use of unnecessarily large bearings to obtain a given result.

[1] Vice-President, Wright-Martin Aircraft Corporation, New Brunswick, N. J.

No. 1696

FIRE ENGINES AND THE ESSENTIALS OF FIRE FIGHTING

By Charles H. Fox, Cincinnati, Ohio
Member of the Society

The importance of the fire engine and the methods employed in fire fighting are frequently underestimated by both layman and engineer, and this is due in no small part to the fact that the principles involved are not sufficiently understood. In this paper the author presents the essentials of effective fire fighting and shows the important relation thereto of the fire engine.

A brief historical sketch of the development of the fire engine is first given, and performances and methods of rating fire engines, as outlined by the National Board of Fire Underwriters, are next presented. The paper concludes with a discussion of the losses in fire hose and the determination of nozzle areas.

STEAM power was not successfully applied to fire engines until the beginning of the year 1853. Up to that time the so-called "hand engines" were used exclusively and it should also be understood that at that time the present-day system of water works, was still in its infancy and, therefore, the chief dependence for a supply of water for fire-extinguishing purposes was upon methods of storage in vogue before water mains came into general use.

2 The conventional hand fire engine of that day comprised a rectangular wooden box suitably mounted on four low wheels. Pumps, of the piston type, were housed within and firmly fixed to the floor of the box; working levers were provided and motion was imparted to the pistons by a host of firemen lined up on opposite sides of the apparatus. At this early period fire hose was not plentiful, the best was crudely made up of leather, and the pumps were, therefore, placed close to the scene of the fire. Water, largely conveyed by a hand-to-hand passing of fire pails, was poured into the engine trough, where it was picked up by the pumps and forced through the leading hose and onward to be thrown on the fire.

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

Somewhat later it became customary to equip these hand engines with a non-collapsible suction hose, so that water could be drawn directly from cisterns or wells, but the wooden tub or reservoir always remained a characteristic feature of these old-time machines.

EARLY STEAM FIRE ENGINES

3 Early in January 1853, Mr. A. B. Latta successfully tested his new steam-driven fire engine. Mr. Latta was a citizen of Cincinnati and although his pioneer effort resulted in the production of an extremely heavy machine, the engine was purchased by the city and known as the *Joe Ross*. This first steamer marked the beginning of a new epoch in fire fighting.

4 The real basic element of Latta's invention was embraced chiefly in a quick-steaming water-tube boiler, dependent entirely upon a forced circulation of water through the steam-generating coils. The fire pumps were of an ordinary piston type. They were laid horizontally, and occupied positions forward on opposite sides of the machine. Two steam cylinders, in alignment with the pumps, were placed so that the pump rods extended to the steam pistons; while the steam piston rods, passing through the rear cylinder heads, were each coupled to a crosshead and by the use of connecting rods motion was communicated to the two rear wheels of the engine. This arrangement obviously provided means for propelling the apparatus under its own power. As a matter of fact, however, the *Joe Ross* and later engines of similar model were always drawn by horses, as the rear driving wheels could be disconnected from the engine and the pumping mechanism by means of clutches.

5 In the light of what is now known concerning the actual power requirements of a fire engine, one must believe either that the volunteers who manned the primitive hand pumps were muscular giants, or that the machine, as compared with power-driven engines, must have been woefully ineffective. The latter view is of course the correct one, and since the volunteer work was not satisfactory there was a great demand for more advanced methods.

6 Mr. Miles Greenwood, another of Cincinnati's public-spirited citizens and prominent as a manufacturer, was at that ime greatly interested in fire-department affairs. He was a staunch advocate of anything which would improve conditions, and to further his unselfish aims he took personal charge and became Chief of the Department with the result that, within three months following

the installation of the *Joe Ross* engine, the fire-fighting forces were completely reorganized under his leadership and the first paid fire-department was inaugurated in the city of Cincinnati. The year 1853 therefore also marks the beginning of the present paid system and which has since been extended and generally applied to all large fire departments.

7 Latta's second steam fire engine was built and installed in the year 1853. The purchase of this machine was made possible by popular subscription and the engine was named and long known in Cincinnati as the *Citizens' Gift*. From the beginning, Latta's new industry did not result in a monopoly, for rival steam fire-engine builders were soon in the field. Before tracing the further development which followed, it seems proper to refer briefly to the salient factors associated with fire streams and the fire engine.

ESSENTIALS OF EFFECTIVE FIRE FIGHTING

8 It may first be stated that many of the methods employed by fire departments are not properly appreciated, either by the layman or the engineer. The importance of the fire engine is frequently underestimated and the possibilities of averting disaster are also undervalued, this being due in no small part to the fact that the principles involved are not sufficiently understood. There is, therefore, need for greater knowledge of fire prevention and fire fighting.

9 When fire must be fought, fire streams can be effective only when the water is expelled from the nozzle at an appropriate speed. In other words, unless enough of the initial pressure available for starting the flow through the hose survives at a point immediately back of the nozzle orifice, the resulting jet will not measure up to its mission. The characteristics of a fire stream — good, bad or indifferent — are directly dependent upon the velocity of the jet and obviously the velocity is proportionate to the surviving pressure just mentioned. For the best results the flow may be too slow, while on the other hand disappointment will follow when the velocity of discharge goes beyond what might be termed the maximum economical limits of nozzle pressure.

10 It can be shown that little is to be gained by forcing nozzle discharges at much beyond 100 lb. pressure and the proper remedy when only such high pressures are available is to substitute a nozzle of larger bore. On the other hand if the jet seems to lag, it is an

indication that the surviving pressure is not high enough to afford the required velocity of discharge, and, under these circumstances, if the initial hydrant or pump pressure is already at its highest possible point, or if it is also impossible to augment the volume of water passing, then the proper remedy is to substitute a nozzle having a smaller bore.

11 A well-trained fireman does not necessarily have to be familiar with the hydraulic formulæ by which these varying effects can be accurately analyzed, but it would be most desirable if he could be educated to the proper use of fire nozzles under the widely varying situations encountered in the fighting of fires.

12 It may be of interest in this connection to note that the range of nozzle discharge pressures is not large between a stream of inferior reach and the swiftest jet within the economical power limit. While it is true that larger nozzles will "carry" farther at the same velocity than nozzles of smaller bore, yet for practical purposes it will be found that the most efficient fire streams are developed under the following conditions:

13 A jet issuing at a velocity under 60 ft. per sec. (about 25 lb. pressure) would not be considered a good stream. Above 60 ft. per sec. the jet stiffens and as the velocity of flow increases the stream becomes the more effective. However, when the flow attains a speed of approximately 120 ft. per sec., with a corresponding discharge pressure of nearly 100 lb. per sq. in., a point is reached where additional forcing fails to contribute much in the way of added distance. A rough but nevertheless rational assumption and one which works out well in practice is that stream velocities from 60 to 90 ft. per sec. can be easily managed, but above 120 ft. per sec. the streams are difficult to control and the pressures are needlessly wasteful.

FIRE-ENGINE PERFORMANCE

14 The function of a fire engine is either to draw water from any basin or other conveniently located source or, when fire hydrants are available to make up the pressure which is seldom high enough in ordinary water mains to serve for effective fire service. In fighting fire, it is not uncommon to elevate the nozzle far above the source of the water supply. This procedure of course involves loss of forcing pressure, which is in proportion to the static head of the column. The greatest power-absorbing medium between the source of supply and the point of discharge is the fire hose. It may also be

said that here is involved the point which is least understood in the subject of hydraulics as applied to fire-fighting practice.

15 Fire-engine pump pressures must necessarily be kept within limits compatible with the strength of the fire hose, and when working pressures are maintained at upward of 250 lb. there is always considerable risk, because the average fire department is not always supplied with hose which will safely withstand the high pressures which modern fire engines are capable of developing. The initial pressure as indicated at the pumps drops off steadily toward the point of discharge and it should be well understood that all pressure which is not thus absorbed by the friction of the flow through the hose is finally manifested as velocity at the nozzle orifice. Hence, unless the several factors that enter are considered together it cannot possibly follow that the best results will be attained.

16 The preceding paragraph expresses the gist of fire-engine performance and the variable features always associated with the work may be definitely stated as follows:

Pumping Capacity: The number of U. S. gallons discharged per minute.

$\left.\begin{array}{l} a \text{ At 120 lb.} \\ b \text{ At 200 lb.} \\ c \text{ At 250 lb.} \end{array}\right\}$ pressure registered at pump.

Water Supply: The source and adequacy of the supply —

d If at draft — height of lift represented by the vertical distance of water surface below center of pump intake

e If at hydrant — capacity as indicated by pressure registered when water is flowing.

[NOTE: High static pressure, as indicated when hydrant is not flowing, is no index to its capacity to discharge any required volume.]

The Layout: Conditions affecting the discharge are —

f Size and character of the fire hose
g Number of hose lines in use
h Lengths of hose stretched between engine and nozzle
i Number of streams played
j Bore and style of nozzles used
k Elevation at the point of discharge.

METHODS OF RATING FIRE ENGINES

17 During a long period, manufacturers of fire engines were unrestricted as to the ratings which they assigned to their machines and when "gallons per minute" were given, no definite discharge pressures were associated with the expressed pumping capacity. In pumping tests, the actual discharge was seldom checked or verified, with the result that comparisons of fire-engine performance were more a matter of guess than of accurate determination. This condition, however, no longer obtains, and the change is largely due to the exacting supervision initiated and persistently pursued under the auspices of the National Board of Fire Underwriters. A corps of expert engineers is constantly in the field for the purpose of investigating and keeping in touch with the fire-fighting situation in all cities, and pursuant to this policy of the insurance interests, it is now generally known what may be expected of fire engines when new and in prime condition, and periodical inspections disclose weaknesses subsequently developed in service.

18 The competency of fire-engine operators is a matter which the manufacturers cannot control, and as the conditions under which fires must be fought are so variable, it is also quite impossible to make fire engines so completely automatic that they can be operated and successfully maintained without fairly skilled attention. The latent possibilities in the best fire engines, hose and other modern appliances can only be realized when such apparatus is well manned.

19 According to the standards formulated by the National Board of Fire Underwriters in addition to the normal rating of a fire engine, expressed in gallons per minute, there must also be sufficient power to expel the rated volume of discharge at a pressure of not less than 120 lb. per sq. in. Inasmuch, however, as certain situations may demand higher initial pressures, the pumps must also be so related to the attending power plant that higher pressures can be realized. Therefore further qualifications are demanded, as follows:

20 One-half the rated capacity should be discharged by the pumps at 200 lb. pressure and one-third the capacity at 250 lb. pressure. Furthermore, fire engines should not be limited to 250 lb. pressure, for the fire departments of our large cities are confronted with the so-called "sky-scrapers," and when water must be forced to the upper floors of these tall structures, the overcoming of the static head alone greatly lowers the effective pressure. However, when

the power of an engine is fully utilized in forcing water to extinguish fire in a lofty building the work and consequent strains are perfectly legitimate. On the other hand, when pressures much in excess of 250 lb. are employed simply to overcome resistance, which could be reduced by a more intelligent use of hose and nozzles, then the work evidently represents not only a waste of power but also very bad practice.

LOSSES IN FIRE HOSE

21 A good engineering axiom to observe is that all work should be accompanied with the least expenditure of force, and the aim here is to express the fact, that many firemen have yet to learn how to smooth their own way. The point can best be illustrated by an example.

22 The inside diameter of fire hose ordinarily used is 2½ in. Cotton-jacketed, rubber-lined hose is the most common kind and lengths of 50 ft. (each such length constituting a section) are the recognized standard. The conventional way of expressing friction loss, or the loss of pressure when water is flowing through the hose, is in terms of pounds per square inch for a unit length of 100 ft. for any given rate of flow stated in gallons per minute. This drop in pressure or so-called friction loss varies somewhat with quality or make-up of the hose; varying diameters and the fact that some makes of hose present a smoother waterway than others are also causes which prevent formulating a coefficient which can be applied to all kinds of 2½-in. fire hose and by which friction losses can be predetermined with precision.

23 It, therefore, follows from the foregoing statements that all tables setting forth friction loss in fire hose should be accepted only as close approximations, and in actual practice the results may, therefore, not agree. However, within the range of velocities and the lengths of hose ordinarily encountered in actual service, it has been found that for water flowing through 100 ft. of average good quality 2½-in. rubber-lined and cotton-jacketed fire hose, the friction loss will be about 14 lb. when the rate of the flow represents 250 gal. per min. Accepting this approximation as a basis, it must also be kept in mind that the friction loss increases in direct proportion to the length of the hose; hence, for the same rate of flow 250 gal. per min. through 200 ft. the total loss would be twice as much as indicated for 100 ft.

24 The effect of varying the rate of flow follows a different and more intricate law, the friction loss increasing more nearly in pro-

portion to the square of the flow. Therefore, if the loss per 100 ft. is 14 lb. when 250 gal. are flowing, it will be quite correct to assume that it will require nearly 2×2 or four times as much pressure, i.e., 56 lb. to discharge 500 gal. through 100 ft., 112 lb. through 200 ft., etc. Stated as formula

$$L = 0.00023\ G^2$$

where L is the friction loss per 100 ft. of 2½-in. rubber-lined, cotton-jacketed fire hose of good quality, and G is the flow in gallons per minute. The results obtained by using this approximate formula agree closely with those in the tables compiled by Mr. John R. Freeman, which appear in Fire Stream Tables, issued by the Inspection Department of the Associated Factory Mutual Fire Insurance Companies, Boston, Mass.

25 The red book, Fire Engine Tests and Fire Stream Tables, published by the National Board of Fire Underwriters, New York, gives the formula:

$$L = 2Q^2 + Q$$

in which L represents the friction loss, as before, and Q the volume flowing in gallons per minute divided by 100. Mathematical precision is always laudable, but in action, firemen could hardly be expected to work out problems other than such as might readily be solved mentally. However, extreme accuracy is entirely unnecessary in fire fighting, but, if guns can be aimed accurately while a battle is raging in war, it would seem altogether feasible that fire engines could be set to work with hose layouts arranged with nozzles compatible with the general conditions of a situation, so that the efficiency latent in modern apparatus and appliances would be more frequently realized.

26 A close approximation, reasonably well gaged according to a correct and definite line of reasoning is preferable to merely a guess or no system at all. Given sufficient study, the difficulties of apparently intricate hydraulic formulæ will vanish most surprisingly, and the real need today is for a greater cultivation of mental capacity to the end that commanding officers in the fire service become especially proficient in obtaining the best possible results with the apparatus at their disposal.

NOZZLE AREAS

27 It will be evident, from what has been presented, that the characteristic of a fire stream develops according to the following interlocking factors:

a The pressure which survives at the nozzle
b The volume of water reaching the nozzle
c The relation of the nozzle bore to *a* and *b*.

The difficulties which attend in the existing system of nozzle bores is largely due to the fact that the diameters vary after the order of ordinary machine-shop reamers, *i.e.*, 7/8 in., 1 in., 1 1/8 in., 1 1/4 in., and it, therefore, follows that the areas of the orifice have increments which increase in arithmetical progression.

28 A more logical system has long been proposed, wherein the nozzle orifices would be multiples of a fixed standard. In accord-

TABLE 1 CALIBER SYSTEM OF STANDARD FIRE STREAMS

Devised and Proposed by Charles H. Fox

Caliber Value	Area Sq. In.	Diam. of Bore, In.	Nominal Diam., In.
25	0.3068	0.625	5/8 Exact
50	0.6136	0.883	7/8 plus 0.008
75	0.9204	1.082	1 1/16 plus 0.019
100	1.2272	1.250	1 1/4 Exact
125	1.5340	1.397	1 3/8 plus 0.022
150	1.8408	1.530	1 1/2 plus 0.030
175	2.1476	1.652	1 5/8 plus 0.027
200	2.4544	1.767	1 3/4 plus 0.017
225	2.7612	1.875	1 7/8 Exact
250	3.0680	1.976	1 15/16 plus 0.039
275	3.3748	2.072	2 1/16 plus 0.010
300	3.6816	2.165	2 1/8 plus 0.040
325	3.9883	2.252	2 1/4 plus 0.002
350	4.2951	2.337	2 5/16 plus 0.025
375	4.6019	2.420	2 3/8 plus 0.045
400	4.9087	2.500	2 1/2 Exact

Note: — The true basic caliber unit is the area of a circle 1/8-in. in diam., vis.: 0.012271875.

ance with this method it was suggested that the 1 1/4-in. nozzle, a size now most common, be designated as "100 caliber." Other nozzles in the same system would be made with orifices of such areas that the water-discharging capacity at the same pressures would be respectively as the caliber number by which each is to be distinguished. Therefore, instead of nozzles measuring 5/8, 3/4, 7/8, 1 1 1/8, 1 1/4, 1 3/8, 1 1/2, 1 5/8, 1 3/4, 1 7/8, and 2 in. in diameter, the caliber system would leave the 5/8- and 1 1/4-in. sizes as before, calling these 25 and 100 calibers, with others, to cover practically the same limits, but with fractional bores arranged in multiple, such as 25, 50, 75, 100, 125, 150, 175, 200, 225, 250, 275 calibers. Table 1 gives these proposed caliber values in more detail and for a complete discussion the reader should consult, Caliber of Fire Streams, Report of Transactions of the International Association of Fire Engineers, 1911.

29 Referring briefly to the gasoline-powered fire engines of today, the development has passed beyond the stage which first marked early endeavors to supplant the horse-drawn steam fire engines. It is now conceded that in all points of efficiency and economy, the well-built gasoline type is superior to the steamer. This statement does not exclude the vital essential of reliability and it is also true that a comparison made upon the basis of their nominal ratings, will show that the better gasoline-driven engines will deliver fully one-third more water when working steadily for any considerable length of time simply because no stoking is required. It is not an easy task to keep the boiler of a steam fire engine up to its maximum working limits, and the elimination of the labor involved explains why this estimate in behalf of the gasoline engine is reasonable.

30 Experience has also shown that the earlier objections with respect to the possibilities of tractive failures by reason of adverse weather conditions were more imaginary than real and as the result of all the advantages attending the introduction of gasoline fire engines, the steam-driven types are passing out of use by force of the dictum which spells the survival of the fittest.

DISCUSSION

Clarence Goldsmith (written). The paper as presented covers the general subject under discussion in a most complete manner and no particular phase of the many items involved has been given undue prominence to the exclusion of others. However, in order to refresh the minds of those who have dealt with some of the problems in times past and to put on guard those who may be called upon to apply some of the principles without having time to give them proper consideration, it may be desirable to treat certain parts of the subject more in detail.

Although the principles of design of the centrifugal pump and of the gasoline engine are well known by the designers of each class of equipment, yet, the observations of the writer during the past ten years show that much trouble is encountered when the centrifugal-pump manufacturer purchases a gasoline engine and attempts to drive his pump with it, or when the manufacturer of a gasoline engine attempts to drive a centrifugal pump with his engine.

The pump manufacturer has tested his pump and determined its most economic speed, capacity and pressure, while the gasoline-engine manufacturer has determined the horsepower delivered at different speeds. In many cases sufficient study is not given to the performance curves of the two pieces of equipment before the capacity and size of each which are to be selected to form the combined motor-driven pump are determined.

In order to have the equipment successful it is fundamentally necessary that the power developed by the gasoline engine at any speed shall exceed that required to operate the centrifugal pump at the same speed. To the casual reader this statement may appear entirely superfluous, yet the failure to meet this axiomatic requirement has cost many manufacturers much time, money and trouble.

The writer is in absolute accord with the statement that the efficacy of fire streams up to 2¼ in. in diameter cannot be materially improved by increasing the nozzle pressure beyond 100 lb. per sq. in. Observations made on a 1½-in. stream by experienced servers showed that the increase in the effective vertical reach when the nozzle pressure was increased beyond 100 lb., was not over 15 ft. up to 150 lb. pressure and that when the nozzle pressure was increased beyond 150 lb. the effective vertical reach decreased. This decrease was evidently due to the increased resistance which the air offered to the surface of the water jet. It is thus evident that a pressure of from 80 to 100 lb. at the nozzle is required to develop efficient streams from deluge sets, turret nozzles, deck guns, ladder pipes and water towers; for hand lines, even when operated by trained men, 60 to 80 lb. is sufficient. There are a few cases however when pressures in excess of 100 lb. may be of some advantage on 2 and 2¼-in. tips. For instance, in overhauling lumber when stacked, overturning walls or breaking through obstructions when the nozzle can be brought up close and horizontal or vertical reach are not desired, pressures as high as 200 lb. at the nozzle enable the operator to take advantage of the added momentum of the stream and accomplish more efficient work.

In passing, it may not be out of place to remark that one of the best methods of placing a fire stream in favorable position for observation is to make the observations at night with the stream directed between a bank of electric lights and the observer.

The capacity of a pumper or steamer should always be stated in gallons per minute against the observed net pressure, the latter to be the actual difference in the pressures observed on the suction and discharge sides of the pump. As an example two cases will be cited: (1) An engine is taking suction from a reservoir 5 ft. below the suction inlet, the friction loss through the suction hose when delivering 750 gal. per min. is 6½ ft., making the total lift 11.5 ft. or 5 lb.; the observed discharge pressure is 115 lb., therefore the net head pumped against by the engine is 120 lb. (2) An engine is taking suction from a hydrant fed by a street main carrying a pressure of 55 lb., when the engine is delivering 750 gal.; the friction loss through the hydrant and branch is 3 lb. and through the suction hose 2 lb., making the total friction loss 5 lb., thus delivering a pressure of 50 lb. on the suction side of the pump; the observed discharge pressure is 170 lb., therefore the net pressure is 120 lb.

Many engineers of steamers have maintained that their engines worked better at draft than when suction was furnished under pressure from a hydrant. It is the opinion of the writer that this is an erroneous notion and he has never seen it substantiated in practice. At any rate it is an established fact that the pressure furnished on the suction side of the pump increases the capacity of a pumper when delivering at pressures above that pressure at which it is rated.

If a pumper is rated at 750 gal. at 120 lb. net pressure and water is delivered to the suction side of the pump at 0 lb. pressure, then it will discharge somewhat more than one-half this quantity at 200 lb. pressure and a little more than one-third the quantity at 250 lb. pressure. Now if water is delivered to the suction side of the pump under 80 lb. pressure the discharge pressure will be 200 lb. and the capacity will remain at 750 gal., and if the water is delivered to the suction under 130 lb. pressure the pump will continue to discharge 750 gal. per min. with a discharge pressure of 250 lb.

It is thus evident that the pressure in the water mains should be taken advantage of in all cases. This is particularly important in large fires where high pressures are required to develop the more powerful streams and to overcome the friction losses in long lines of fire hose which must be laid from the engines located farthest from the fire. Where good hydrant pressures are available, say, from 40 to 60 lb., the more nearly can the rated capacities of the engines be maintained when required to work against discharge pressures greater than 120 lb.

The one and main object of the pumper or steamer is to deliver water at the nozzle at a pressure suitable to develop an efficient fire stream. Now it has been shown that any increase in discharge pressure reduced the delivery of the apparatus; therefore every means should be adopted which will relieve the necessity of raising the discharge pressure. Increasing the carrying capacity of the hose lines from the engine to the nozzle will cut down the friction loss which in turn will cut down the pressure at the engine which is required to deliver the water at the nozzle under suitable pressure.

The use of 3-in. hose will accomplish this result in the most economical manner and practically the same results can be accomplished by siamesing $2\frac{1}{2}$-in. hose lines. For a given quantity of water flowing, the friction loss through a 3-in. hose line is about one-third that through a $2\frac{1}{2}$-in. line of equal length. Two $2\frac{1}{2}$-in. lines siamesed offer about one-fourth the frictional resistance of a single $2\frac{1}{2}$-in. line. It, therefore, is advisable to lay 3-in. hose lines from pumpers to the entrance of buildings and to ladder pipes, turret nozzles, deluge sets and water towers in order to enable the pumpers to deliver larger quantities of water.

The successful fighting of a fire depends much upon the development of powerful hose streams and these in turn depend upon the efficient operation of the pumpers or steamers. An untrained and unexperienced engineer may so operate an engine that is in first-class condition that it will not give as good service as a much poorer engine in well-trained hands. So much depends upon the operation of pumpers and steamers that it is of prime importance that the crews be trained in the operation of their machines at capacity at frequent intervals, say, once a month.

ARTHUR M. GREENE, JR., spoke of the prize offered about 1838 by the Mechanics Institute of Boston for a successful fire engine and described the apparatus brought out by the firm of Braithwaite and Ericsson. Beneath the driver's seat was a bellows, attached by a connecting rod to the rear wheels of the device so that draft was provided while the vehicle was being drawn to the scene of the fire. At the front of the engine was a large air chamber around which was coiled the smoke pipe leading from the boiler.

THE AUTHOR. Mr. Goldsmith expresses some further truths associated with the ratings of fire engines which should be better understood and, referring especially to the work done on the intake side of the pump, i.e. when the water supply is drafted, or, the gain realized when the flow to the pump is under pressure, I am pleased to see these points incorporated in his discussion of the subject, because it frequently occurs in practice, that by neglecting this, either not enough or else too much credit is accorded the engine which is tested.

It also is true that effective work may frequently be done by using streams of exceptionally high velocity, all of which is justified when circumstances warrant such use of excessive power. The writer's argument however should be taken to center on the desirability of attaining the maximum effect at the nozzle with the minimum waste of power and, as Mr. Goldsmith indicates, this can be accomplished by using either 3-in. hose instead of 2½-in. or the friction losses between the pump and the nozzle may be materially reduced by using double leads of 2½ in. hose, joined into one at a point near to the nozzle.

Next to an intelligent management of the engine and pumps and yet of equal importance, comes the judicious disposal of the avenues through which water is forced to the nozzle orifice, and it is to this end that the writer's paper was written. Much more could be mentioned, but the salient features only could be given adequate expression.

No. 1697

AN ELECTRICAL DEVICE FOR MEASURING THE FLOW OF FLUIDS IN PIPES

By Jacob M. Spitzglass, Chicago, Ill.
Member of the Society

This paper deals with the theory and experimental development of an electrical device in which an ammeter and watt-hour meter are utilized in measuring the flow of fluids in pipes. The electric current which is used is regulated by the differential pressure of the fluid passing through the pipe.

The principle of the device involves a combination of the physical laws governing the flow of fluids in pipes and the flow of an electric current. The units of flow measurement are represented by general equations covering the relation between the velocity of the fluid in the pipe and the differential column obtained by the device. For the units of the electrical measurement, in the standard adopted the maximum capacity of flow is represented by a current of 1 ampere at a constant pressure of 40 volts.

The diagrammatic relation of the units involved in the electric measurement of the flow is shown by two curves; one in the form of a parabola representing the relation of differential head to the current — the other in the form of a hyperbola representing the relation between the current and the corresponding resistance in the circuit.

A summary is given of the experimental work in preparing a resistance which would vary according to the relation established, and which would operate under the usual conditions of flow. Particulars are given of the early experimental devices and of the improvements made until a satisfactory model was obtained, and the method of testing is discussed and a typical run described in detail.

In conclusion, a number of instances are cited where the instrument described has made possible the adoption of central measuring stations in large manufacturing plants, resulting in improved economy of operation.

DESPITE the fact that the science of mechanical engineering is much older than that of electrical engineering, its methods of measurement are nevertheless in many respects much behind those afforded by the latter. A striking example of this is found in a comparison of the methods of measuring fluid motion in pipes and the flow of an electric current. The instrument used for the electric current is simple and direct-reading, and while there have been many excellent devices adopted for measuring the flow of fluids in pipes, it has been quite generally agreed that an instrument similar to the ammeter or wattmeter would be of great value.

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

2 Recently the writer had the privilege of experimenting with a flow-measuring device in which these instruments are applied. Measurement is accomplished by means of an electric current which is so regulated by the differential pressure of the flow that it represents the amount of fluid passing through the pipe.

3 The main features of the device are shown diagrammatially in Fig. 1. The U-tube, partly filled with mercury, is made to balance the impact pressure of the flow in the pipe by the rise of mercury in the low-pressure side of the tube. The mercury column also forms a part of the electric circuit, as shown in the figure. This electric circuit contains a fixed external resistance R_1 in series with a

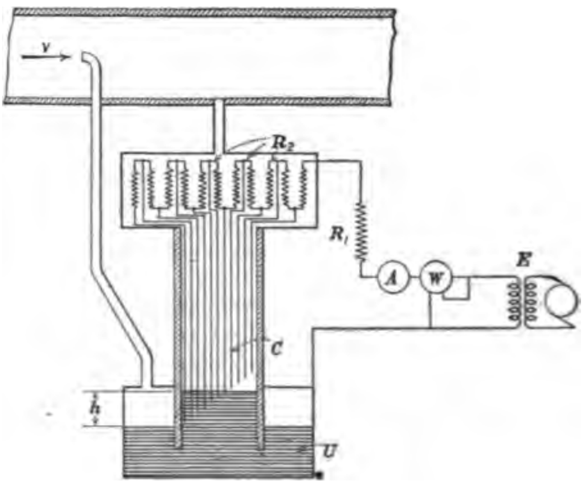


FIG. 1 DIAGRAMMATIC SKETCH OF AUTHOR'S ELECTRICAL DEVICE FOR MEASURING FLOW OF FLUIDS IN PIPES

variable internal resistance R_2, a constant electromotive force E, an ammeter A and a watt-hour meter W. In the contact chamber C, which forms the low-pressure side of the U-tube, there are a number of conductors of varying length placed above the mercury column, and as the mercury rises it makes contact with one conductor after another. The variable resistance R_2 is subdivided by these conductors into resistance steps corresponding to the varying length of the conductors, so that the rise and fall of the mercury column varies the amount of resistance and thereby regulates the amount of current passing through the circuit.

4 The basic principle of the device accordingly involves the laws governing the flow of fluids through pipes along with those

governing the flow of an electric current. The problem of establishing the theoretical relation between these fundamental laws offered little difficulty because of the similarity between the units of flow measurement, such as pressure and velocity, and the units of electric measurement, such as voltage and current. On the other hand, the attempts to apply the theory to a working model were beset with numerous difficulties, and the obstacles that were over-

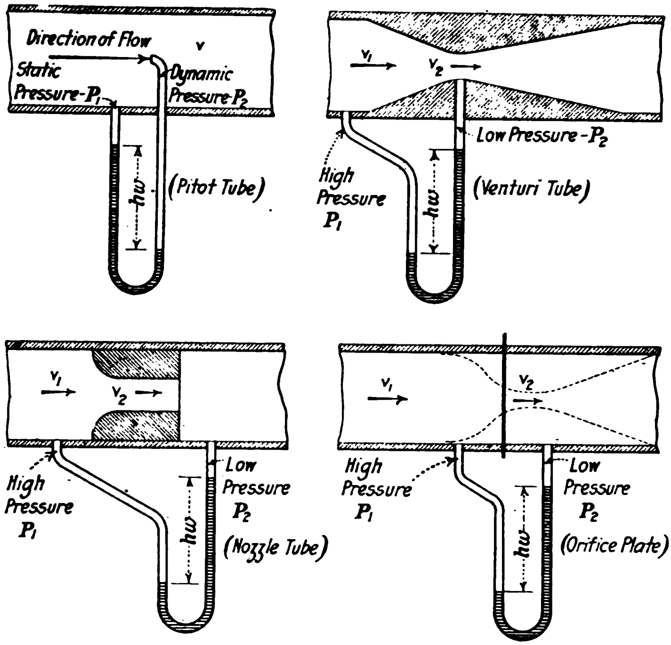


FIG. 2 METHODS OF DETERMINING VELOCITY PRESSURE

come during the long period of experimental work presented many problems which are briefly dealt with in later paragraphs.

UNITS OF FLOW MEASUREMENT

5 The relation between the pressure and velocity of fluids in its simplest form is represented by the well-known equation

$$\frac{v^2}{2g}\gamma = P_2 - P_1 = hw$$

or

$$v = \sqrt{\frac{2g(P_2 - P_1)}{\gamma}} = \sqrt{2gw}\sqrt{\frac{h}{\gamma}} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [1]$$

where v and γ represent the velocity and density of the fluid; $(P_2 - P_1)$ the equivalent differential pressure; h the height and w the density of the liquid column balancing the differential pressure of the flow.

6 This differential pressure hw may be obtained, as shown in Fig. 2, either directly by balancing the difference between the dynamic and static sides of a pitot tube inserted in the line, or indirectly by balancing the difference between the high- and low-pressure sides of a venturi tube, nozzle tube or orifice plate. In the case of the pitot tube, the differential column in the U-tube represents the flow or motion existing at the given section of the line, but in the venturi tube, nozzle or orifice, the column obtained represents the *change* of motion produced by the artificial obstruction of the passage at the given section of the pipe.

7 In any case, however, the relation between the differential column thus obtained and the velocity of the fluid in the pipe may be represented by Equation [1], provided there is introduced the experimental coefficient derived for the given tube or orifice. Thus in general,

$$v = C\sqrt{2gw}\sqrt{\frac{h}{\gamma}} \quad \text{[2]}$$

The volume of the fluid Q passing per unit of time through an area A is given by the equation

$$Q = Av = AC\sqrt{2gw}\sqrt{\frac{h}{\gamma}} \quad \text{[3]}$$

the corresponding weight G per unit of time is

$$G = Q\gamma = (A\sqrt{2gw})C\sqrt{h\gamma} = KC\sqrt{h\gamma} \quad \text{[4]}$$

and the total weight for a given period of time t is

$$Gt = KCt\sqrt{h\gamma} \quad \text{[5]}$$

UNITS OF ELECTRIC MEASUREMENT

8 Having adopted the foregoing general equations for the flow of fluids, the corresponding electric measurements may be covered by the following definitions:

I = current in amperes flowing through the electric circuit of the measuring device. The instrument was designed so as to have one ampere represent the maximum capacity of the meter.

E = electromotive force of the circuit. A uniform pressure of 40 volts was selected to represent the average density of the fluid measured

Wt = amount of electric energy expended in the circuit of the device in a period of time t

R = total resistance of the circuit in ohms

F = rate of flow in the pipe corresponding to the electric current in the circuit, or the ratio of G to I. F is the "indicating factor" of the flow meter.

T = total amount of flow or weight of fluid corresponding to the electric energy passed through the circuit. T is the ratio of G to W, and is designated as the "totalizing factor" of the flow meter.

9 Since by definition, $FI = G$ and from Equation [4] $G = KC \sqrt{\gamma}\sqrt{h}$, therefore $FI = KC\sqrt{\gamma}\sqrt{h}$, or

$$I = \frac{KC}{F}\sqrt{\gamma}\sqrt{h} \quad \ldots\ldots\ldots\ldots \quad [6]$$

The value of K is constant for any given set of conditions as determined from Equation [4]. The value of C, depending upon the particular design of the tube or orifice, is also constant for any given case.

10 To find the value of F, let $I_{max.}$ be the current in amperes corresponding to the maximum capacity of the meter, $G_{max.}$, which in turn corresponds to the maximum differential column $h_{max.}$ From Equation [6]:

$$I_{max.} = \frac{KC}{F}\sqrt{\gamma}\sqrt{h_{max.}} \quad \ldots\ldots\ldots\ldots \quad [7]$$

whence

$$F = KC\sqrt{\gamma}\frac{\sqrt{h_{max.}}}{I_{max.}} \quad \ldots\ldots\ldots\ldots \quad [8]$$

and since $I_{max.}$ is equal to unity,

$$F = KC\sqrt{\gamma}\sqrt{h_{max.}} \quad \ldots\ldots\ldots\ldots \quad [9]$$

11 The quantity $\sqrt{h_{max.}}$ is called the characteristic or the scale of the given meter and it determines the capacity of the meter, depending upon the amount of differential column $h_{max.}$ which the meter is able to develop and record.

12 Combining Equations [6] and [9],

$$I = \sqrt{\frac{h}{h_{max.}}} \quad \ldots\ldots\ldots\ldots \quad [10]$$

It is interesting to note that $h/h_{max.}$ represents the relative value of the differential column for a given rate of flow, and 100 $h/h_{max.}$ is the percentage variation of the head in any given meter. From Equation [10] it follows that in order to represent the amount of flow, the current I should be numerically equal to the square root of the relative height of the mercury column in the U-tube of the meter. From the same equation,

$$h = I^2 h_{max.} \qquad [11]$$

That is, the height of the column for a given flow is numerically equal to the constant $h_{max.}$ times the square of the current flowing through the circuit.

13 From Ohm's law ($E = IR$) we obtain by substitution

$$R = E\sqrt{\frac{h_{max.}}{h}} \qquad [12]$$

That is, the resistance R in the circuit should be numerically equal to the voltage divided by the square root of the relative height of the differential column.

14 It remains to determine the value of T, the "totalizing factor" of the instrument, or the ratio of G to W. Since, $Wt = EIt$ and by definition $TWt = Gt = FIt$, therefore

$$T = \frac{F}{E} \qquad [13]$$

That is, the totalizing factor of the meter is equal to the indicating factor divided by the voltage in the circuit.

15 Fig. 3 shows diagrammatically the relation of the units involved in the electric measurement of the flow. The parabolic curve to the right shows the variation of the current in the electric circuit representing the capacity of the flow and corresponding to the percentage variation in the differential column balancing the velocity pressure of the flow. This curve represents the solution of Equation [11]. The hyperbolic curve to the left of the diagram shows the relation between the current and the corresponding resistance at the given voltage of the electric circuit. The solution of Equation [12], or the relation of R to h, is obtained indirectly by following from a given value of R on the resistance curve to the corresponding value of h on the current-and-capacity curve.

16 It will be observed that the diagram does not include the first 10 per cent of the flow capacity inasmuch as this represents

only 1 per cent of the differential column, which is as low as a practical device is able to measure with any degree of accuracy. This disadvantage, however, is offset by the fact that the scale of the flow meter can be so chosen that the desired measurements will fall within the active part of the scale.

PRACTICAL APPLICATION

17 After the relations between the various factors had been determined, the problem reduced itself to the construction of a

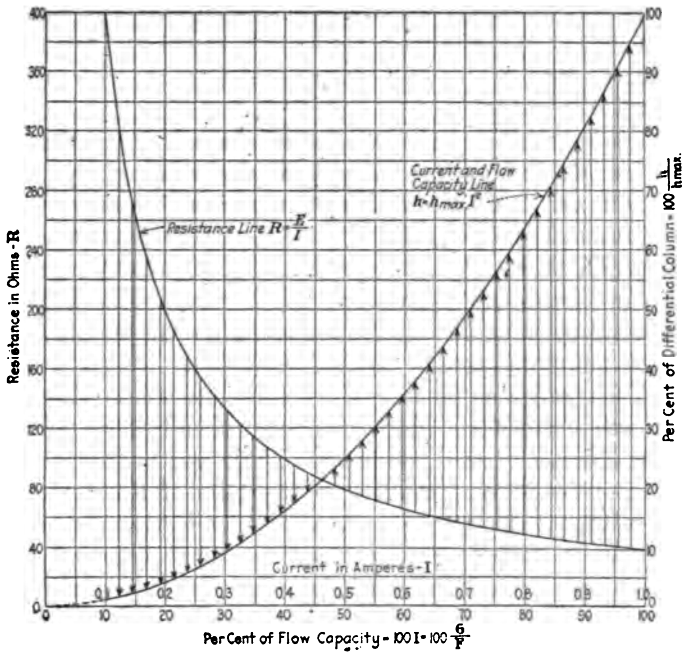


FIG. 3 DIAGRAM SHOWING RELATION BETWEEN UNITS OF FLUID FLOW AND ELECTRIC UNITS

resistance which would be regulated by the differential column of the flow according to the solution of Equation [12]. The first attempt in this direction was made by inserting a continuous resistance coil in a water manometer where the height of the water column would reduce the amount of resistance in the coil by short-circuiting the part immersed in the water. The first trials were made with direct current, and while it was anticipated that electrolytic action would be set up between the metal conductors and the water, it was nevertheless expected that this action would not take place when alternating current was used.

18 There was little information available on the subject and it was therefore necessary to determine experimentally the amount of resistance needed. After obtaining some idea of the amount required and being hindered by the accumulation of deposit in the container, which was at first attributed to the electrolytic action of the direct current, provisions were made to continue the experiments with alternating current. It was discouraging, however, to note that practically the same action took place between the metal conductors and the water when alternating current was used. Repeated analyses of samples of the deposit disclosed that it was a formation of oxide, due to the corrosion of the metal conductors in contact with the water.

19 Besides the formation of deposits, there were other disadvantages in short-circuiting the metallic resistance by a water column. On one hand the conductivity of the water varied with its hardness, thus introducing a variable resistance in the part of the column which was covered by water, and on the other the vapors formed over the surface of the water had a tendency to short-circuit the resistance coils, again introducing a similar variable in the portion of the column above the level of the water.

20 When repeated attempts to eliminate these defects had failed it was decided to adapt mercury instead of water for the regulating column of the instrument. The use of mercury, however, necessitated radical changes in the form of the device. The effective column of mercury for the average velocity pressure would be too small to cover the continuous resistance coil and produce the desired regulation of the resistance. It was therefore found necessary to regulate the resistance by steps through conductors coming in contact with the top of the mercury column. Fig. 4 shows the elementary form of experimental device adapted for this purpose.

21 In this elementary device the successive conductors were divided into steps of equal height from the zero level of the mercury column. The electromotive force of the circuit was maintained constant at 40 volts. The resistance of the circuit, which amounted in all to 400 ohms, was subdivided by the contact rods into 40 consecutive steps, and the amount of resistance provided for each step was determined from Equation [**12**]. Using these values the maximum current of 1 ampere corresponded to the minimum resistance of 40 ohms, while the minimum current of 0.1 ampere corresponded

to the maximum resistance of 400 ohms. Between these limits the rise and fall of the mercury column produced by the variation of the head on the high-pressure side of the U-tube would vary the amount of resistance in the circuit in accordance with the hyperbolic curve shown on the left of Fig. 3, thereby regulating the amount of

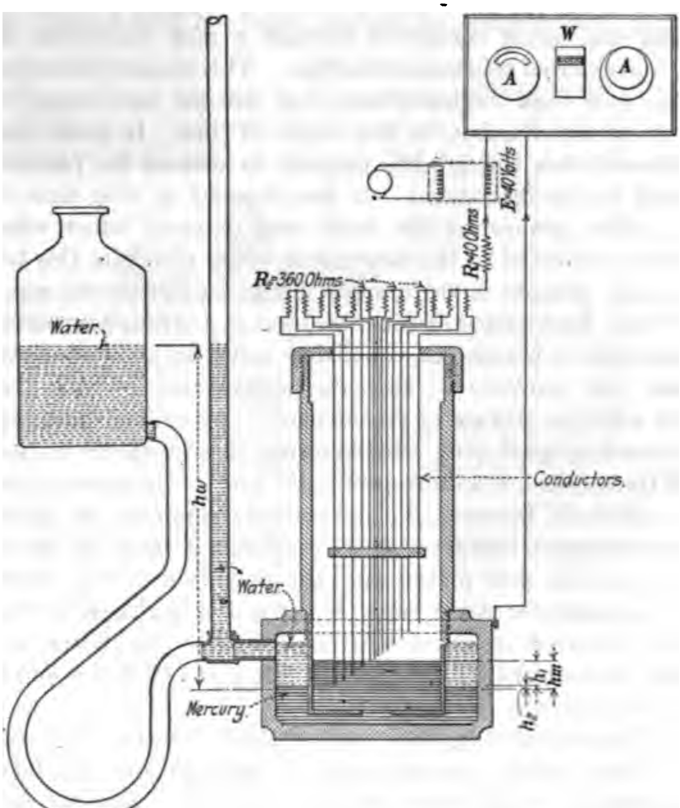


FIG. 4 DEVICE IN WHICH RISE OF MERCURY COLUMN REGULATES AMOUNT OF CURRENT

current passing through the circuit in accordance with the parabolic curve on the right of the figure.

22 The operation of the elementary device was successful from the very start, but only as long as the contact chamber was kept free from water did the regulation of the current correspond with the variation of the head or the height of the mercury column. On the other hand, the instrument when equipped with an oil seal and

connected to a pitot tube in the steam line could not be made to operate properly. When left under pressure the oil would leak through the fiber plug and allow water to get into the contact chamber, which would immediately put the instrument out of order.

23 To overcome this action the body of the instrument was extended to include also the resistance coils attached to the contact rods and the circuit completed through a plug connecting the internal and external resistances together. This change eliminated the leakage of oil from the instrument, but still the water could not be kept out of the chamber for any length of time. In some cases the water would blow through the mercury as soon as the pressure was admitted to the instrument. It was thought at that time that a bypass valve connecting the static and dynamic tubes when the pressure is admitted to the instrument would eliminate this trouble. After many changes in the original design an instrument was made which for a short period of time was used as a steam-flow meter. In this instrument, besides the equalizing valve, an additional overflow chamber was provided to keep the water from reaching the conductors over the surface of the mercury. A terminal post replaced the connecting spark plug, thus allowing an adjustment of the position of the conductors with respect to the level of the mercury column. These additions, however, did not entirely eliminate the possibility of water coming in contact with the resistance elements of the device. With a uniform flow in the pipe the operation of the instrument would continue for some time, but when a slight disturbance of pressure occurred in the line it would cause the water to blow through the mercury into the contact chamber and this would immediately discontinue its operation.

24 Notwithstanding this objectionable feature, the convenience of the electric measurement of the flow and the fact that the instrument would function as long as there was no water in the contact chamber have encouraged further experimenting for the elimination of defects. The problem was finally solved in the fall of 1917 by the addition of a mercury seal connected in parallel with the working base of the instrument. The function of the mercury seal in this case is quite similar to that of a fuse plug in an electric circuit, with the added advantage that it is self-replacing.

25 The principle of its operation is illustrated in Fig. 5, which shows a section of the meter body and seal chambers. It can be seen that the U-tube joining the two compartments of the seal will contain a column of mercury about one-half the height of the column

in the meter body. Under normal operation the mercury column in the seal acts in unison with the mercury column in the meter body and does not interfere with the proper transmission of the differential pressure in the meter. When, however, some disturbance of pressure occurs in the line sufficient to break the seal, the mercury spreads over the larger area of the compartment, equalizes

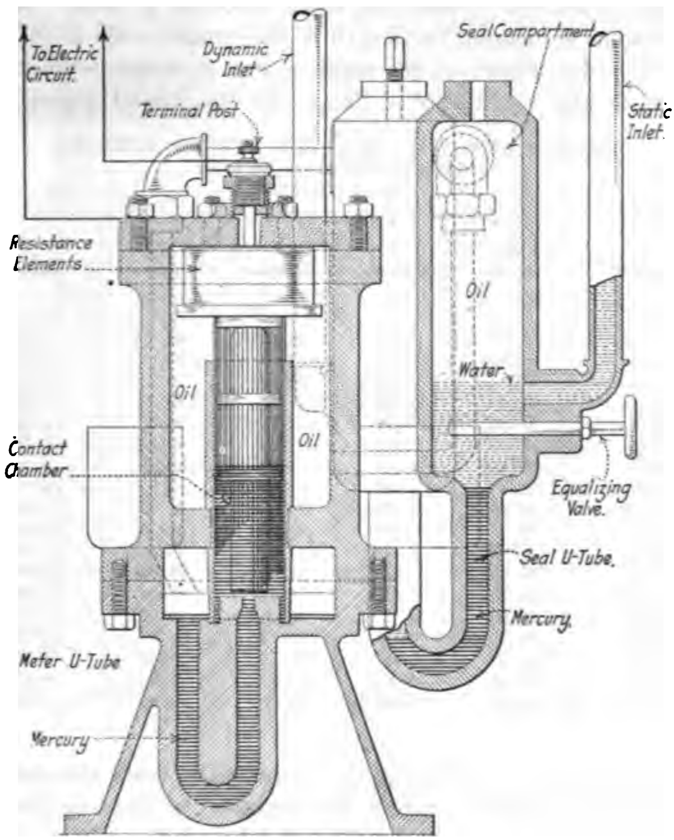


Fig. 5 Latest Type of Author's Flow Meter with Mercury Seal

the pressure in the two compartments in the same manner as would an automatic opening of a bypass valve, and thus prevents the breaking of the higher column in the U-tube. As soon as the abnormal differential pressure is released, the mercury drops back into its place and reëstablishes the necessary seal between the two compartments. In this model a large quantity of oil is trapped in the two compartments of the seal and in the meter body, eliminating the possibility of water blowing through the mercury and coming in contact with the resistance elements of the meter.

26 In the latest type of the flow-measuring device the contact rods were changed from their former equal spacing above the zero level to spaces varying in height so as to give at each step equal increments of current representing equal amounts of flow. This was accomplished by gaging the length of the contact rods to follow the parabolic curve shown at the right in Fig. 3, which represents the solution of the equation $h = I^2 h_{max.}$ In the actual gaging of the

TABLE 1 DATA OBTAINED FROM TEST OF ELECTRIC FLOW-MEASURING DEVICE
(E_1 = 112 volts)

h in. water	E_2 volts	I amperes	h in. water	E_2 volts	I amperes
0.45	40.15	0.110	16.12	39.45	0.592
1.06	40.15	0.170	17.43	39.45	0.613
2.06	40.15	0.235	19.44	39.35	0.642
2.81	40.05	0.260	21.00	39.25	0.666
3.56	40.05	0.280	22.75	39.25	0.689
4.00	39.95	0.305	24.00	39.25	0.713
4.56	39.95	0.326	25.25	39.25	0.740
5.125	39.85	0.347	27.00	39.25	0.765
5.875	39.85	0.370	29.125	39.15	0.790
7.00	39.75	0.392	30.00	39.05	0.814
7.56	39.65	0.418	31.625	39.05	0.838
9.125	39.75	0.434	34.06	39.05	0.860
9.875	39.75	0.458	35.37	39.05.	0.882
10.812	39.65	0.480	37.00	39.00	0.905
11.25	39.55	0.505	38.75	38.90	0.930
12.50	39.65	0.523	40.75	38.80	0.953
13.56	39.55	0.542	42.875	38.70	0.978
14.87	39.55	0.566	44.75	38.45	1.000

contact rods $h_{max.}$ is taken as the distance between the zero level of the mercury and the end of the contact rod showing the maximum flow. The successive heights of the rods for the given equal increments of current are determined more conveniently by differentiating Equation [11], $h = I^2 h_{max.}$ The first differential, or $dh = h_{max.}\ (2\ IdI + dI^2)$, represents the respective increments of h corresponding to the given increments of I. The second differential, or $d^2h = 2h_{max.}\ dI^2$, represents the respective difference in the successive increments of h, from which it is noted that the distance between the successive contact rods is increased uniformly by the constant quantity $2\ h_{max.} dI^2$.

TESTING FOR ACCURACY

27 The accuracy of a flow-measuring device is necessarily made up of two factors. One is the accuracy with which the device registers the differential pressure equivalent to the flow in the pipe, and the other is the accuracy with which it will indicate or record this differential pressure or the equivalent units of flow. In the usual application of the flow meters, where the pitot tube, the venturi

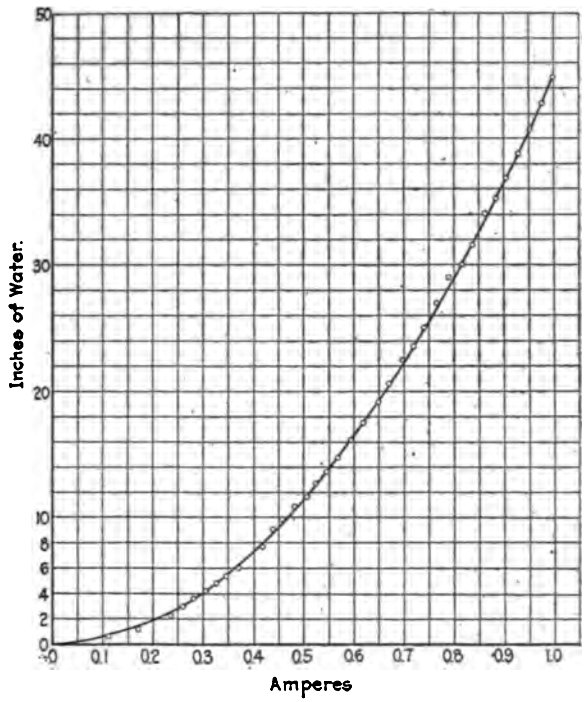


FIG. 6 COMPARISON OF CURVE OBTAINED FROM EQUATION [11] WITH DATA PLOTTED FROM TABLE 1

tube or the orifice plate is used for obtaining the differential pressure of the flow, the efficiencies of these devices have been determined by numerous tests and are at present well known. Their nature is such that a given flow will always produce the same effect under the same conditions, since they do not possess any working parts to vary the relative effectiveness of their operation. On the other hand, the indicating or the recording elements of such devices may vary from time to time depending upon the condition of the moving parts in these elements. It is necessary, therefore, to have convenient means for testing them in order to ascertain their accuracy at frequent intervals.

28 Table 1 gives the data of a typical test on the resistance element of the flow-measuring device, showing the relation between the differential column and the corresponding readings of the electric current.

29 In Fig. 6 the points obtained from the test are indicated by the small circles, and for comparison a curve is shown giving the theoretical relations according to the equation $h = I^2 h_{max}$. In this test the differential pressure was obtained by varying the height of a water column connected to the dynamic side of the meter. The electric current was supplied to the indicating instrument through a transformer, and the primary or line voltage was kept constant by varying the field of the generator supplying the power, thus approximating the actual condition of an average installation. The secondary voltage varied due to the transformer regulation, but the resistance element of the instrument is designed to compensate for such regulation, so that the indicated variations of current gave a fairly accurate measurement of the differential pressure.

CONCLUSION

30 The fact that the flow of fluids can be measured electrically has made possible many important installations where no other method could be employed. In one instance a large manufacturing concern had been contemplating for a long time the adoption of a system for measuring the amount of steam, air and water used by its various departments, but was hindered by the fact that the various lines were distributed over a wide area and in some places were carried through sub-basements, where measuring devices would be inaccessible; also much time and a large force of employees would be required to read the various instruments about the plant and to integrate the recording charts. As soon as the concern discovered that flow could be measured electrically, that the indicating instruments did not have to be located where the flow was to be measured, and that the integrating device was merely a watt-hour meter which integrated the flow independently of the other instruments, a measuring system was instituted for all its products and many wasteful uses of power were thereby eliminated and an accurate distribution of costs established throughout the factory.

31 The adaptability of the integrating feature to the electrical measurement of flow is of great importance since the readings from the watt-hour meter are more accurate than those taken from the recording ammeter and just as accurate as the instantaneous readings of the indicator. This feature therefore eliminates the necessity of planimetering the charts and insures accurate results for any variation of flow.

32 When measuring the flow of steam generated by a battery of boilers the flow indicators are placed in front of each boiler, showing the momentary performance for the guidance of the fireman. At the same time, supplementary recorders connected electrically with the indicators are placed conveniently for the supervision of the chief operator.

33 Recently the manufacturers of water gas adopted the use of low-pressure exhaust steam for gas generation, which created an urgent demand for a measuring device to operate intermittently, varying every few minutes from zero to maximum. After many unsatisfactory trials of mechanical devices the electrical method of flow measurement was adopted, as this made it possible to measure successfully the steam required for the manufacture of water gas and resulted in a great economy.

34 The main advantage, however, of the electrical method of flow measurement is the accuracy with which the differential pressure is transmitted through a mercury column, which column is not hindered in its movements by any mechanism and is therefore free to attain the true level under all conditions of flow. Furthermore, the electrical instruments used to register the flow can be checked at any time without interfering with the operation or installation of the measuring device.

DISCUSSION

Herbert B. Reynolds (written). The flow meter described in this paper certainly has a great many advantages and should have a very wide field of application, due to its remote indicating features.

As stated in the paper, the resistances and contact points are so arranged that the current flowing through the ammeter varies directly as the fluid flow. If direct current is used, the deflection of the indicating needle will be in direct proportion to the fluid flow.

In other words, it would be possible to provide a meter with a uniform scale, which would be more desirable than the scales found on most flow meters. I am under the impression that alternating current is used exclusively for the operation of these meters, which means that, although the current strength varies as the liquid flow, the indications vary as the square of the liquid flow, due to the well-known scale characteristic of all alternating-current meters. However, it seems to me that a uniform deflection could be obtained by simply adjusting the resistances and contacts, as is done in obtaining a straight-line law between the liquid flow and current flow. I would be interested to know if the author has made any attempt to obtain a uniform scale in this manner.

In a great many cases it is desirable to have an indication of the total flow through two or more pipe lines on one dial. It seems to me that this instrument could be used for this purpose, by placing a mercury column, with its resistances and contact points, on each pipe line. The terminals of these mercury columns could then be connected in parallel or in series, with the indicating meter in the main circuit. This probably has been considered and found impracticable; however, if a scheme like this could be worked out, it would broaden the field of the instrument.

Geo. F. Gebhardt (written). During the past 20 years I have had exceptional opportunity for testing practically all types of steam flow meters exploited in this country, both in the laboratory and under service conditions. Most of these devices gave consistent results in the laboratory where the various factors entering into the calculation of the weight of flow were known, but under service conditions the indicators of the dial and chart frequently departed considerably from condenser weights.

A study of the discrepancies showed that the error laid chiefly in the "factor" for converting velocity-pressure variations to weights, rather than in any inherent defect of the meter. These "factors," as established by the manufacturers, are based upon what are assumed to be service conditions for the proposed installation, and naturally failure to predict the true conditions will result in error. In several instances where the meter has been discarded as unreliable it was ascertained that the factors were merely calculated and that no calibration tests of any kind had been made.

The steam flow meter has been wonderfully developed during the last decade, and where the rate of flow does not fluctuate widely and the steam conditions are fairly constant it is a dependable and accurate means of measuring the weight of the flow. However, where the rate of flow fluctuates rapidly and there is considerable variation in the pressure and quality of the steam, the indicated readings are generally not in accordance with the actual weights flowing. Proper calibration under service conditions greatly reduces the error but few plants are equipped for this purpose.

The many advantages of the electrical method of control over the mechanically operated mechanism are enumerated in the author's paper. My own experience has shown the electrical control to be accurate and dependable. Meters operating on this principle, however, are limited to plants supplied with current and are subject to error on account of voltage variation.

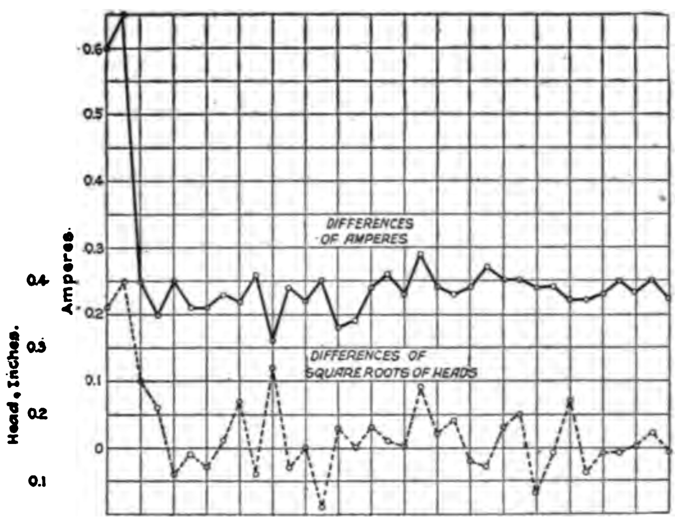


FIG. 7. DIFFERENCES BETWEEN SUCCESSIVE SQUARE ROOTS OF HEADS AND AMPERE READING FOR 35 INCREMENTS OF PRESSURE

I would like to ask the author if he has had any success in applying his electrically-controlled meter to pulsating flow, as in connection with high-speed reciprocating engines or slow-speed reciprocating pumps.

GEO. H. BARRUS (written). The paper contains a table giving the amounts of current which were shown by this instrument under various increasing heads, starting with 0.45 in. water pressure and rising by 35 increments of varying amounts to a total of 44.75 in. water pressure. *It is interesting to analyze this table and see with what degree of sensitiveness or accuracy the indications of current respond to the gradual increase of head.* The analysis can best be made by taking the successive increments of head and comparing them with the corresponding increments of current. To be comparative the square-roots of the heads are first obtained and the differences of the successive square-roots taken. Then the differences of successive currents are found, using for convenience the column of amperes. These two sets of differences are as follows:

Number	Differences between Successive Square Roots of Heads, In. Water	Differences between Successive Ampere Readings, Amperes	Number	Differences between Successive Square Roots of Heads, In. Water	Differences between Successive Ampere Readings, Amperes
1	0.36	0.06	19	0.15	0.21
2	0.40	0.065	20	0.24	0.29
3	0.25	0.025	21	0.17	0.24
4	0.21	0.02	22	0.19	0.23
5	0.11	0.025	23	0.13	0.24
6	0.14	0.21	24	0.12	0.27
7	0.12	0.21	25	0.18	0.25
8	0.16	0.23	26	0.20	0.25
9	0.22	0.22	27	0.08	0.24
10	0.11	0.26	28	0.14	0.24
11	0.27	0.16	29	0.22	0.22
12	0.12	0.24	30	0.11	0.22
13	0.15	0.22	31	0.14	0.23
14	0.06	0.25	32	0.14	0.25
15	0.18	0.18	33	0.15	0.23
16	0.15	0.19	34	0.17	0.25
17	0.18	0.24	35	0.14	0.22
18	0.16	0.26			

These differences are plotted on the chart of Fig. 7 in which the upper record is that of the ampere differences and the lower one, the differences of head computed in the manner stated. It will be seen at a glance that when viewed in this way there is great disparity between the two records. In only a few instances do the variations in current respond immediately and proportionately to those of the head. No doubt the gaps between the contact points of the instrument are responsible to some extent for these conditions because no change of current would be likely to occur while the head is changing from one point to the next, but it is not clear that other causes are absent. It would be interesting to have further information regarding this matter and it is suggested that such information might be obtained by taking a series of electrical readings corresponding to much smaller increments in head, say one-quarter or one-half an inch, throughout the whole scale. The comparison might also be extended by getting a series of readings going down the scale, and demonstrate whether contact is made or broken at the same reading of head whether rising or falling.

The records shown on the chart bring out the fact that the instrument is not scientifically accurate. Nevertheless the statements offered in Par. 30 of the paper indicate that its utility as a practical device for the electrical measurement of the flow of fluids has been well proved in actual use.

A. H. Anderson (written). I have had a wide experience with fluid meters since the old Sargent steam meter which accomplished something which is never attempted in present day meters. It compensated automatically for the change of steam pressure within a wide range. The Sargent meter was applicable only in sizes of pipe under 6 in. diameter and I do not know that it is used today in any size. Then came the Gebhardt steam meter, patented in 1905 by Prof. Geo. F. Gebhardt, of the Armour Institute of Technology, which was the fore-runner of the Republic meter.

No mention is made by the author of the error introduced into his calculations by change of steam pressure or quality of steam. It is also essential that the pipes connecting the meter to the pitot tube be kept full with water.

It seems to me that the value of the meter depends upon the selection of the correct coefficient for the pitot tube or orifice.

I have seen many fluid meters in operation, but in only one instance has any provision been made to check the accuracy of the installation by actual weight. Where such a check has been provided, meter inaccuracies have been detected in time for early correction. Fluid meters today are examples of marvelous ingenuity, but their presence must not lull us into false security. Every plant using fluid meters should also have a weighing device in the feed line for occasional use.

One example of the utility of the Republic meter is on a boiler in the Union Stock Yards, Chicago. Normally the boiler is developing 750 hp. but let the inspection door be opened for about one minute and the fluid meter will begin to drop until the horsepower is about 500 and after the door is closed, several minutes elapse before the meter returns to normal. The variation, of course, is due to the chilling effect of the excess air.

William B. Fulton asked if the author had had any experience with his meter in connection with thickened fluids, such as paper-mill stock containing 3 or 4 per cent stock, the remainder being water. The author suggested the use of a pitot tube with large openings. The change of resistance, due to the thickened fluid, would be recorded by the instrument, if the connection did not clog, which would not occur if the openings were large enough. The flow could be measured positively. The instrument would give the velocity from which could be computed the volume.

Austin R. Dodge said that the electric type of flow meter, as the author had stated, could be easily checked at any time but this was true of other types of meters and did not depend upon the electrical method of operation.

Also it seemed to him that with an automatic integrating attachment it was quite as simple to obtain the total flow as with the watt-hour meter.

E. G. Bailey said that in metering steam there was a minimum of four distinct fluids to deal with: steam, the condensed water in the connecting pipes, air, which invariably separated and gave a false head, and the mercury in the measuring device. In most practical installations, to keep each of these fluids separate and in its proper place so as to prevent a false head was a big part of the battle, and anyone who could add two other fluids, oil and electricity, and keep them in their proper places certainly deserved a great deal of credit. Looking at it in another way, the fluid meter problem involved two factors, a means of obtaining a pressure difference and a means of measuring this pressure difference. The author has made 50 per cent more of a problem of it in using this pressure difference to control an electric meter electricity, adding a third and completely distinct step.

There was a fifth mobile substance which had been mentioned in the discussion of paper stock, namely scale, sediment and other matter which invariably flowed through steam lines. This should be considered carefully in view of its possible effect on any type of flow meter.

Mr. Bailey asked the author how near was the specific gravity of the oil used in the oil seal to that of the water and how was this taken care of in the calibration of the instrument to prevent a false head in the oil seal.

Wm. B. Gregory said that while he had had considerable experience with pitot tubes and other means of measuring water he had never used an instrument of the type mentioned in the paper. He expected to use such a meter during the summer in tests at a large sugar refinery and wished to know in how far the results could be depended upon and what the probable error in operation would be.

The Author. With reference to Mr. Reynolds' discussion, the purpose of the straight-line law between the liquid flow and current flow was necessary mainly for enabling the watt-hour meter to integrate the total current and thereby represent the total flow. It results in a uniform deflection on direct-current instruments, which are used sometimes, while on alternating current instruments, the deflection is not uniform.

The measurement of the total flow through two or more pipe lines on one dial has been found possible in actual practice in many cases. In several large boiler installations, there are two outlets to each boiler and one watt-hour meter totalizes the flow of both outlets.

With reference to the other written discussion, Mr. Barrus probably appreciates the fact that the accuracy of the meter lies in the cumulative indications and not in the difference between one contact point and the other. Mr. Barrus, in his last paragraph, stated that the records shown on the chart bring out the fact that the instrument is not scientifically accurate. This is true. When measuring steam with a pitot tube or an orifice plate, which are only accurate within two per cent at the best working conditions, it would be absolutely out of the question to look for a scientifically accurate instrument. All the errors found by Mr. Barrus are within one per cent, which is absolutely insignificant for commercial purposes.

The author, in answer to a question by Alan E. Flowers regarding the relation between the average and maximum flow as measured by the pitot tube replied that the question opened an entirely different field and one which he would not, at that time, go into. In reply to a second question by Mr. Flowers about tests which would show the relation between the total flow and the indication of the meter the author said that he believed that Professor Gebhardt had answered it by stating that all meters were accurate enough in the laboratory under proper conditions of operations and upon tests made under these conditions only could conclusions be based. For accuracy, the customer must test his meter in his own plant and under the actual conditions which prevail, thus establishing a "factor" for its use. Tests of the meter, showing its accuracy, could be supplied in great quantity.

With regard to the question of variation in voltage proposed by Professor Gebhardt, the author stated that the meters were generally installed in plants where voltage regulation was normally excellent and that they were connected with the main switchboard and not to a line where variations would be likely to occur. Further, in transforming the current, two transformers were used.

With reference to the error introduced by pulsating flow, which in some instruments might be as high as 60 per cent, the author stated that the only way to overcome the difficulty was to calibrate the meter for the actual conditions.

With reference to the device on the Sargent meter which compensated for pressure, mentioned by Mr. Anderson, the author said that he was attempting the design of such a device which would be perfected upon the completion of some research work. He thought that to buy apparatus for testing a flow meter was to spend more money than would be warranted.

The author did not agree with Mr. Dodge that a mechanical device could be as accurate as a watt-hour meter.

No. 1698

CRUDE-OIL MOTORS VS. STEAM ENGINES IN MARINE PRACTICE

By J. W. Morton, Philadelphia, Pa.
Junior Member of the Society

This paper is a discussion of the various factors to be considered in choosing the form of motive power for war vessels and cargo ships; also a presentation of the advantages and disadvantages as compared to steam engines of high- and low-powered oil motors of both the constant-pressure and constant-volume type. The relative values of four-stroke cycle, two-stroke cycle, double-acting, and horizontal, oil motors are also compared and some details of their construction such as lubrication systems, piston cooling and scavenging pumps are analyzed.

THE number of articles which have appeared of late describing the performances of the so-called motorships would seem to indicate that for marine purposes the crude-oil motor is rapidly replacing the steam engine. The chief reason for this is not to be found, as might be expected, in the lower operating cost of the crude-oil motor, but is rather due to other factors which, when considered, lead to the conclusion that the crude-oil motor is probably the most economical prime mover of today.

2 Perhaps the correctness of this statement can best be shown by comparing a steamship with a motorship. In the case of a war vessel, for instance, there are eight factors which ought to be considered in choosing the form of motive power, namely weight, space occupied, radius of operation, preparedness, crew necessary, fuel, auxiliary equipment, and cost.

3 The weight of the power plant is of great importance, for a saving therein can be utilized to increase the armor of the ship. If, for example, a warship of 3500 tons displacement be equipped with crude-oil motors there will result a saving as indicated by Table 1, which is based on a single steam engine developing 4400 hp. and 3 units, of 1600 hp. each, of a standard make of crude-oil motor.

Presented at the Spring Meeting, Detroit, June 1919, of The American Society of Mechanical Engineers.

4 In the case cited about the same space would be required for both motor plant and steam plant, but the total horsepower could just as well be developed by two units of slightly larger size, and if this were done approximately one-third the space would be saved. Moreover, if oil motors were used the space occupied by the coal bunkers could be partially released, as crude oil can be stored in tanks under the engine floor and below the protective line.

5 It is obvious that a warship must possess a great cruising radius and if weight for weight of engines be considered then the

TABLE 1 COMPARATIVE WEIGHTS OF POWER PLANTS IN STEAMSHIPS AND MOTORSHIPS

	Steamship		Motorship		Savings in Motorship	
					weight, tons	Per cent
Engine, tons	335		264		71	...
Per cent of displacement		9.6		7.5	...	2.1
Fuel, tons	246		86		160	...
Per cent of displacement		7.0		2.5	...	4.5
Crew, provisions, water, tons	91		82		9	...
Per cent of displacement		2.6		2.3	...	0.3
Totals	672		432		240	6.9

ratio R of the cruising radii of a steamship and a motorship may be expressed by the formula

$$R = \frac{V_c \times H_o \times W_c}{V_o \times H_c \times W_o}$$

where

V_c = cu ft. occupied per ton of coal = 45
V_o = cu. ft. occupied per ton of oil = 35
H_c = B.t.u. per lb. of coal = 14,000
H_o = B.t.u. per lb. of oil = 18,500
W_c = lb. coal consumed per e.hp-hr.
W_o = lb. fuel oil consumed per e.hp-hr.

In the case of the warship under consideration if W_c be taken as 1.75 and W_o as 0.4, then the ratio will be

$$R = \frac{45 \times 18500 \times 1.75}{35 \times 14000 \times 0.4} = 5.7$$

The cruising radius will, of course, vary with the type of engine and ship, but nevertheless the average ratio, as expressed above, may be taken at least as 1 to 5 in favor of the crude-oil motors.

6 A motorship in sharp contrast to the steamship is always ready for action, as no time is lost in getting up steam. Moreover, fuel is consumed only when the motors are running, and the motorship is capable of maintaining full speed as long as there is a supply of fuel.

7 As there is no need for stokers on a motorship, the crew can be decreased about 10 per cent, and this of course permits of a corresponding saving in provision, water storage and quarters.

8 In the case of a steam-driven warship the smoke from the stack frequently betrays the location of the ship, for it is well known that a vessel can be easily located by its smoke long before the masts or hulls are in sight. Another drawback to the steam-driven warship lies in the fact that the smoke covers the ship with a film of soot which, getting into vital parts of auxiliary machinery, necessitates frequent cleaning and unnecessary wear and tear. On the other hand, when oil is used there is no smoke, the handling of the oil is both cleaner and easier, and the life of machinery and equipment is greatly increased.

9 When a change is made from steam to oil the auxiliary machinery is usually electrically driven, as this has been found to be most satisfactory. The dynamos are usually driven by separate oil motors. Hot water for heating and sanitary purposes is obtained from a small oil-fired boiler and while the first costs of such installations are high, they are in time offset by the lower operating costs.

10 Prior to the war the first cost of a crude-oil motor plant was approximately twice that of a steam plant. This was partly offset, however, by the savings in operating expenses, under which item comes cost of fuel, labor and supplies. The cost of fuel naturally depends upon the prevailing market for coal and oil. In regard to maintenance there is practically no information available as the crude-oil motor is still in its infancy so far as this country is concerned.

CARGO SHIPS

11 Many of the preceding statements concerning the installation of crude-oil motors on battleships can also be properly applied to cargo ships. During the last four years a number of such vessels have been equipped with crude-oil motors of both the two-stroke and four-stroke cycle type, and while engineers are not agreed as to the best type, the four-stroke cycle is to be preferred.

12 One of the most successful installations of this kind is that

TABLE 2 DATA ON VARIOUS CARGO MOTORSHIPS

(Number of main motors on each ship—2)

Name of Ship	Year placed in service	Owner	Hull					Machinery							
			Length ft.-in.	Beam, ft.-in.	Depth, ft.-in.	Capacity, deadweight tons	Speed in knots	Total i.hp.	No of cylinders in each motor	Bore, mm.	Stroke, mm.	R. p. m.	Ratio of Stroke to bore	Piston speed, meters per min.	M. e. p., kg. per sq. cm.
Suecia	1912	Nordstjernan	362-0	51-3	25-6	6550	10¾	2000	8	500	660	140	1.32	185	6.2
Selandia*	1912	East Asiatic Co.	370-0	53-0	30-0	7400	12	2500	8	530	730	140	1.38	204	6.2
Pedro Christophersen	1913	Nordstjernan	362-0	51-3	25-6	6550	10¾	2000	8	500	660	140	1.32	185	6.2
Siam	1913	East Asiatic Co.	410-0	55-0	30-6	9700	11¼	3000	8	590	800	125	1.36	200	6.2
Annam	1913	East Asiatic Co.	410-0	55-0	30-6	9700	11¼	3000	8	590	800	125	1.36	200	6.2
Kronprins Gust. Adolf	1914	Nordstjernan	362-0	51-3	25-6	6550	10¾	2000	6	540	730	140	1.35	204	6.4
California	1913	United Steamship Co.	405-0	54-0	35-0	7250	11¼	2600	8	540	730	140	1.35	204	6.2
Kronsan Margareta	1914	Nordstjernan	362-0	51-3	25-6	6550	10¾	2000	6	540	730	140	1.35	204	6.4
Fionia*	1914	East Asiatic Co.	395-0	53-0	30-0	6700	13½	4000	6	740	1100	100	1.49	220	6.34
Malakka	1914	East Asiatic Co.	410-0	55-0	30-6	9200	11¼	3100	6	630	960	125	1.525	240	6.2
Tongking	1914	East Asiatic Co.	410-0	55-0	30-6	9200	11¼	3100	6	630	960	125	1.525	240	6.2
Pacific	1914	Nordstjernan	362-0	51-3	25-6	6550	10¾	2000	6	540	730	140	1.35	204	6.4
San Francisco	1915	Nordstjernan	362-0	51-3	25-6	6550	10¾	2000	6	540	730	140	1.35	204	6.4
Panama	1915	East Asiatic Co.	410-0	55-0	30-6	9200	11¼	3100	6	630	960	125	1.525	240	6.2
Australien	1915	East Asiatic Co.	410-0	55-0	30-6	9200	11¼	3100	6	630	960	125	1.525	240	6.2
Columbia	1915	East Asiatic Co.	425-0	55-0	30-6	9500	11.15	3100	6	630	960	125	1.525	240	6.2
Chile	1915	East Asiatic Co.	425-0	55-0	30-6	9500	11.15	3100	6	630	960	125	1.525	240	6.2
Oregon	1915	United Steamship Co.	405-0	54-0	36-6	8270	11.0	2800	6	590	900	140	1.525	252	6.1
Peru	1916	East Asiatic Co.	425-0	55-0	30-6	9500	11.15	3100	6	630	960	125	1.525	240	6.2
														(Avg.)	6.24

*Cargo and passenger vessel.

the motorship, and transmitting power through a single slow-running screw as in the case of the steamship. The propeller on the steamship has a diameter of 17 ft., while those of the motorship are only 10 ft. The ratio of indicated to effective horsepower is therefore the same for both ships and the total efficiency of engines and propellers is also about the same.

13 In order to correspond to the motorship in horsepower the steamship would be obliged to carry an extra boiler. This would increase its weight 570 tons, whereas the weight of the machinery on the motorship is only 470 tons. The length of the engine room of the motorship is 41 ft. while the length of engine room and necessary boilers in the steamship is 66 ft., and this despite the fact that the steamship has less horsepower.

14 The fuel consumption during the trial trip of the *Suecia* was 0.2948 lb. per i. hp., or 0.3685 lb. taking the mechanical efficiency at 80 per cent, a very satisfactory value. Table 2 affords a comparison of this vessel with other motorships.

MOTOR SIZES

15 The horsepower of a crude-oil motor, whether of the constant-volume or constant-pressure type, is somewhat limited by practical considerations of construction. At the present time the maximum size is 2500 hp. per shaft and from six to eight cylinders for the two- and four-stroke cycle constant-pressure type, although some experimental engines have been built as large as 6000 hp. This is in sharp contrast, however, to the power developed by steam which runs as high as 15,000 hp. for the steam engine and 35,000 hp. for the steam turbine.

16 Crude-oil motors of small horsepower are usually of the high-pressure type, medium, so-called semi-Diesel or hot-bulb type and two-cycle constant-volume type. Prominent among such semi-Diesel engines are the Bolinder, Scandia, Avance, Alliance, Tuxham, Gidion and Holm.

17 It has been proved both by experiment and practice that the constant-volume, high-power, single-cylinder type of motor is unsatisfactory, and chiefly because of the fact that the enormous heat generated at the pistons during the explosion must travel a great distance before it is transferred to the water jacket, and as a result the central part of the piston is heated to such an extent that preignition frequently occurs. Furthermore, the sudden rise

TABLE 2 DATA ON VARIOUS CARGO MOTORSHIPS

(Number of main motors on each ship—2)

Name of Ship	Year placed in service	Owner	Hull: Length ft.-in.	Hull: Beam, ft.-in.	Hull: Depth, ft.-in.	Hull: Capacity, deadweight tons	Hull: Speed in knots	Machinery: Total i.hp.	Machinery: No of cylinders in each motor	Machinery: Bore, mm.	Machinery: Stroke, mm.	Machinery: R. p. m.	Machinery: Ratio of Stroke to bore	Machinery: Piston speed, meters per min.	Machinery: M. e. p., kg. per sq. cm.
Suecia	1912	Nordstjernan	362 0	51 3	25 6	6550	10¾	2000	8	500	660	140	1.32	185	6.2
Selandia	1912	East Asiatic Co.	370 0	53-0	30 0	7400	12	2500	8	530	730	140	1.38	204	6.2
Pedro Christophersen	1913	Nordstjernan	362 0	51 3	25 6	6550	10¾	2000	8	500	660	140	1.32	185	6.2
Siam	1913	East Asiatic Co.	410 0	55 0	30 6	9700	11¼	3000	8	590	800	125	1.36	200	6.2
Annam	1913	East Asiatic Co.	410 0	55 0	30 6	9700	11¼	3000	8	590	800	125	1.36	200	6.2
Kronprins Gust. Adolf.	1914	Nordstjernan	362-0	51-3	25 6	6550	10¾	2000	6	540	730	140	1.35	204	6.4
California	1913	United Steamship Co.	405-0	54-0	35 0	7250	11¼	2600	8	540	730	140	1.35	204	6.2
Kronsan Margareta	1914	Nordstjernan	362 0	51 3	25 6	6550	10¾	2000	6	540	730	140	1.35	204	6.4
Fionia*	1914	East Asiatic Co.	395 0	53 0	30 0	6700	13½	4000	6	740	1100	100	1.49	220	6.34
Malakka	1914	East Asiatic Co.	410 0	55-0	30 6	9200	11¼	3100	6	630	960	125	1.525	240	6.2
Tonking	1914	East Asiatic Co.	410 0	55-0	30 6	9200	11¼	3100	6	630	960	125	1.525	240	6.2
Pacific	1914	Nordstjernan	362-0	51-3	25 6	6550	10¾	2000	6	540	730	140	1.35	204	6.4
San Francisco	1915	Nordstjernan	362-0	51-3	25 6	6550	10¾	2000	6	540	730	140	1.35	204	6.4
Panama	1915	East Asiatic Co.	410-0	55-0	30-6	9200	11¼	3100	6	630	960	125	1.525	240	6.2
Australien	1915	East Asiatic Co.	410-0	55-0	30-6	9200	11¼	3100	6	630	960	125	1.525	240	6.2
Columbia	1915	East Asiatic Co.	425-0	55-0	30-6	9500	11.15	3100	6	630	960	125	1.525	240	6.2
Chile	1915	East Asiatic Co.	425-0	55-0	30-6	9500	11.15	3100	6	630	960	125	1.525	240	6.2
Oregon	1915	United Steamship Co.	405-0	54-0	36-6	8270	11.0	2800	6	590	900	140	1.525	252	6.1
Peru	1916	East Asiatic Co.	425-0	55-0	30-6	9500	11.15	3100	6	630	960	125	1.525	240	6.2
														(Avg.)	6.24

*Cargo and passenger vessel.

in pressure sets up detrimental vibrating stresses in the material, and it is also exceedingly difficult to obtain a proper mixture of air and oil, rapid ignition and complete combustion and exhaust.

18 On the other hand, the constant-pressure oil motor of high power must have numerous cylinders all of the same size and the high pressure, which lasts approximately 10 per cent of the stroke, necessitates a careful design of the crank mechanism. This applies especially to the crankshaft, which is both difficult and costly to produce. Moreover, in motors with only a few cylinders the flywheel must be heavy, large and expensive if a great variation in speed is to be cared for. Difficulties have also been encountered in keeping the piston leak-proof — if it fits too tightly the result is a large frictional loss, and if the clearance is too great there will be a loss of pressure. The greater the piston area the more effective must be the cooling system as the surface exposed per pound of working medium decreases as the cylinder dimensions increase. This accounts for the lower thermal efficiency and the horsepower available per unit of piston surface with large cylinders.

LOW-POWER MOTORS

19 *Advantages.* The efficiency of an oil motor is greater than that of a steam engine, especially in small units, and furthermore it decreases less with increasing load. The fuel consumption of an oil motor is independent of the attendant's skill, as it shows the same value in daily work as on a test stand. With a steam engine, however, the fuel consumption is dependent upon the attendant and especially the stokers, who must be experienced. In steam plants the boilers and pipes must be able to withstand high pressures and temperatures, but with oil motors a high pressure exists only in the cylinder and a few small pipes. Moreover, the oil motor is simpler in construction, and the weight and space occupied considerably less. It is of interest to note in this connection that the steam turbines in torpedo boats weigh about 45 lb. per b.hp. and aeroplane motors but a little over 1 lb. per b.hp.

20 The small oil motor has other advantages over the steam engine and these can be briefly enumerated as follows: Starting takes only a few seconds; fuel is consumed only during operation; combustion in the cylinders can take place uninterruptedly for a long period of time; the engines require little attention and the oil tanks can be replenished more easily than coal bunkers can be filled and without the dirt accompanying the latter procedure.

21 *Disadvantages.* It must not be thought that oil motors have no disadvantages, for there are some, chief among which may be mentioned the following: Fuel oil is more expensive than coal and is not available in all ports; furthermore, lubricating-oil consumption is greater than in the case of steam engines as one must deal with higher temperatures and, as a rule, higher pressures. The operation is also more irregular than that of a steam engine, as oil motors cannot be run as slowly as is desirable when reversing from ahead to astern. The causes of interruptions (especially with electrical ignition) are often difficult to locate; the motor frequently works noisily with smoking and malodorous exhaust; the dangers due to fire and explosion are greater, and the life of the machine is less.

HIGH-POWER MOTORS

22 *Advantages.* The statements made in regard to the low-power motor are also true to a certain extent in the case of the high-power motor, but in addition the following items should be noted: With larger output the economy of a power plant becomes of greater importance. The low efficiency of most steam plants is due to the fact that the heat generated must travel through heavy boiler plates, and that there is a great loss of heat through the exhaust, and through condensation of steam. Furthermore, the steam plant operates with a small temperature difference, and to increase this difference is exceedingly difficult because it is very desirable to work with higher temperature limits. With saturated steam the pressure soon increases beyond a practical value and with superheated steam the walls of the superheater soon reach their limit, as they should never be allowed to become red hot. On the other hand, in the oil motor the heat formation and utilization take place in the working cylinders, and the higher temperature limit is therefore the same as the temperature of combustion. The value of the lower temperature limit is greater than in a case of a steam plant, but the temperature difference of the process is higher.

23 Another advantage of the oil motor is due to the ease with which the air supply necessary for complete combustion is regulated and since the point of ignition is practically constant, there results complete combustion. On account of the great heat which exists in a boiler room it is very difficult for the stokers to maintain normal pressure in a steam boiler; the coal consumption is accordingly increased by reason of careless firing but the speed of the vessel decreases. On the other hand, in the case of the motorship the

reverse holds true. If the fuel oil becomes warmer it is more easily vaporized, with the result that fuel consumption decreases while the speed of the vessel increases.

24 Since fuel oil can be stored in tanks a considerable saving can be made in space, more oil can be carried and a greater cruising radius thus afforded. The weight of a slow-running four-cycle stroke oil motor for cargo ships is about the same as that of a steam plant but far less space is required. The horsepower and speed are about the same for a twin-screw motorship as for a single-screw steamship, but as the former's screw diameters are smaller their efficiency is greater and the vessel can thus be more easily maneuvered. The cylinders of a motorship are all similar, and since they work independent of each other, one or more of them can be put out of service and the motor still continue to operate.

25 Steam boilers and their fittings require as a rule a considerable outlay for repairs and maintenance; also the sides and bottom plates of the hull which are nearest to the coal bankers are likely to corrode because of the sulphur content of the coal, and in addition sea water in the ballast tanks may also cause corrosion of the plates. On the other hand, a motorship has no boilers and the fuel is stored in bottom tanks, and this very fact is a great advantage since the tanks are protected by the oil.

26 *Disadvantages.* In the case of a single-cylinder oil motor the torque acting on a single shaft is limited, but everything else being equal, such as shape and line of hulls, propellers, r.p.m. and b.hp., the i.hp. of the motorship will be greater than that of the steamship. Since the reciprocating parts in a crude-oil motor weigh more than those of a steam engine the balancing of the masses is more difficult and the shifting forces greater. At light loads the pistons in a large motor are not tight enough to maintain high compression, and since the combustion space is not sufficiently warm for positive self-ignition, large motors can only be run when under light loads at approximately one-third to one-fourth normal speed. Furthermore, the heat transmitted to the cooling water per unit area of the cylinder is about two and one-half times as great in the combustion chamber as in the cylinder walls, and accordingly the heat drop in the cylinder wall is very considerable and tends to introduce stresses in the material. If the limits of elasticity are exceeded these stresses will be transmitted to distant parts. The material is also subjected to high stresses as a result of rapidly repeated power impulses, high temperatures and great pressure in the cylinder.

CONSTANT-PRESSURE VS. CONSTANT-VOLUME MOTORS

27 *Advantages.* The constant-pressure oil motor consumes less fuel than the constant-volume type, and this is due to the fact that it has a greater compression and a higher expansion ratio. Furthermore, the fuel oil is introduced into the combustion chamber in its natural condition and is, therefore, more completely vaporized, with the result that combustion is more complete. A cheaper grade of oil can, therefore, be used. The mean effective pressure is also greater and therefore greater output can be obtained and preignition practically excluded.

28 *Disadvantages.* The constant-pressure oil motor also has its disadvantages. In the first place this type of motor costs more to produce than the constant-volume type, and in addition an air compressor is necessary. The motor itself operates at high pressures and temperatures and the air compressor necessitates tanks for storing the air for starting and running purposes.

CONSTANT-VOLUME MOTORS

29 *Low-pressure Type.* The low-pressure oil motor is perhaps the simplest prime mover of its kind and is especially adopted for use in small boats where the operators are usually unskilled. The chief advantages of this type of motor are its low first cost and small operating expenses. Furthermore, the low temperature in the cylinder and the low explosive pressure, together with its simplicity of operation make it a motor of great durability. This type, however, has some disadvantages and chief among them are its small power output, large cylinder dimensions, excessive weight and the long time required for starting.

30 *Medium, or so-called Hot-Bulb Type.* The so-called hot-bulb motor operates at a medium pressure with or without water injection. While the tendency today is to avoid water injection, if the hot-bulb motor be nevertheless operated with water injection its fuel consumption will be comparatively low; the temperature of inlet air low; overheating of cylinder and piston will seldom occur; the cylinder will remain clean for a longer period, and the piston rings will not gum or stick. The water tank, however, adds weight, and considerably reduces cargo space. Furthermore despite careful attention, corrosion and rust will rapidly destroy the motor, especially if sea water is used. The most serious objection, how-

ever, lies in the fact that the amount of water injected must be regulated according to load, and this necessitates constant attention.

31 On the other hand, if the medium-pressure motor is operated without water injection, the cost of maintenance is low, the mechanism construction simple and little attention is necessary. Its chief operating difficulty lies in the inability to properly time the supply of fuel oil as its injection against the highly compressed air is exceedingly difficult in the short time allowed before ignition takes place.

32 *High-pressure Type.* The advantages of high-pressure motors with self-ignition and injection of fuel oil during the high-compression cycle are practically the same as those of the medium-pressure motor without water injection. High-pressure motors, however, are usually operated with an oil chamber and such a type is very economical, has a high mean effective pressure, its cylinders are small, its weight low, and it can be started without any preliminary operations. Furthermore, since the fuel oil is not injected against a high pressure the oil pumps can be of simple design and are consequently easily maintained. Like all other motors, however, this type has its disadvantages — chiefly the fault of design and construction rather than operation. If the size of the oil chambers or atomizer holes be improperly proportioned, fuel-oil consumption will greatly increase. Since this type of motor cannot be started by hand, pressure air is necessary and therefore, the motor must be manufactured with great care. This, of course, means a higher initial cost. In operation, the ignition and combustion cannot always be controlled and if inefficiently operated, preignition will occur.

FOUR-STROKE-CYCLE MOTORS

33 *Advantages.* This type of motor is particularly well adopted to high-speed work since it has a separate suction and exhaust stroke and thus the cylinder is filled each time with a full and new charge. Compared to a corresponding two-stroke motor it uses less fuel per b.hp. Furthermore, as it does not employ a scavenging pump, receivers, etc., the motor construction is simplified and better adapted for continuous hard work than the two-stroke-cycle motor.

34 *Disadvantages.* The chief objections to this type of motor are its increased weight and the greater space required. On account of the larger number of valves the valve gear is more complicated

and noisy. Except with very large outputs the piston can act as a crosshead, and unless the cylinder diameter be greater than 20 in., water cooling is unnecessary. The crank motion of this type of motor necessitates a heavy crankshaft and heavy flywheel if smooth running is to be obtained. Furthermore, the exhaust valves are subjected to high temperatures and occasionally give rise to considerable trouble. This type of motor is exclusively used by the Danish and Dutch and now the American merchant marine and also finds a large field in the automobile and aeroplane industries, as gasoline motors work, as a rule, on the four-stroke cycle.

TWO-STROKE-CYCLE MOTORS

35 *Advantages.* Compared with a four-stroke motor the two-stroke-cycle type has a greater output per cylinder, and everything else being equal, this excess will usually be about 75 per cent. On account of its greater pressure at the beginning of the compression stroke, the weight of the charge can be increased if the stroke volumes are equal. On the other hand, power is exerted only during three-quarters of the stroke, as during the remaining portion exhaust takes place. Thus the total stroke volume is not completely filled at the beginning of compression with a fresh charge, and the result is that for the same cylinder dimensions the output is only 75 per cent greater than for a four-stroke-cycle motor. This difference, however, becomes even less for high-speed two-stroke motors since the mean effective pressure is lower. By forced scavenging through the valves the fresh-air charge can, however, be made cleaner. In large units only the starting and fuel valves are subjected to high temperatures. In the open-type four-stroke motor, vapor from lubricating oil and exhaust gases escape into the engine room, thereby causing impure air conditions. This, however, does not occur in a two-stroke motor which has a closed crankcase. Finally, the two-stroke engine is very smooth-running because of its more uniform torque.

36 *Disadvantages.* The chief disadvantage of the two-stroke motor is its high operating cost. This is due to its higher loss of fresh-air charge through exhaust ports, smaller utilization of the working fluid, and higher heat loss due to cooling, friction and increased pump work. The mean effective pressure is lower because during the working stroke exhaust also takes place. The mean temperature during the cycle is higher than in the four-stroke motor and conse-

quently cooling of the piston is necessary. Furthermore, the piston can act as a crosshead only in small motors; lubricating-oil consumption is greater, and the exhaust ports become overheated if not water-cooled. Finally, the stresses exerted on the moving parts are very large and, therefore, there must be ample sliding and bearing surfaces. The two-stroke type of motor is chiefly used in vessels where weight, space and first cost are the deciding factors. Its reliability and economy, however, have not been so marked as in the case of the four-stroke motor, although in the Scandinavian countries, Germany and Italy it has been used to a considerable extent.

DOUBLE-ACTING MOTORS

37 Stationary double-acting motors have been used with fairly good results but double-acting marine motors are as yet in an early stage of development. The chief advantages of the double-acting motor are its light construction, better balancing, its economical operation and the comparatively small space which it occupies. However, the first cost is higher, and the lubricating-oil consumption is greater than in the case of a single-acting motor. This type is also more complicated and requires cooling of pistons and piston rods as the latter usually pass through the hot combustion chamber.

HORIZONTAL MOTORS

38 The horizontal motor is chiefly used for stationary plants because this type is more accessible, can be greatly overloaded, is cheap to construct and can be placed in a space with low headroom. Like all other types of internal-combustion engines, the horizontal motor has many disadvantages, chief among which are the following: The piston rings have a tendency to stick due to lubricating oil collecting on the lower half of the cylinder; imperfect lubrication thus results and this causes leakage and increases fuel consumption. The motor also has a tendency to rock and this necessitates a large foundation. Furthermore, the location of the crankshaft makes it exceedingly difficult to secure a direct drive of electric generators. These disadvantages in stationary plants usually outweigh all other considerations, and as a result the horizontal, double-acting, constant-pressure motor in large units is the type usually installed. On the other hand, in spite of the higher initial cost, the vertical motor is always used on ships as its operation, durability and great overload capacity make it an ideal installation.

MOTOR DETAILS

39 *Mechanisms vs. Direct-acting Trunk.* Trunk mechanism is applied in small-size motors and the crosshead guide is used only in large motors. This is due to the fact that motors with trunk pistons are cheaper to produce; the overall height is reduced and it meets the requirements for a non-leakable piston. The trunk mechanism, however, has certain disadvantages and these may be enumerated as follows: Side pressure is taken up by the hot part of the cylinder and since one side of the piston may become hotter than the other, the piston will naturally bend toward that side. The cylinder will, therefore, have a tendency to wear oval in shape, causing the piston to blow. Furthermore, the piston will, notwithstanding guards, draw lubricating oil up into the combustion chamber, and this of course increases operating expense. On the other hand, when the crosshead is used the lubricating oil can circulate in great quantities and inspection is also simplified.

40 *Forced-Feed vs. Gravity System Lubrication.* The maximum pressures exerted between the wearing surfaces in an oil motor are greater than those in a steam engine, and consequently lubricating systems must be different for each type. In large oil motors high pressures are used to force the oil to all vital parts, and the wearing surfaces are thus separated by a film of oil. The loss due to friction is, therefore, reduced and the method also assists in keeping the surfaces cool and at the same time increasing their life. Forced-feed lubrication also requires less attention and is independent of the engineer's skill as long as the run is not hindered. The oil consumption is economical as the oil can be filtered and used again. There is no splashing of oil in or about the engine room and for this reason the system is well adapted to the high-speed motors. The forced-feed lubricating system with its double oil pumps, etc., is, however, quite expensive and since the mechanism is not visible, a breakdown is not easily detected; consequently, if one occurs, a serious delay will be occasioned. The forced-feed lubrication system with a close crankcase is used to great extent in high-powered motors whereas the centrifugal or gravity system is used in small motors.

41 *Piston Cooling.* Piston cooling may be accomplished by the use of either oil or water. If oil is used it may be combined with the forced-feed lubrication system, and in such case should the cooling oil leak into the bottom of the crankcase, no harm will be

done. On the other hand, if water is used and it gets into the crankcase the oil will be expelled from the sliding surface and serious damage may result. If sea water is used as a cooling medium, storage tanks are unnecessary, but if oil or fresh water is used, tanks and coolers must be provided and for large motors, these require considerable space. As a result large units use sea water for cooling purposes and medium-size motors employ oil.

42 Scavenging Pumps. Piston Type. The simplest and cheapset type of scavenging pump is a piston working in a closed crank pit. As a rule, however, this type of pump supplies insufficient air since the quantity of air is limited to the stroke volume. The cylinder is also insufficiently cooled and the volumetric efficiency decreases.

43 Stepped-Piston Type. Scavenging air obtained from a pump with a stepped piston is usually sufficient to meet all purposes, but if the piston is not tight, exhaust gases may mix with the fresh air, in which case there is great danger from explosion of the lubricating-oil vapors in the receiver. The stepped piston increases neither the length nor width of the motor, and if breakage occurs on a scavenging pump, only that individual cylinder is affected. The moving parts of such a pump are heavy and consequently large masses have to be accelerated and retarded. The motor, however, is increased in height, even where the wristpin is secured to the working cylinder, because the ratio of L (length of connecting rod) to R (crank radius) must be greater than normal in order to insure clearance between piston and shaft or bearing cap. This ratio, however, becomes less if the wristpin is secured in the pump piston, but the overall height of the motor will be greater. On the other hand, if the piston pump takes the place of a crosshead, the length of the wristpin can be increased and the bearing pressure thus reduced.

44 Special Pumps. The first cost of special scavenging pumps is high; furthermore, they require more room and, should they fail to function the several working cylinders will be greatly affected. The chief advantage of special pumps is their lower oil consumption and the ease with which sufficient cold air can be obtained.

45 Each of the types of pumps mentioned has its various uses. The piston type is usually found on small motors, the stepped-piston type on submarine engines and motors of medium size, and separate or special pumps on heavy-duty motors installed on cargo and passenger ships.

46 *Ignition Devices of Constant-Volume Engines.* Although it has been stated that the constant-volume crude-oil motor is an un-

satisfactory type, its ignition devices are of interest. If the "electric type of ignition" is employed the motor can be started immediately. The time of ignition can be controlled and varied at will. Electric ignition, however, is expensive and it is difficult for unskilled mechanics to keep it in order.

47 Another type of ignition device is the so-called "ignition tube." With this tube a greater compression ratio can be used and fuel consumption is also decreased. With this device the time of ignition can also be controlled and varied at will, but a steady or continuously burning torch is necessary and this is a great objection, especially at sea.

48 A third type of ignition system is the well-known "head" or "hot bulb," which is simple in construction, action and operation. At normal speeds this type of ignition is the most satisfactory and, furthermore, it is not affected by damp air or water spray. The chief objections, however, lie in the fact that it requires a long time to start the motor; the heads are easily cracked; the compression ratio is small and the mean effective pressure is low. Furthermore, if the load changes during operation the hot bulbs will often run cold in the four-stroke cycle and hot in the two-stroke cycle. Broadly speaking, however, this type of ignition has been found to be most satisfactory and is very largely used for all types of small motors using either kerosene or heavy oil as fuel.

DISCUSSION

Louis Illmer (written). In the paper under discussion, Mr. Morton apparently favors the four-stroke oil engine and appears to give insufficient credit to the recent advances made in heavy-duty two-stroke engine design as applied to the merchant-marine service.

Undue complication of mechanism is probably chiefly responsible for the slowness on the part of the American shipbuilder in adopting the oil engine for the propulsion of large cargo ships, since each additional vital moving part adds to the first cost, the liability to accident and the overhauling required for maintenance of operating efficiency.

Other things being equal, the oil engine with the least number of vital working parts will most adequately meet the demand for reliable low-cost power, particularly so under American conditions of high labor and relatively low fuel costs.

It is conceded that a perfected oil engine of the two-stroke type admits of the simplest possible construction for marine propulsion. The four-stroke marine engine suffers from inherent difficulty in reversing the mechanically operated inlet and exhaust valves as driven from the half-speed camshaft drive. Furthermore, the long-stroke slow-speed engine of the four-stroke type does not compare favorably in weight, floor space or in first cost with the cargo-ship type of two-stroke engine.

With but a single power impulse per cylinder for every two revolutions, the heavy power parts of a four-stroke engine do not work to advantage during the idle inlet- and exhaust-stroke periods. Hence the two-stroke engine is enabled to show a decided weight reduction and consequent lowering in first cost. For example, a six-cylinder slow-speed two-stroke engine suitable for cargo-ship propulsion can readily be made to deliver approximately $1\frac{3}{4}$ times as much shaft power for the same weight and floor space as will a four-stroke oil engine of the same speed and bore. This advantage is of decided importance when larger merchant ships are to be engined, since the usefulness of such a prime mover is determined in a large measure by the ultimate power capacity to which a compact six- or eight-cylinder unit can be built.

Another advantage of the two-stroke engine is that it requires no mechanically operated inlet and exhaust valves. The cylinders of such oil engines are best charged from the under side of the power piston, through inlet ports overrun by the piston. If desired, the scavenging air supply may be supplemented by additional low-pressure pumps but this is not essential when liberally proportioned injection-air compressors are provided.

With this mode of charging a two-stroke cylinder, the timing events of the inlet and exhaust ports must be correctly proportioned and the shape of the piston deflector lug must be properly designed to produce effective scavenging of the power cylinder, otherwise the incoming fresh-air charge will blow out of the exhaust ports without displacing the burnt products of combustion. Deficiency in oxygen for combustion of the fuel oil has been responsible for the failure of a number of two-stroke engine designs.

The described mode of charging two-stroke power cylinders reduces the valve gear parts to a minimum and requires only one air starting valve and one fuel valve opening into each cylinder head. Since these valves run in unison with the piston movements, they may be eccentric-driven and reversed by a Stephenson link in the manner of a steam-engine gear. The elimination of the half-speed camshaft drive is a feature of the two-stroke engine that makes for a simple and compact reversing valve gear.

While the thermal efficiency of a well-designed two-stroke engine is not quite as high as that of a four-stroke engine, the difference is not large. Most of the loss of economy is chargeable to the increased pump work required to charge the power cylinder, but the relatively small gain of about 10 per cent in brake efficiency in favor of the four-stroke engine is not in itself sufficient to offset the other advantages which a perfected two-stroke engine affords.

A good two-stroke engine is somewhat more difficult to design than a four-stroke engine, partly because of the greater heat flow through the cylinder walls. For equal bore dimensions, the rate of heat flow in a two-stroke engine is approximately 1.6 times that in a four-stroke engine running at the same speed.

The limiting rotative speed at which an oil or gas engine may be safely run is largely determined by the temperature assumed by the cylinder-bore wall, and if this is not kept within prescribed limits, troubles from piston lubrication and cracking of cylinder parts will result.

The temperature head required to drive heat at a given rate of flow through the cylinder wall increases with the wall thickness. For equal shaft power the bore dimensions of a four-stroke oil engine are about $\frac{4}{3}$ those required for a two-stroke engine. Allowing for the thicker four-stroke cylinder wall and assuming equal rotative speeds, the rate of heat flow in the case of the two-stroke engine should not exceed $\frac{5}{4}$ that obtained in a four-stroke cylinder of equal power capacity.

In high-speed marine oil engines, which are usually worked up to their limiting rate of heat flow, the four-stroke cycle offers some advantage. On the other hand, the relatively slow speed demanded for cargo-ship engines is exceptionally favorable to the two-stroke type, since these engines operate under heat-flow conditions so moderate as to allow ample cooling of all vital parts even in the largest-sized cylinders. It is therefore readily possible to keep the bore-wall temperatures of such two-stroke engines well below the critical limits required for safe and reliable running. To further safeguard against fatigue and breakdown, it is advisable to provide for liner cylinders and such other constructive features common in high-powered oil-engine practice, so as to give the cylinder parts the requisite long life and complete immunity against cracking.

The greater frequency of impulse and the consequent more even turning effort of the two-stroke engine largely improve the speed control and reduce to a minimum the flywheel effect needed for a smooth-running marine engine.

Still another feature of the two-stroke cycle that gives promise for rapid future development is its special adaptability to the hot-bulb type of engine.

Judging from the recent trend of marine oil-engine developments, it now appears that a combination of the constant-pressure Diesel cycle with the hot-bulb constant-volume cycle is likely to result in a cycle which is especially suited to American oil-engine requirements. The lowering of the maximum Diesel working pressure to approximately that used in the automobile engine should increase the reliability of operation.

The constant-volume or explosive engine, when working with a compression pressure of about 250 lb. per sq. in., shows a thermal shaft efficiency but little inferior to that of the Diesel engine. By further reducing the compression to about 150 or 175 lb. per sq. in. the fuel economy of the explosive engine is but slightly sacrificed, while for a given size of crankshaft the weight and cost relations as taken upon the power-output basis show a considerable improvement as compared with the high-compression engine.

An engine operating with the combined type of cycle should be provided with an efficient timed spray valve for the fuel-oil injection, and in large engines this should be preferably of the air-injection type used for Diesel engines.

When working with such limited compression, self-ignition may best be obtained by holding a portion of the products of combustion from stroke to stroke within a water-jacketed vaporizer chamber, without, however, requiring the use of a hot plate. The trapped hot burnt gases may then be used to preheat the air that is pressed into the bulb chamber to a point where it is capable of promoting self-ignition of the injected fuel oil. This sets up a light explosion at constant volume, after which the remaining oil may be gradually fed into the power cylinder in the manner of a Diesel engine.

While the water-cooled bulb will not of itself start the engine from the cold, it does not require an external flame; instead it is only necessary to preheat the small amount of air enclosed within the vaporizer chamber, which may readily be done by means of an electric coil heater or spark plug. After the first few explosive charges are thus ignited, the required heat transfer takes place from stroke to stroke to make the engine self-igniting.

The jacketed vaporizer is especially applicable to oil engines of the single-acting type, the bulb chamber being preferably formed centrally in the cylinder head about the cylinder axis. The timed fuel valve can then be made to inject straight through the bulb chamber and directly against the hot piston head. This arrangement gives the nozzle considerable distance for proper spray formation before striking the piston top, after which the oil charge spreads out over the piston-head surface and intimately mixes with all the air throughout the combustion chamber.

Finally it is pointed out that this late development in true semi-Diesel engines is especially suited to the two-stroke-cycle engine. In the four-stroke engine considerable constructive difficulties are involved in placing the mechanically operated inlet and exhaust valves about the water-cooled vaporizer chamber. Furthermore, in a two-stroke engine the vaporizer chamber may be kept at a smaller size with respect to the piston displacement, due to the fact that the confined hot gases have less chance to cool off between power impulses.

These and other advantages previously cited would indicate that a two-stroke semi-Diesel engine along the lines discussed should in the near future find favor for the propulsion of merchant ships and be capable of fully establishing the inherent possibilities of a slow-speed oil engine for marine service.

W. D. Ennis (written). The value of this paper would be increased if the author would add particulars regarding the motors used. The mean effective pressures range from 87 to 91 lb. per sq. in., values often surpassed by stationary Diesel engines. Apparently the cylinders are the ordinary four-cycle single-acting form, but there seems to be no statement of the fact.

A chief argument in favor of the crude-oil motor appears from Table 2, where "repeat orders" in 1916 are quoted for lines which began with this construction in 1912. What has happened since 1916?

It is to be assumed that the eight factors listed in Par. 2 are not presented as a complete list. In fact, the author refers to others, admitting a shorter life for the crude-oil motor and referring to the absence of sufficient data on maintenance costs. He also discusses what is almost an overwhelming disadvantage for any but slow-speed cargo ships, the multiplicity of cylinders. The largest cylinder in Table 2 is below 30 in. bore and develops only 333 hp. The largest plant listed is 4000 hp.

It is doubtful whether steam can ever be eliminated in naval practice. Requirements for heating, humidifying and evaporators do not tend to decrease. It is even proposed now to use steam heat for submarines. It is difficult for internal-combustion engines to displace steam where they cannot displace it completely.

Table 1 is obviously based on coal fuel for steam. If oil fuel is used, the ratio *R* directly following the table becomes about 2.0 instead of 5.7.

Forrest E. Cardullo characterized the adoption of the Diesel and analogous cycles for a standard as an advance too far in the direction of theory. He did not believe that there had been successfully developed an engine capable of burning continuously all types of crude oil in a satisfactory manner. The engines become dirty and efficiency is lost. He said he knew of no reason why a satisfactory producer for gasifying oil could not be designed which would operate continuously at 90 per cent efficiency or even higher. With this producer it would be possible to develop an internal combustion engine of the automobile type which, with the same power, would have less internal friction than the Diesel engine. There would be certain practical disadvantages to such a combination, the principal ones being the extra space required and the additional labor for operation. He referred to the number of times a plant operating Diesel or semi-Diesel engines must be laid up in order to clean the engines. The practical weight of experience and of actual reliability and cheapness, he believed, was in favor of the producer-gas engine operating with gas generated from fuel oil.

Lewis H. Nash concurred in the idea of adopting the principle of gasifying oil in some type of gas producer. He knew of gas engines which had been in operation for 25 years with negligible cost for repairs. He considered that the mechanical difficulties with the Diesel engine far outweighed its theoretical advantages.

Henry B. Oatley, referring to Table 1 in which Mr. Morton presented the comparative weights of power plants in steamships and motorships, said that in the comparative installations aboard the U. S. S. *Maumee* and *Tioga*, steam and motor-driven ships, respectively, the Diesel-engine plant weighed decidedly more than the steam plant. The difficulties of getting high pressure and the limitations of superheaters of which the author spoke while discussing the heat elements of temperature range for the various types of engines existed, according to Mr. Oatley, only in the present motor designs, but he saw no difficulty in the further development of the types which have been successfully operated at pressures as high as 500 or 700 lb. per sq. in. Thus, with higher pressures and with higher degrees of steam temperature the boiler plant, in point of weight and size, would become markedly smaller, and the contrast in point of space occupied aboard the ship for the steam plant as compared to the motor plant would decrease, possibly to the point of being on a par. Mr. Oatley further criticized the argument advanced for preferring the motorship over the steam-driven ship by reason of the depreciation of the hull because of the sulphur content of coal and the amount of coal space required in the steam-driven ship, on the ground that rapid progress had been made in the utilization of oil as fuel for steam-driven ships, and that this practice would in all likelihood continue to increase.

O. C. Berry,[1] commenting on Mr. Cardullo's suggestion of gasifying oil in some type of gas producer, mentioned that in experiments conducted at Purdue University for the purpose of finding the best means of introducing into the mixture the heat required to vaporize the liquid fuel, it had been concluded that two elements are required to get the mixture in a burnable condition. In the first place, the temperature of the mixture must be high enough to maintain in the liquid fuel sufficient vaporizing pressure so that the fuel, once reduced to a vapor, will stay in a vaporized state and get into the cylinder as a vapor and not a liquid. The other was the time element. Because of the smallness of this time element, though the temperatures are high enough to maintain the required temperatures, still the mixtures retain their wet condition when they arrive in the cylinder. For this reason, observed Professor Berry, the hot spot had come into automobile practice as one of the solutions for the vaporization of kerosene, and he expressed his belief that it could be applied to good advantage to the use of fuels heavier than kerosene. A properly designed hot spot, he said, would eliminate the difficulty found in gas producers, where the temperature is often so high that it is not possible to keep a high enough compression without getting preignition.

[1] Professor of Mechanical Engineering, Purdue University.

Mr. Cardullo said in reply that in mentioning the gas producer he had intended to convey the idea of the actual combustion of oil in a producer so as to form a clean, burnable mixture in the engine.

L. H. Johnson spoke of his experience with the two-cycle semi-Deisel hot-bulb, hot-surface engine and stated that after investigating the various theories advanced to explain this type of engine he had finally formulated the following conclusion as to its operation: That the oil must be injected into the engine in the form of a spray or a mist, and if low compression is to be used, it must be sprayed against a hot surface and vaporized; then this oil gas mixes with the air compressed in the chamber and explodes by the heat of compression or the heat stored in the combustion chamber. Tests, he said, had convinced him that it is necessary for successful operation of the engine to remove the deposit of carbon or residue of material that is left in the cylinder. In the experiments to which he referred no trouble had been found in maintaining the temperature necessary for successful operation and the main problem had been the elimination of the deposits.

The Author wrote in reply to Mr. Illmer's discussion that he was, at present, in favor of the four-stroke cycle oil engine for marine purposes. Space restrictions in the original paper had prevented his comparing the various types in detail. He thought that the two-stroke cycle engine had not been perfected to a point which would warrant its use at the present time in marine installations. He doubted the advisability of charging the two-stroke engine from below the piston on account of the danger of adulterating the working charge. Scavenging with valves located in the head was superior, but the design introduced complications.

He believed that with solid-fuel injection, the four-stroke engine could be made with fewer working parts, and that larger output could be obtained with less weight. He felt that the trend of manufacturers was toward the Diesel engine and away from the hot-bulb engine, in the larger sizes.

Answering Mr. Ennis' request for more complete information regarding the motors mentioned in the paper, the author stated that they were of the four-stroke, single-acting type and were built by Burmeister and Wain, Copenhagen, Denmark. He realized that mean effective pressures greater than 91 lb. per sq. in. were obtainable in stationary practice, but this did not extend to marine practice. There were excellent reasons for this, such as the difference in cooling media, ventilation, speed, and foundation. It was due to lack of raw material throughout the war that there had been little activity in the building of marine motors by the Copenhagen concern. The building numbers of present orders ran from 276 to 344 with deliveries stipulated as far ahead as 1922. The East Asiatic Company of Denmark had 30 on order, the largest of which was to be 6500 i.hp. in 16 cylinders. It was quite true, as Mr. Ennis pointed out, that, compared with oil rather than coal, the value of R derived in Par. 5 of the paper would be between 2 and 2.5.

With regard to the reliability of the motorship, the author cited the record of the *George Washington* which made a non-stop run of 68 days without mishap or cleaning of any sort. Every valve taken out for examination was found to be without fault and replaced.

No. 1699

A SUGGESTED FORMULA FOR RATING KEROSENE ENGINES

By D. L. Arnold, Chicago, Ill.
Member of the Society

At the present time there is no standard method of rating kerosene oil engines, and consequently engines of the same displacement are given different ratings by each manufacturer. A standard formula would therefore be of value and in this paper the writer discusses the various formulæ now in use and suggests that a piston displacement of 13,000 cu. in. per minute be taken as one brake horsepower. The subject of a standard nameplate is also discussed and a suggested form is given.

AMONG the various companies manufacturing kerosene engines there seems to be no uniform standard of rating in use, and as a result engines of the same bore, stroke and speed — or in other words, of the same piston displacement per minute — are given different ratings by almost as many manufacturers. In the past little attention has been paid to this minor detail, as it seems to have been considered, but the purchaser of engines and tractors has had much experience with this inconsistency. For instance, one buys an engine of 4 hp. that will actually develop 6.5 hp.; another buys one also rated 4 hp. but it will only develop 5 hp., and it will in all probability have a different piston displacement per minute. Apparently this means nothing as far as the customer is concerned, for in each case he receives more horsepower than he believed he was purchasing. A very great difficulty does, however, arise when these engines are judged by the amount of maximum work which they will perform in a given length of time, not on brake tests but on users' equipment, and the result is that the purchaser of the smaller engine is greatly dissatisfied, despite the fact that his engine will develop a 25 per cent overload.

2 The foregoing is only a sample case of the inconsistency which

---

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

has resulted in many bills now being introduced, or about to be introduced, in state legislatures, specifying how an engine shall be rated. If these bills should all pass it might be that each state would require a different rating, and thus the manufacturers would be obliged to furnish a different nameplate in each state. It would therefore seem advisable for the manufacturers to get together and adopt a standard rating for kerosene engines; and this work would be greatly facilitated if the engineers of this Society and of the Society of Automotive Engineers would coöperate and adopt a standard rating for this type of power plant.

TABLE 1 FORMULÆ NOW IN USE FOR FOUR-STROKE-CYCLE ENGINES

D = diameter of piston, in. L = length of stroke, in.
n = number of cylinders N = number of revolutions per minute

No.	Authority	Formula
1	N.A.C.C.[1] or A.L.A.M.	$\frac{D^2n}{2.5}$ (at 1000 ft. per min.)
2	E. P. Roberts	$\frac{D^2LNn}{18{,}000}$
3	French Automobile Club	$0.45\ D^2n$ (at 985 ft. per min.)
4	Royal Automobile Club, British	$0.405\ D^2n$
5	Royal Automobile Club, Swedish	$\frac{D^2LNn}{15{,}240}$
6	Prof. H. L. Collanders	$0.565\ D(D-1)n$
7	T. Thornycrofts	$\frac{D^2L^{0.75}n}{2700}$
8	M. Faroux	$0.121\ D^{2.4}L^{0.6}$
9	M. Arnon	$0.1\ D^2n$
10	E. W. Roberts[2]	$\frac{D^2LNn}{X}$

[1] Vol. 1, p. 30, S.A.E. Standards Data Sheets.
[2] Gas Engine Handbook, p. 131. X is a factor varying with the fuel and cycle. For 4-cycle engines X equals 16,000 for natural gas and 14,000 for gasoline.

3 If standard rating is to be of service, it must give equal protection to the manufacturer and the customer, and any formula which is obtained must therefore be based upon practice covering a sufficient period of time. Before taking up the proposed formula, however, the writer wishes to review some of the formulæ that are in use today and which are listed on Table 1.

4 The A.L.A.M. formula[1] is based on the assumption that there is developed within the cylinder a mean effective pressure of 90 lb. per sq. in., that the engine operates with 75 per cent mechanical efficiency, and that the piston speed is 1000 ft. per min. With ungoverned engines this formula is perhaps as good as any, but for engines operating with governors it is obvious that if a piston speed of other than 1000 ft. per min. is maintained, it will be necessary to make a correction to cover this point, in which event the formula is unwieldly and inconvenient.

5 An empirical formula[2] which is considered as very conservative practice when compared with the various ratings now used by engine manufacturers is as follows: B.hp. = $0.00006042\ D^2LNn = V/13{,}000$. Before deriving this formula the writer compared the ratings of 115 engines manufactured by 61 companies and found them to vary from 8,256 to 19,985 cu. in. per b.hp. Forty-six engines were rated using 11,000 cu. in. or less as 1 b.hp.; 29 used

[1] Derivation of A.L.A.M. Formula:

Let, HP_b = brake horsepower
HP_i = indicated horsepower
HP_r = rated horsepower
A = area of piston, sq. in.
D = diameter of piston, in.
E = mechanical efficiency (assumed at 0.75)
L = length of stroke, in.
n = number of cylinders
P = mean effective pressure, lb. per sq. in. (assumed at 90 lb.)
V = piston displacement per minute, cu. in.
N = revolutions per minute
s = piston speed, ft. per min. (assumed at 1000)

$$HP_b = HP_i \times E = \frac{1}{4}\,\frac{PASNn}{33{,}000}\,E = \frac{90 \times 0.7854\ D^2 = 1000 \times n \times 0.75}{4 \times 33{,}000 \times 12}$$

$$= \frac{D^2n}{2.489},\ \text{or for practical purposes,}\ \frac{D^2N}{2.5}$$

Also, since $s = \dfrac{2LN}{12}$

$$HP_b = \frac{90 \times 0.7854\ D^2 \times 2LN \times n \times 0.75}{4 \times 33{,}000 \times 12} = \frac{D^2LNn}{14{,}939} = 0.00006693\ D^2LNn$$

and substituting V for $0.7854\ D^2LNn$

$$HP_b = \frac{90 \times V \times 2 \times 0.75}{12 \times 33{,}000 \times 4} = \frac{V}{11{,}733}$$

[2] Derivation of Empirical Formula:

$$HP_r\ \text{(recommended)} = \frac{0.7854\ D^2LNn}{13{,}000} = \frac{V}{13{,}000}$$

$$HP_r\ \text{(transformed)} = \frac{D^2LN}{16{,}550}n = 0.00006042\ D^2LNn$$

11,000 to 12,000 cu. in.; 12 used 12,000 to 13,000 cu. in.; 10 used 13,000 to 14,000 cu. in. and 18 used above 14,000 cu. in.

6 It will thus be seen that 87 engines or approximately 75 per cent were rated as using less than 13,000 cu. in. per b.hp. and while this value may be more conservative than some manufacturers might wish to use, it gives a standard basis of comparison and allows the engine to develop a fair percentage of overload above its rating, since any engine should be able to develop as a maximum

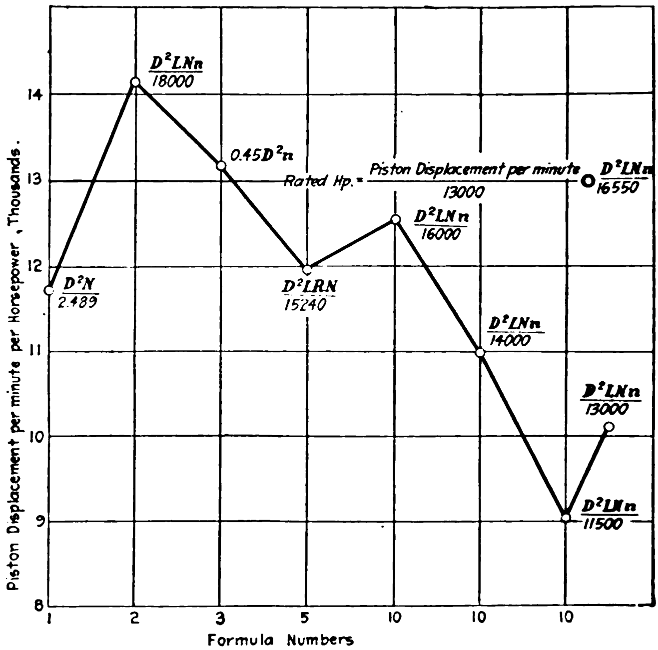


FIG. 1 COMPARISON OF ENGINE-RATING FORMULÆ IN TERMS OF PISTON DISPLACEMENT PER MINUTE

1 b.hp. per 11,000 cu. in. piston displacement per min. when the engine is in good condition.

7 At this time it seems desirable to compare this new formula with those already in use, and for that reason Table 2 has been compiled, from which have been plotted the values shown on Fig. 1 which shows a wide variation in some of the formulæ. It should also be noted that this chart does not show the maximum horsepower that can be expected from an engine but has

reference only to the rated brake horsepower which should be reasonably expected from a kerosene engine. If one engine can develop a higher maximum than another, or, in other words, can make good on a lower piston displacement per minute than its competitor, the credit is certainly due to the manufacturer who can consistently produce these conditions. However, practically all engines of correct design will develop a maximum horsepower according to the formula heretofore given, namely 1 b.hp. per 11,000 cu. in. piston displacement per min.

8 With reference to the rating of four-stroke-cycle engines, it would seem best for all concerned that this rating should be

TABLE 2 COMPARISON OF ENGINE-RATING FORMULÆ IN TERMS OF PISTON DISPLACEMENT (V) PER MINUTE

Formula No.	1	2	3	5	10			11
Authority	N.C.C.C. or A.L.A.M.	E. P. Roberts	French Automobile	Royal Automobile Club Swedish	E. W. Roberts			Rated Horsepower Recommendation
					16000	14000	11500	
Expressed in piston displacement per minute	$\frac{V}{11,733}$	$\frac{V}{14,137}$	$\frac{V}{13,180}$	$\frac{V}{11,969}$	$\frac{V}{12,566}$	$\frac{V}{10,995}$	$\frac{V}{9,032}$	$\frac{V}{13,000}$
Expressed in terms of Roberts formula	$\frac{D^2LNn}{14,939}$	$\frac{D^2LNn}{18,000}$	$\frac{D^2LNn}{16,781}$	$\frac{D^2LNn}{15,240}$	$\frac{D^2LNn}{15,999}$	$\frac{D^2LNn}{13,998}$	$\frac{D^2LNn}{11,499}$	$\frac{D^2NLn}{16,550}$

made the nearest whole horsepower to that determined by the standard formula, it being a very easy matter to change the rated speed of the engine in order to accomplish the exact rating. In any event, however, the rated horsepower would not, if the foregoing condition is adhered to, vary more than plus or minus 0.5 hp.

9 Having established a standard rating for this class of engine, there should also be a standardized nameplate; to state the hp. alone with no reference to speed is insufficient and misleading. This

nameplate should be clear and concise, leaving no cause for doubt as to what is meant. Many companies use the form of rating as follows:

HP 16 Speed 1000 RPM

Others simply use:

HP 16

and still others reverse the figures with relation to the lettering. It would seem, however, that the most exact way of placing the rating on the nameplate would be according to the following form:

16 HP at 500 RPM

It has been the custom of many manufacturers to leave out the little word "at," which, when included in the formula, leaves no room for doubt as to what is meant.

10 In considering the internal-combustion-engine rating we must also consider the tractor rating as the internal-combustion engine forms its power unit and the tractor rating is therefore dependent upon the engine rating. The owner of a tractor is not only interested in the amount of power that can be delivered by the engine but also in the amount that can be delivered to the drawbar at the different speeds. Here again, however, manufacturers have been inconsistent in the ratings which they have made. The majority of manufacturers have followed the rule that drawbar horsepower should be considered as 50 per cent of the rating of the power unit and in actual practice covering many years of experience this seems to be a very conservative figure to use. It is true that the tractor will often develop greater drawbar horsepower, but when taking into consideration the wide range of conditions through which the tractor must work, such as changes of soil and class of work, a 50 per cent rating for the drawbar pull seems on the whole to be the best value to use; therefore, on the tractor engines it would seem that the best nameplate would be the following:

16 Brake HP at 500 RPM
8 Drawbar HP at 500 RPM of the engine
Drawbar pull:
—— lb., reverse speed at —— miles per hour
—— lb., first speed at —— miles per hour
—— lb., second speed at —— miles per hour
—— lb., third speed at —— miles per hour
Drawbar pull and HP are on the average good footing.

11 The following formulæ are those which it would seem are best adopted to the rating of internal-combustion engines and tractors:

$$\text{Rated engine horsepower} = \frac{0.7854\, D^2LNn}{13{,}000} \quad \ldots\ldots\ldots\ldots \quad [1]$$

$$\text{Drawbar horsepower} = \frac{\text{Rated engine horsepower}}{2} \quad \ldots\ldots \quad [2]$$

$$\text{Drawbar pull, lb.}^{1} = \frac{375 \times \text{rated drawbar horsepower}}{\text{travel in miles per hour}} \quad [3]$$

$$\text{Motor torque,}^{2}\ \text{in-lb.} = \frac{63{,}025.21 \times \text{brake horsepower}}{N} \quad \ldots \quad [4]$$

12 Practical work in experimental testing laboratories and on regular test floors has proved that four-stroke-cycle engines operating on kerosene will develop for periods of two hours or more 1 b.hp.

[1] Derivation of Drawbar Pull Formula:
Let F_d = drawbar pull, lb.
HP_d = drawbar horsepower
S = speed in miles per hour

$$\text{Then } F_d = \frac{HP_d \times 33{,}000 \times 60}{5280 \times S} = \frac{375\, HP_d}{S}$$

[2] Derivation of Motor Torque Formula:
Let HP_b = brake horsepower
l = length of brake arm, in.
W = lb. pull at end of arm
N = revolutions per minute
T = motor torque, in-lb.

$$\text{Then } HP_b = \frac{2\pi\, lnW}{33{,}000}$$

$$T = \frac{HP_b \times 33{,}000 \times 12}{2\pi N}$$

$$= \frac{63{,}025.21 \times HP_b}{N}$$

per 11,000 cu. in. piston displacement per min. and a few engines have for short periods of time developed 1 b.hp. for every 9800 cu. in. piston displacement per min. Therefore, making allowances for general wear, mishandling and improper adjustment, the formulæ given in Par. 11 will be seen to be conservative and well adapted for the rating of this type of engine.

DISCUSSION

HARRY F. SMITH thought that it would be altogether out of place to attempt to create any arbitrary standard of cubic-inch displacement for the determination of the horsepower of the kerosene-oil engine. He said he knew of tests recently made on substantially identical kerosene-oil engines in which the b.hp. developed differed by 100 per cent, due to change in cylinder design. That meant that the thermal efficiency of one engine was 100 per cent better than the other. It seemed to him, therefore, that the manufacturer who was in a position to double the thermal efficiency of the kerosene oil engine ought to be entitled to whatever benefits might accrue thereby, and not be handicapped by the action of the Society or of State legislatures in rating his engine at a certain horsepower according to its size or the number of pounds of cast iron put into it.

STAFFORD MONTGOMERY inquired whether the rating of kerosene engine builders from 8000 to 20,000 cu. in. per b.hp. was limited to kerosene engines or whether it applied also to gasoline engines. This question was referred to the meeting by the chairman.

FREDRIK OTTESEN protested against manufacturers of large gas engines having to adopt the formula set forth in Mr. Arnold's paper.

WILLIAM T. MAGRUDER remarked that in The Ohio State University School of Military Aeronautics they had formulated a statement which was fairly accurate, that instead of using 13,000 or 14,000, 10,000 cu. in. represented quite closely the horsepower of an aeroplane engine up to 100 per cent capacity, and increased speed after that did not give correspondingly increased horsepower. He said that the horsepower of a steam tractor is unknown until tested for the reason that its overload capacity is not like that of a boiler, one or two or three hundred per cent, but from four to five hundred per cent; and that in tests performed at the laboratories of the university, tractors nominally rated at 20 hp. gave over 100 hp. and kept it up; on the other hand, in actual tests of kerosene and gasoline tractors on blocks, either in field work or in comparative b.hp. tests, many of the tractors fail to operate at their rated horsepower and not even their own experts could always get from them what the nameplates indicated they should deliver. Such being the conditions in practice, it appeared to him that by the standardization of horsepower, misunderstandings arising from the erroneous interpretation of the discrepancy between rated and actual horsepower could be avoided. He further observed that the question was bound to be taken up soon by the various legislatures and, as Mr. Arnold said in his paper, the conditions at present existing in the gas tractor and the kerosene tractor would be duplicated in the automobile if the Society did not take prompt and effective steps to translate the idea of a horsepower so as to make it intelligible to the average man who knows nothing about machinery or about mechanics.

John Chucan assented to the remarks made by Professor Magruder and asserted that he had actually used for several years a formula identical with that proposed by Mr. Arnold and had found it to be very conservative. The formula, as he understood it, was intended only for tractors and not for automobile or any other engines.

The Author. There seems to have been some argument and objection to the proposed formula though it seems to me to result from a failure to realize the necessity of such formula.

First: It must be understood that any formula that can be devised at this time is purely empirical and is established for a basis of comparison.

Second: That having established a standard basis by which engines of this type shall be rated, each engine can then be compared in relation to the per cent of maximum overload they can sustain under actual brake test for a given period of time.

Third: This formula applies equally to gasoline as well as kerosene engines, the only difference being that under certain conditions the gasoline will probably sustain a greater maximum.

No. 1700

STANDARDS OF CARBURETOR PERFORMANCE

By O. C. Berry,[1] Lafayette, Ind.
Non-Member

The type of test which a carburetor is most frequently given to establish its merit or lack of merit is carried out in the following manner: An engine in good mechanical condition is tested, using a carburetor which bears a high reputation. These tests are then repeated, using the new carburetor, and its merits are thus reported in terms of the comparative performances of the engine. The results of these tests are valuable and convincing, and while they will always be the final criterion of carburetor performance, they fall a little short of the ideal in that they fail to show the reasons for the differences. Several writers have recently pointed this out and suggested that the performance of a carburetor should also be expressed in terms of its ability to perform those functions which are essential to proper carburation. In order to do this it will be necessary to determine these essentials. Each function must then be studied to see when and how it operates, its comparative importance, and the conditions for the best results. This is a difficult task, but its accomplishment should prove of great benefit. In this paper a list of the essential factors, as now understood, has accordingly been made. Some experimental data are also presented which it is hoped will help to establish some of the standards of performance. These data cover the following points:

(*a*) *The efficiency and power capacity of an engine as affected by the richness of the mixture*
(*b*) *The effect of the speed of an engine upon its mixture requirements*
(*c*) *The effect of the torque produced by the engine upon its mixture requirements*
(*d*) *The effect of the dryness of the mixture upon the mixture requirements, power and efficiency of the engine:*
  1 *The heat for drying the mixture coming from the sensible heat in the air*
  2 *The heat for drying the mixture coming from a "hot spot" in the intake manifold.*

Further research is now under way which is expected to throw light on other phases of the problem.

SATISFACTORY carburation of a liquid fuel depends jointly upon the carburetor, the intake manifold, and the temperature of the combustion-chamber walls. These parts may therefore be considered the carburating apparatus and the headings under which

[1] Professor of Mechanical Engineering, Purdue University.

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

are grouped the tests to determine the degree of excellence with which each of these parts performs its respective functions may all be listed together. These headings expressed in the general order of their importance are as follows:

a The range of flow-rate capacities, or, in other words, the maximum and minimum number of cubic feet of air per minute that can be handled
b The richness of the mixture as affected by: the rate of flow of the air through the carburetor; sudden changes in the rate of flow; the amount the throttle is open
c The pressure drop through the carburetor at different rates of flow
d The thoroughness and uniformity with which the fuel and air are mixed
e The uniformity of the richness of the mixture furnished to the different cylinders
f The temperature and dryness of the mixture entering the cylinders
g The temperature of the combustion-chamber walls, particularly the piston head.

The relative importance of these items is largely a matter of opinion and will be different under different conditions. For example, the capacity for handling large amounts of air is much more important in passenger cars than in trucks and tractors, while in tractors using kerosene the temperature of the mixture and combustion-chamber walls becomes comparatively much more important than it is in passenger cars burning gasoline.

a RANGE OF FLOW-RATE CAPACITIES

2 In passenger-car engines great importance is attached to flexibility, and the buying public would not consider a car that was notably defective in this respect. The power developed by an engine varies almost directly with the weight of air used per minute. The carburetor must therefore be able to supply this air, properly mixed with fuel, through a wide range of flow rates. As in practice the carburetor is frequently found to be one of the main factors limiting the flexibility of the engine, the range of flow-rate capacities is placed at the head of the list as probably the most important item by which the carburetor is to be judged.

3 Carburetors are usually rated as to size by the size of their outlet flanges. No account is taken of the maximum or minimum rate of flow, though in practice there is a marked lack of uniformity in the flow-rate capacities of carburetors of the same rated size. Nevertheless these carburetors are supposed to be interchangeable. The range of flow-rate requirements of various engines also differs widely, being large for the passenger-car engine and small for the truck and tractor engines.

4 The passenger car is expected to idle down to one or two miles per hour on high gear and at the same time have a maximum speed capacity well above 50 miles per hour. Few carburetors are capable of furnishing a proper mixture over so wide a range, although most carburetors are sold under claims of wonderful flexibility. The passenger-car engineer can only be sure of a proper carburetor by specifying the exact flow rates required to secure the desired flexibility. Similar specifications with the less exacting truck and tractor engines, which are usually governor-controlled, will frequently result in a marked saving through the substitution of a cheaper carburetor of more rugged design.

5 A carburetor is truly suited to an engine only when its flow-rate capacities correspond to the requirements of the engine, and this condition will be attained with the greatest certainty when the requirements of the engine and the capacities of the carburetor are both given in cubic feet of air per minute. It is therefore suggested that the flow-rate capacities of all carburetors should be stated definitely in cubic feet per minute, and that this information should always accompany the statement of the size of the carburetor flange.

6 The air required by an engine may be computed as follows: With the carburetor adjusted so that the engine carries its full torque with open throttle, the air required per brake horsepower per minute will remain nearly constant irrespective of the amount of gasoline used or the speed of the engine. For the usual passenger-car type of engine with a compression ratio of about 4 to 1 this constant is about 2.1 cu. ft. per min. at full power, and seldom exceeds 2.3 cu. ft. per min. The air used when idling at any speed is almost exactly one-quarter of that used under full load at the same speed. With these data, which are based upon the performance of the engines tested at Purdue, it is easy to compute the air requirements for any engine of the same type, for any torque and any speed.

b RICHNESS OF THE MIXTURE

7 The thermal efficiency of an engine at any speed and load is affected more by the richness of the explosive mixture than by any other factor. It is therefore highly important that the carburetor deliver the mixture to the engine in the proper proportions, at all speeds and loads. In order to be sure that this is being accomplished, it will be necessary to do several things. In the first place, the richness of mixture giving the best power and the one giving the best efficiency must each be experimentally determined. This must be done for various speeds and loads, so as to learn what effect a change in either will have on the mixture requirements of the engine. The temperature and the dryness of the mixture must also be varied to determine whether or not they affect the power or efficiency of the engine. After thus establishing the results desired of the carburetor, it will be necessary to determine the types of failures observed in carburetors in meeting this requirement. Tests may then be outlined that will establish the degree of excellence of the carburetor with respect to the richness of the mixture furnished to the engine.

8 In the Purdue tests the richness of the mixture is expressed in pounds of gasoline per pound of dry air. With a dry mixture at half load and 1000 r.p.m., regular firing may be obtained with mixtures between 0.0575 and 0.12. The best efficiency under the same conditions accompanies a mixture of about 0.067, and the best power, 0.08. The point at which the engine begins to miss is hard to determine as the missing appears so gradually. Likewise it is hard to choose the point of highest power or highest efficiency, as the curves are both flat on top. These points were carefully estimated, however, and the figures reported are close approximations. The method used in obtaining these results and a few of the characteristic curves will be presented at the end of this paper.

c PRESSURE DROP THROUGH THE CARBURETOR

9 The ideal condition of perfect volumetric efficiency is not attainable, since the air and gasoline must be caused to flow into the cylinders. It is always necessary to have the gasoline in the float chamber of the carburetor at a lower level than the delivery orifice in order to prevent overflowing when the engine is not running, and the suction in the carburetor must be great enough to overcome this safety head before any gasoline will be delivered. The

best attainable condition, therefore, will be to have just enough vacuum in the carburetor to cause a satisfactory flow of fuel and air, and no more. Because of the influence of volumetric efficiency on engine capacity the importance of a small carburetor vacuum is very great. For this reason designing engineers should insist on data showing the pressure drop through the carburetor necessary to give the desired rates of flow. This drop in pressure should be measured at the throttle on the carburetor side and the method of making the measurement should be specified very carefully. The demand for this information would soon induce carburetor manufacturers to publish guaranteed vacuum-air-flow curves, thus making possible intelligent selection of a carburetor for a given service.

d THOROUGHNESS OF THE MIXING

10 The thoroughness and uniformity with which the fuel is mixed with the air is important. One of the greatest helps in reducing the fuel to a gas is to atomize it thoroughly and mix it with the air. This gives the fuel a large amount of exposed surface and helps to bring all parts of the mixture up to the same degree of saturation. When the fuel leaves the carburetor as a liquid not well mixed with air, it flows toward the cylinders more slowly than the air, and consequently collects in the manifold and arrives at the cylinders in waves, causing the mixture to vary widely in richness. It is very difficult to design a manifold that will distribute the liquid evenly to all of the cylinders. All of these facts add to the importance which we must attach to the thoroughness with which the fuel is atomized and mixed with the air.

11 The mixture in all parts of each individual cylinder must be uniform and the fuel reduced to a gas at the end of the compression stroke or else the combustion will not be complete, thus lowering the power and efficiency of the engine and causing uneven running. Therefore, if the engine runs with a regular and even exhaust, the thoroughness of the mixing is probably good along with all of the other factors influencing engine performance. The best direct test of the quality of the mixing is to have the carburetor discharge into a section containing glass placed between the carburetor and manifold. Fig. 4 shows the design used in the Purdue tests. The best dry mixtures appear as colorless and dry as pure air, while wet mixtures resemble a fog, and in most cases streams of liquid fuel are seen following a spiral path along the walls of the manifold.

e UNIFORMITY OF MIXTURE TO ALL CYLINDERS

12 The richness of the mixture entering the various cylinders often differs widely, especially when very wet mixtures are used. Some manifold designs also aggravate this condition. As dry mixtures not only are seldom used but are also of questionable desirability, it would seem that the problem resolves itself largely into one of manifold design. A direct quantitative test of the richness of mixture to each cylinder is impracticable, but the results attained by any given manifold may be tested by removing the exhaust manifold and observing the flames from the exhaust openings. This can be done to best advantage in comparative darkness. By adjusting the carburetor for continuously leaner mixtures the impoverished cylinders will be caused to miss, and then by gradually enriching the mixture the yellowish flame will indicate the cylinder with the rich mixture. Uniformity is of course the desired goal.

f TEMPERATURE AND DRYNESS OF THE MIXTURE

13 With the rapidly increasing difficulty in vaporizing the commercial liquid fuels, the temperature of the mixture becomes a more and more important consideration. Before the fuel can be burned it must be vaporized, and this requires both heat and time. The heat may come from the sensible heat in the air entering the carburetor, or from hot surfaces in the carburetor, intake manifold or combustion chamber. Often the temperature and vapor pressure of the fuel are high enough to maintain a dry mixture, once it is established, but the time element is lacking and the mixtures are consequently quite wet. There is a difference between requiring a mixture to be dry as it enters the cylinder, and dry at the end of the compression stroke. The heat due to compression raises the temperature of the mixture well up toward the ignition point by the end of the compression stroke, so that any fuel suspended in the form of a fog will tend to be vaporized. The temperature of the piston head is also high enough to aid materially in flashing into a gas any of the fuel which may enter the cylinder as a liquid. The objects to be accomplished are to have the mixture dry and thoroughly mixed at the end of the compression stroke in order to get good combustion, and to have the gas temperature as low as possible at the end of the suction stroke, in order to keep up the volumetric efficiency. It is therefore desirable to make the fullest possible use of the heat in the combustion-chamber walls, piston head

and compression, and to introduce the mixture into the cylinder as wet as possible and still be sure of having it dry before it is burned.

14 Interesting light is thrown on this point by some of the Purdue tests. A series of tests was run at 1000 r.p.m. and a half-load throttle setting with all of the conditions constant except the temperature of the air to the carburetor. Under these conditions an air temperature of about 300 deg. fahr. was necessary in order to have the mixture in the manifold dry. With the air at 300 deg. fahr. and the mixture dry the highest power was 12.5 b.hp. and the highest efficiency 18 per cent. As the temperature of the air was lowered the power was increased, until at 80 deg. fahr. the best power was 16 b.hp. or an increase of 3.5 hp. Below 80 deg. fahr. the firing of the engine became irregular and the power dropped off. The richness of the mixture giving the best efficiency remained the same, as the air temperatures were lowered to 150 deg. fahr., indicating that the mixture was dry when combustion started and the combustion complete. Below 150 deg. fahr. the mixture for best efficiency commenced to grow richer, indicating incomplete combustion and a possibility that the fuel was not completely vaporized before combustion took place. At 80 deg. fahr. this tendency had not become sufficiently pronounced, however, to counterbalance the increase in power, so that the efficiencies increased slightly, even down to 80 deg. fahr. At this point the best efficiency was 18.8 per cent, or an increase of 0.8 per cent. The engine ran smoothly throughout the entire temperature range when good pulling mixtures were used and it was found that with high temperatures a leaner mixture could be fired.

15 It is possible to compute approximately the change in power that will accompany a definite change in the temperature of the mixture furnished to the engine. The computation is based upon the fact that the density of a gas varies inversely as its absolute temperature. Since the power generated in the cylinder varies almost in direct proportion to the weight per minute of the mixture used, it will vary almost in the inverse ratio of the absolute temperatures of the mixtures. As an illustration, suppose the engine will develop 12.5 hp. when the mixture has a temperature of 250 deg. fahr. What will it develop when the temperature is 60 deg. fahr.? The power times the inverse ratio of the mixture temperatures is as follows:

$$12.5 \times \frac{460 + 250}{460 + \ \ 60} = 17.06$$

This indicates that 17.06 hp. is to be expected with the 60 deg. fahr. mixture. As a matter of fact, the temperature in the cylinders at the end of the suction stroke rather than the intake manifold temperature determines the density of the charge, but this temperature cannot be measured. Again, the indicated rather than the brake horsepower is the one that varies with the density of the charge, while the brake horsepower is the one that is usually measured. These are the main reasons why the results obtained are only approximately correct. The figures given are from two Purdue tests run at the same speed with the same throttle orifice, and the power was increased from 12.5 to 16.0 instead of 17.06 b.hp.

16 The task of determining the best method of introducing into the mixture the heat for vaporizing the fuel is both important and difficult, and will require considerable careful experimentation. The Purdue tests with the "hot spot" warrant the conclusion that it is superior to any method of preheating the air, in that it dries the mixture sufficiently at lower temperatures and, therefore, does not decrease the power of the engine so much. These tests also indicate that the design of the hot spot is still in need of experimental development.

g TEMPERATURE OF THE COMBUSTION CHAMBER

17 When the mixture is dry as it enters the cylinders, or the fuel so well atomized that it remains suspended in the air and is entirely vaporized during the compression stroke, the heat absorbed from the combustion-chamber walls does not improve the carburation, but decreases the power capacity of the engine without improving either its efficiency or the way it runs. These conditions, however, are rare. A considerable portion of the fuel usually enters the cylinders as a liquid, which collects on the piston head. Under these conditions the temperature of the combustion-chamber walls, especially the piston head, becomes very important. The reasons for this may be explained as follows: The piston head is usually at a temperature two or three hundred degrees above the cylinder walls. If the temperature of the walls is 200 deg. fahr. the piston head will therefore be between 400 and 500 deg. fahr. If the wall temperature is lowered to 100 deg. fahr., the piston-head temperature will drop to between 300 and 400 deg. fahr. These temperatures apply to passenger-car engines running under ordinary conditions. Several tests have been run at Purdue to determine the

rate at which Red Crown power gasoline will evaporate from the surface of a hot iron plate. The maximum rate of evaporation seems to occur when the metal is about 450 deg. fahr. If the evaporation in ounces of fuel per square inch of metal per second be taken as 100 per cent at 450 deg. fahr., then the evaporation at 400 deg. fahr. is about 40 per cent, at 350 deg. fahr. about 9 per cent, and at 300 deg. fahr. about 1.8 per cent. It is therefore important that the jacket-water temperature be kept high when the piston head is depended upon to flash any considerable amount of liquid fuel into a gas. This conclusion is borne out by the engine tests. With the air entering the carburetor at 70 deg. fahr. and the jacket water maintained at 110 deg., the engine would not fire regularly with any richness of mixture. When the jacket-water temperature was raised to 200 deg. fahr. the engine would fire some of the richer mixtures regularly and by raising the air temperature to 80 deg. fahr., the engine developed full power and efficiency and would fire a wide range of mixtures.

THE PURDUE TESTS

18 The tests at Purdue University were carried out on a Haynes Light Six and a Willys-Knight four-cylinder engine, mounted on a Diehl electric dynamometer, the majority of the tests, however, being on the latter. Fig. 1 shows the Knight engine mounted ready for a test. Fig. 2 gives a more detailed view of the scales and the electrical apparatus for starting and stopping the tests. The supply of gasoline is piped from tank *A*, Fig. 1, into a 2-qt. glass vessel placed in one of the scale pans. This is shown at *A*, Fig. 2. In order to give the balances freedom of motion, the gasoline was siphoned from this vessel to the carburetor. The balances *B*, Fig. 2, were capable of weighing the gasoline to the one-hundredth part of an ounce, and were equipped with wires dipping into mercury cups *C*, Fig. 2, which completed an electric circuit just when a balance was reached. The hand on the dial of the air meter was equipped with an electrically operated clutch and brake, the stop watch was operated by a strong magnet *D*, Fig. 2, and the revolution counter on the end of the dynamometer shaft was electrically operated. By these means, when the scales came to a balance they would start the stop watch, the revolution counter and the recording hand on the air meter and ring a gong. Weights corresponding to the gasoline to be used were then removed from the scale pan, and when

the scales again came to a balance the recording apparatus was electrically stopped, making it possible to time all of the readings together and make the record at leisure from instruments that were standing still.

19 The air was metered through an Emco No. 4 gas meter *B*, Fig. 1, reading to cubic feet, so that tenths of a cubic foot could be estimated. The drop in pressure through the meter was indicated by the water manometer *C*, Fig. 1. The barometer and the wet- and dry-bulb readings on a hygrometer were taken periodically, in order to determine the pounds of dry air used in each instance.

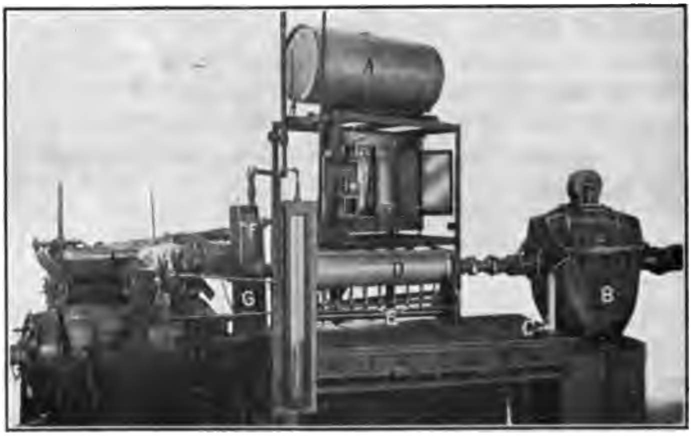


FIG. 1 WILLYS-KNIGHT ENGINE AND EQUIPMENT FOR TESTING ITS CARBURETOR

A, Gasoline tank; *B*, air meter; *C*, manometer; *D*, heater; *E*, gas jets; *F*, thermometer; *G*, water tank

20 The meter was connected to a heater *D*, Fig. 1, by means of rubber tubing, which was kept from collapsing by a coil of wire inside of it. The heater was made up of 4 ft. of 3-in. wrought-iron pipe, surrounded by an asbestos cylinder. Between the pipe and asbestos was a space large enough to allow the pipe to be heated by the flames from the gas burners *E*. The temperature of the air leaving the heater was indicated by the thermometer *F*. This thermometer passed through a packing gland in the air line, so that its bulb was exposed directly to the air inside of the pipe. The temperature of the air could be regulated within close limits by careful adjustment of the gas flames. Between the heater and the

engine the line was covered by a thick layer of hair cloth to prevent radiation. In some instances a section containing glass was inserted between the carburetor and engine.' The glass tubing was of the same inside diameter as the rest of the line and was carefully packed in a section of wrought-iron pipe. The sides of the pipe were milled away so as to offer ample opportunity to observe the mixture in the tube. By holding an electric light behind the glass as shown, very satisfactory observations could be made of the character and behavior of the mixture inside.

21 In the Haynes set-up the temperature of the cooling water

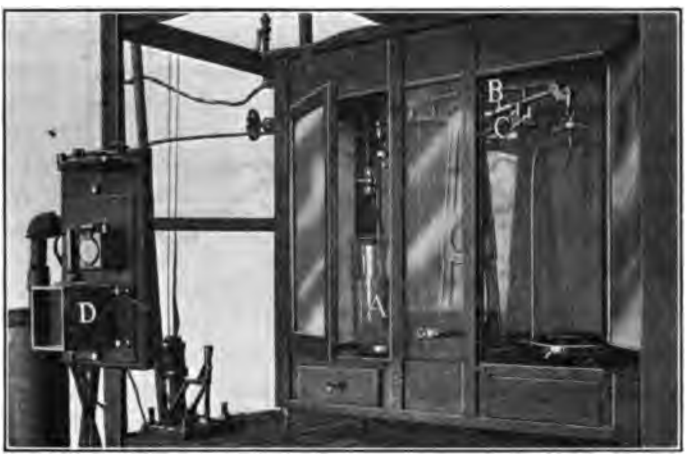


Fig. 2 Scales for Weighing Gasoline, and Electrical Equipment for Starting and Stopping Tests

A, Gasoline beaker; *B*, balances; *C*, mercury cups; *D*, magnet

for the engine was kept between 125 and 135 deg. fahr. Circulation through the jackets of the engine was induced by the standard Haynes engine-driven pump. In the Willys-Knight tests the tank *G*, Fig. 1, was used, but only for a time, it being replaced by a radiator from a Liberty B. truck. Water was also introduced directly from the service lines of the laboratory. In all of these tests the temperature of the cooling water was carefully recorded.

22 The speed of the engines was read on a tachometer as well as being computed from the stop-watch and revolution-counter readings, giving a good check on this important factor. The torque developed by the dynamometer was weighed by means of a sensi-

tive set of Fairbanks scales. This is the same as the brake load on the engine, thus making it possible to compute the power developed by the engine to a satisfactory degree of accuracy.

METHOD OF CONDUCTING THE TESTS

23 The object of the first series of tests was to determine the effect of changing the richness of the mixture on the performance

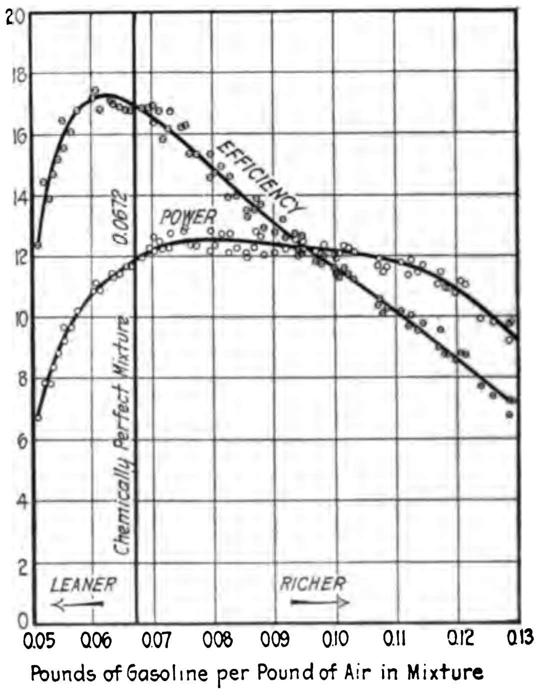


Fig. 3 Power and Efficiency Curves of Willys-Knight Engine

Throttle set for half-load. Speed, 1000 r.p.m. Cooling-water temperature, 200 deg. fahr.

of an engine. It was planned to determine the mixture that would give the best power, the one for the best efficiency and the range of mixtures that could be fired regularly. This was done as follows: A thin steel plate was placed between the carburetor and intake manifold, and the throttle removed. In this plate was drilled a hole of such size that when a powerful mixture was used the engine would develop the desired power at the desired speed. A series of tests was then run with this orifice and at the given speed, and a set of power and efficiency curves plotted similar to Fig. 3. In this case the speed was 1000 r.p.m. and the orifice was for half load. In the first test in this series the weight of gasoline per pound of air in the mixture was computed, together with the power and efficiency developed by the engine. This gave the first point on each of the curves. The mixture was then made slightly richer by opening the gasoline needle and the brake load was adjusted to bring the speed back to 1000 r.p.m. A second test was made under these conditions, and another point on each of the curves determined. This was repeated until the engine would miss and could not carry anywhere near its original load. The amount of gasoline was then gradually decreased each time and a series of tests made until the mixture was so lean that the engine would not perform properly. This process was repeated, making the mixture alternately richer and leaner until a large number of determinations had been made, and the results when plotted gave the desired information with satisfactory clearness. An attempt was made during all of the tests to keep the temperatures of the jacket water and carburetor air constant. as well as the spark setting.

ACCURACY OF RESULTS

24 When a single series of tests is run in a continuous sequence from the lean to the rich end of the mixture range, the plotted points tend to fall in a consistent line and the results may seem to have an accuracy that they do not possess. However, if the series is repeated back and forth a number of times, the points will tend to vary somewhat from the original line and show clearly the degree of accuracy being attained. An engine running under what seem to be constant conditions will vary in its performance, and the best apparatus is liable to a certain amount of error, so that before any far-reaching conclusions are drawn from experimental results it is desirable to know their limits of accuracy. For this reason the points in Fig. 3 were obtained as suggested above, and since this work was done at the beginning of the entire series of tests, before the men became as expert as they were later, it is felt that the errors are never greater than is indicated by these curves.

24 The vertical line in Fig. 3 (at 0.0672) represents the chemically perfect mixture, or the one in which there is just enough oxygen in the air to burn the fuel and no excess of either fuel or air exists. The curves show that the engine will run with a mixture as lean as

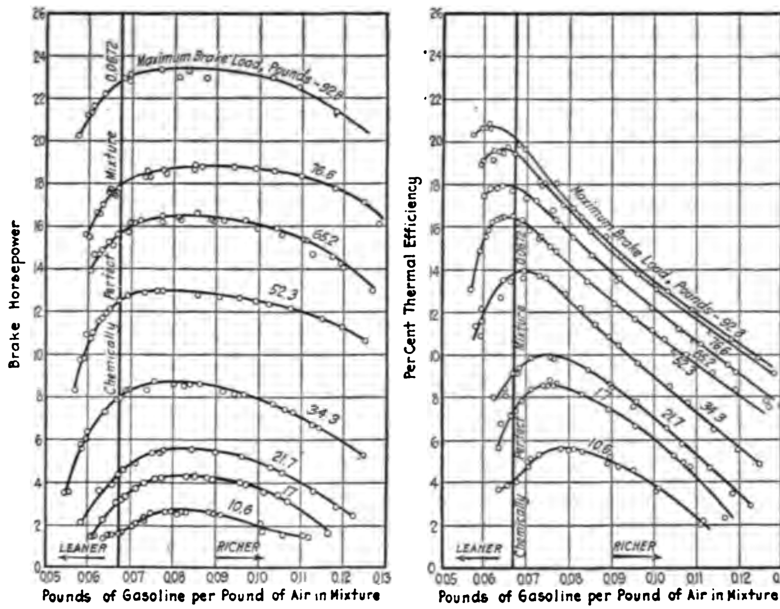
0.05 lb. of gasoline per lb. of air, but will not pull well with so lean a mixture. The test log shows that it misses frequently at this power, but that the performance becomes better as the mixture is made richer, until at 0.055 it fires every cylinder regularly. The best efficiency is obtained at 0.063, when the engine is developing 91 per cent of its maximum power capacity with this orifice at 1000 r.p.m. The best power accompanies a mixture of about 0.08, at which point the thermal efficiency has dropped from 17.25 to 14.8 per cent. The richest mixture that can be fired regularly is about 0.1275, but the engine will run with mixtures as rich as 0.135. Nearly full load can be carried with a mixture as lean as 0.065, or as rich as 0.115. In other words, a carburetor can be adjusted with as lean a mixture as can be used to carry full load, and the amount of gasoline can be nearly doubled, without greatly affecting the power developed or the smoothness of running of the engine. It is practically impossible to stand by the side of an engine mounted on a test block and distinguish any difference in its performance as the mixture is being changed through this range.

25 In applying the information obtained from these tests to other conditions it must be remembered that these results are for half-load at 1000 r.p.m., when a warm mixture was used that was dry as it left the intake manifold. Before applying the conclusions to other conditions, one must learn the effect of the load carried, the speed and the temperature and dryness of the mixture upon the mixture requirements, power and efficiency of an engine.

EFFECT OF LOAD UPON MIXTURE REQUIREMENTS

26 Figs. 4 and 5 show the power and efficiency curves taken from the Willys-Knight engine running at 1000 r.p.m., but with different throttle orifices. The figures given on each curve indicate the largest brake load carried with that particular orifice at 1000 r.p.m. During these tests the temperature of the air entering the carburetor was kept at about 150 deg. fahr., and the cooling water at about 120 deg. fahr. This was true of all of the curves excepting the one for 92.8 lb., in which case the mixture was heated up to 125 deg. fahr. in a "hot spot" and the cooling-water temperature was 160 deg. fahr. Fig. 4 shows that the mixture giving the best power at a fixed throttle setting does not vary with the brake load carried, but remains constant at about 0.08. At light loads the engine will not operate well with as wide a range of mixtures as it can use when

carrying more nearly its full capacity. Fig. 5 shows that with light-load throttle settings the mixtures for best power and best efficiency tend to coincide, but as the brake load is increased the mixture for best efficiency becomes continuously leaner, until at full load it is 0.062. In the case of the higher brake loads the engine will hit regularly and run smoothly with the lean mixtures which give the high efficiencies, but the power developed is reduced considerably below the highest attainable at that throttle setting. The most



FIGS. 4 AND 5 POWER AND EFFICIENCY CURVES OF WILLYS-KNIGHT ENGINE

Throttle set for various loads. Speed, 1000 r.p.m. Temperature of air to carburetor, 150 deg. fahr. Cooling-water temperature, 120 deg. fahr.

satisfactory mixture for general use at or near full load will, therefore, be approximately 0.067, giving almost full power and nearly the best efficiency. For lighter pulling conditions the mixture had better be caused to approach 0.08, the one for best pulling.

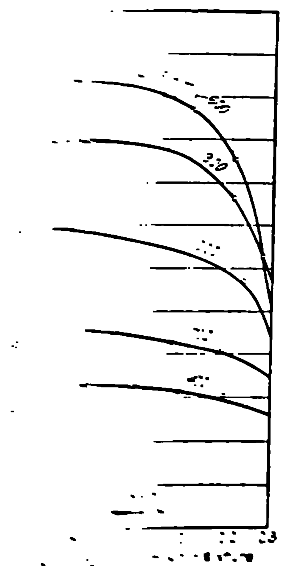
EFFECT OF SPEED UPON MIXTURE REQUIREMENTS

27 In Fig. 6 is shown a set of curves taken from tests on the Haynes Light Six engine running at different speeds from 400 to 1600 r.p.m., and in each case with a throttle orifice giving about

...ture giving the best power ... that at high speeds the ... as much excess fuel as ... that the mixture for the ...

TEMPERATURES

... taken from the Willys-... a half-load orifice as a

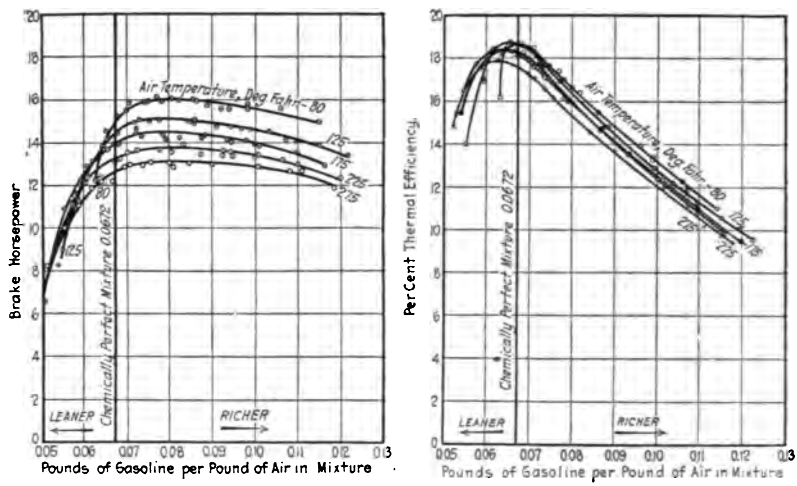


... Light Six Engine

... 100 deg. fahr. Cooling-...

... the carburetor at tempera-... They show that 80 deg.

mixture as lean as 0.055. Fig. 8 gives the efficiency curves. The best efficiencies at 80 and 125 deg. fahr. are nearly exactly the same, 18.75 per cent, but at 80 deg. fahr. it accompanies a mixture slightly richer than for 125 deg. fahr. Each increase in temperature between 125 and 275 deg. fahr. decreases the efficiency slightly. At 150 deg. fahr. the mixture for the highest efficiency reaches its leanest point, 0.063, and remains the same up to 275 deg. fahr. The effects of increasing the temperature of the air entering the carburetor above 80 deg. fahr. are therefore to decrease the power capacity of the engine considerably and its efficiency slightly, but to make it fire regularly when using leaner mixtures.



FIGS. 7 AND 8 HEATED-AIR TEST CURVES, WILLYS-KNIGHT ENGINE

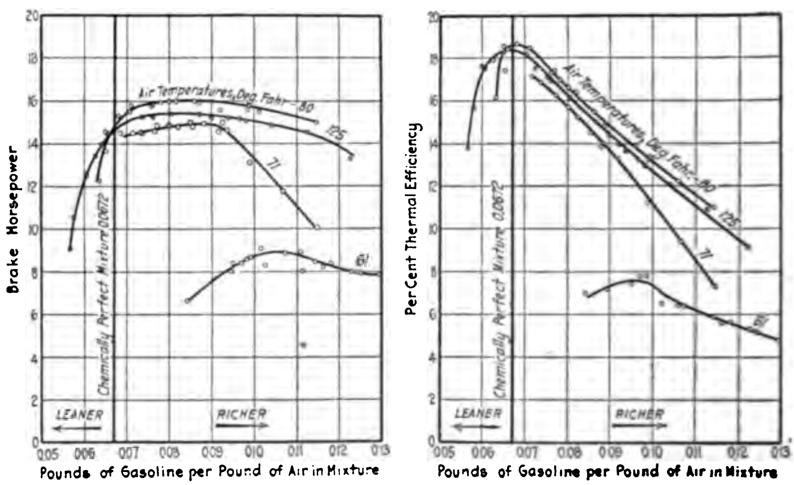
Throttle set for half-load. Speed, 1000 r.p.m. Cooling-water temperature, 200 deg. fahr.

29 Figs. 9 and 10 show the curves for the same series of tests, but with the colder air temperatures. Fig. 9 shows that with an air temperature of 61 deg. fahr. the engine could use only the comparatively rich mixtures and its power capacity was greatly reduced. When the air temperature was increased to 71 deg. fahr. the power was brought nearly up to maximum, but still the engine could not fire the leaner mixtures with regularity. Fig. 10 shows that the efficiency for the 61 deg. fahr. air is very poor, while the 71 deg. fahr. air was much better but the engine stopped before it reached a lean enough mixture to give the best results. It may therefore be seen that 80 deg. fahr. is about the lowest temperature at which the air

may be drawn into a carburetor when using Red Crown power gasoline, to get good carburation in an engine having a 200 deg. fahr. cooling-water temperature. If the cooling-water temperature is lowered the temperature of the air will have to be raised correspondingly, while raising the quality of the gasoline will improve the performance with cold air, cool water or both.

THE HOT-SPOT TESTS

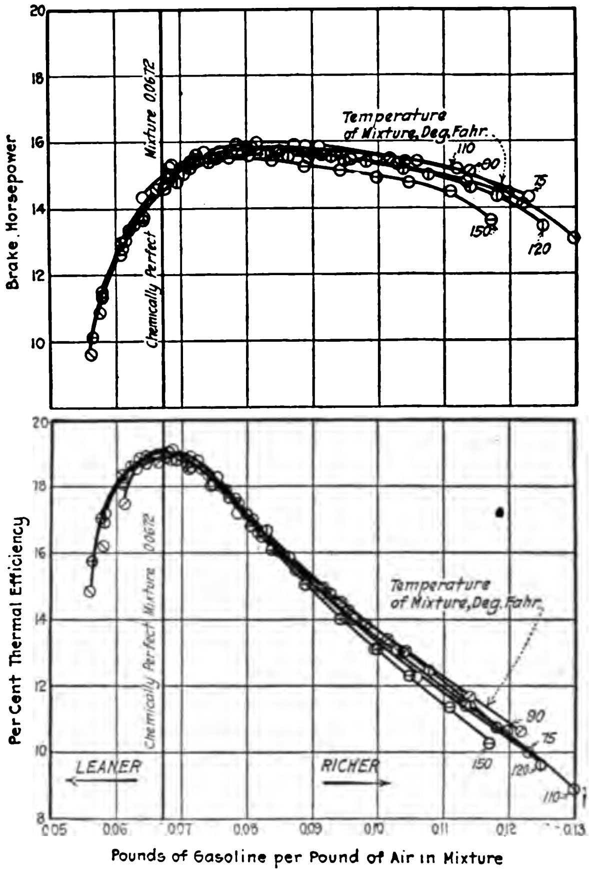
30 Figs. 11 and 12 show curves taken from the Willys-Knight engine using a so-called "hot spot" between the carburetor and



FIGS. 9 AND 10 COLD-AIR TESTS, WILLYS-KNIGHT ENGINE

Throttle set for half-load. Speed, 1000 r.p.m. Cooling-water temperature, 200 deg. fahr.

the intake manifold. This hot spot was designed to flash the liquid fuel into a gas while heating the air as little as possible. The curves show that by using the hot spot the engine is able to fire the lean mixtures without in turn lowering either its power or its efficiency. The explanation is that the factor controlling the power of the engine is the weight of air that can pass through the throttle orifice in a given time. The air always being cold when it passes this point, the power is not affected by the heat. This is always true for a throttled condition, but as the throttle is opened and the manifold or valves become the limiting factors in the production of power, the advantage of the hot spot decreases, but it does not disappear. It is there-clear that the hot-spot method of introducing heat into the ture is always superior to the hot-air method, and particularly when the engine is throttled down.



FIGS. 11 AND 12 "HOT SPOT" TESTS, WILLYS-KNIGHT ENGINE

rottle set for half-load. Speed, 1000 r.p.m. Cooling-water temperature, 200 deg. fahr.

CONCLUSIONS

31 It may be well to again point out the importance of judging merit of a carburating system in terms of the degree of thorness with which the system performs those functions which necessary to proper carburation. It is hoped that a discussion he topic will result in establishing specifications for these tests,

and that many may join in the task of carrying out the experiments that will determine the standards of performance in each case. Tests are needed showing the richness of mixture that will give the best power and the one for best efficiency when using liquid fuels other than gasoline. It would also be interesting to extend the gasoline tests to include wider ranges of speed and higher compression.

32 The problem of determining the best method of introducing into the mixture the heat necessary to vaporize the fuel is one of great importance, especially in connection with the heavier fuels. As a part of this problem it will be interesting to determine the actual temperatures found in the metal of the piston head and cylinder walls of automobile, truck and tractor engines, and how these temperatures affect the carburation.

33 The fullest discussion of these problems and the methods employed at Purdue in attacking them is earnestly solicited, and any suggestions in connection with carrying out this line of investigations will be received most gratefully by those in charge of the Purdue Engineering Experiment Station.

ACKNOWLEDGMENTS

34 The experimental work presented in the foregoing paragraphs was carried out in the laboratories of Purdue University under the auspices of the Purdue Engineering Experiment Station, and has received throughout the personal attention and encouragement of Prof. G. A. Young, Head of the School of Mechanical Engineering. Most of the tests were carried out and the computations made by Mr. C. S. Kegerreis, Research Assistant. It is hoped that the Experiment Station will have a bulletin on carburation ready for distribution by early winter. The plan is to make this a detailed report of the work which is summarized in this paper, together with such other data as are then available for publication. Copies of this bulletin may be obtained by addressing C. H. Benjamin, Director of the Engineering Experiment Station, Purdue University, Lafayette Indiana

DISCUSSION

[illegible] remarked that the carburetor design was intimately connected with the structural details of the engine, with [illegible] compression parts, and especially with the form of the manifold. The real problem of carburetor design, as he saw it, was to get the combination of a carburetor and inlet port in an engine which would give a uniformly suspended mixture of minute droplets in air at the instant the spark fires the charge. But this would not lead, say, to the knowledge of the best kind of a combination carburetor, manifold and engine which would enable an automobile to climb a good stiff hill at ten or fifteen miles per hour, and at the same time give a speed of three miles per hour in a crowded traffic street and a speed of fifty miles per hour on a smooth, clear road.

Mr. Cardullo further observed that a rich mixture could be secured at the instant of acceleration more by properly designing the form of the manifold than by providing methods for increasing the mixture. Also that by reason of the separator action of the manifold it would be impossible to get a representative sample of the mixture entering the cylinder, and he believed the proper thing to do would be to analyze the exhaust of each cylinder for the proportion of carbon dioxide in order to determine whether each cylinder was getting its due proportion of gasoline in the mixture provided.

Thomas J. Litle, Jr.,[1] observed that in automobile practice many difficulties are encountered which do not occur in stationary engine work. With reference to the statement in the paper that the best performance is obtained with a temperature of 80 deg., he preferred to take the temperature of the mixture entering the block, after it passes through the carburetor intake manifold. He claimed that the temperature of the charge as it entered the block, after passing through the carburetor and intake manifold, was the controlling factor in motor performance. He suggested as a future line of research work tapping into the intake passage just at the end of the supply pipe and analyzing a sample of the mixture taken at that point. To a question of Chairman Magruder inquiring how that sample of air and gasoline vapor was going to be obtained, Mr. Litle explained that he had succeeded in drawing it off by lifting the valve of a tall gasometer with a long water still.

The Author. The definition which Professor Cardullo gives of the object to be accomplished in carburetion is one that probably will not satisfy all automotive engineers. The temperature attained

[1] Research Engineer, Lincoln Motor Company, Detroit, Mich.

at the end of the compression stroke will usually be high enough to vaporize all of the minute droplets of the liquid fuel which were suspended in the mixture at the end of the suction stroke. The ideal condition seems to be to have the mixture dry and uniformly mixed at the end of the compression stroke or at the instant the spark fires the charge. There are two main objections to a wet mixture in the manifold. The first is that it tends to wet any surface it comes in contact with and thus gives up a large part of its liquid content on its way to the cylinder and this deposited liquid is very hard to distribute satisfactorily. The second is that in many cases this liquid is not vaporized in time to be mixed with the air and burned, and is consequently wasted.

In carrying out these tests it was decided to avoid the difficult task of obtaining a true sample of a wet mixture, as Mr. Lytle suggests, but to use a mixture that was hot enough to be entirely dry. A part of the tests here reported were accordingly run under these conditions. Furthermore, the tests seem to indicate that the temperature and dryness of the mixture at the end of the compression stroke are the important factors rather than the temperature of either the air entering the carburetor or the mixture in the intake manifold.

In the "hot spot" tests no single test was continued long enough to give any considerable carbon deposite. With the heavier fuels some deposit will probably collect at any temperature that is high enough to vaporize it rapidly, but the Purdue tests offer no information on this point.

No. 1701

PULVERIZED COAL AS A FUEL

By N. C. Harrison, Atlanta, Ga.
Member of the Society

The author first reviews some of the uses of pulverized coal in the industries — such as the cement, steel and copper industries, after which he gives a technical definition of pulverized coal, describes the process by which it is prepared for use and furnishes tables of costs of preparation. The pulverized-coal-burning open-hearth steel plant of the American Iron & Steel Manufacturing Co., Lebanon, Pa., is described and the advantages observed in the plant of pulverized coal, compared with producer gas as a fuel for open-hearth furnaces, are listed.

The use of pulverized coal in stationary boiler plants is discussed, five determining factors in the successful operation of such a plant being taken up in detail. As compared with mechanical-stoker plants the advantages of the pulverized-coal plant are enumerated and certain precautions to be observed with the latter type of plant are brought out.

A report of a test of a 468-hp. Edge Moor boiler with pulverized-coal equipment is included in the paper and the efficiency obtained is compared with the efficiency of a stoker-fed boiler in the same plant, a greater net efficiency being found in the pulverized-coal plant.

The paper concludes with a statement of some advantages obtained in the pulverized-coal plant.

THE peculiar conditions as they exist today, on account of the war and for other reasons, such as the gradual disappearance of sources of fuel, like natural gas, and the shortage in the supply of crude oils, which have become of too great value for ordinary fuel purposes, have compelled those interested to consider the possibility of the adoption of pulverized coal as a fuel, to replace their present methods of operation. Pulverized coal was first used in the United States about 26 years ago for the economical burning of the cement rock in the rotary kilns of the portland-cement industry.

2 Pulverized coal was first applied successfully for economical reasons in connection with the burning of portland cement. The growth of the portland-cement industry also had a great bearing on the development and use of pulverized coal, in that, it is in this

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

For discussion see p. 392.

industry that pulverizing machines were brought to the present high state of development, for in the manufacturing of cement not only the coal is pulverized but for each barrel of cement manufactured weighing 380 lb. there are required about 600 lb. of raw material such as limestone shale or cement rock as well as the 380 lb. of clinker produced by the kilns which must be pulverized in order to make the finished product, so that in the neighborhood of 1100 lb. of raw materials, clinker and coal must be ground to produce one barrel of portland cement. As there are a hundred million barrels of portland cement made in this country annually, these figures will give one a reason why pulverizing machines have been so highly developed during the last few years. Fine grinding of the raw material means reduction in the quantity of fuel required and also makes possible the highest quality of the finished product, so far as the chemical analysis or combination is concerned. Fine grinding of the clinker means increased strength for the reason that the hydraulically active units in cement are in direct proportion to the percentage of fine or impalpable powder in the finished product.

3 This statement is made to impress upon you that equipment for preparing and handling pulverized coal has long since passed the experimental stage and has now been developed to a high state of efficiency and is readily obtainable. Somewhere between 30 and 50 million tons of pulverized coal have been used to date in the manufacture of cement alone. There is now being used approximately 6 million tons annually.

4 The application of this form of fuel has been gradually taken up by engineers connected with other industries, who have speedily recognized its value to such an extent that the steel industry today is using in the neighborhood of two million tons of pulverized coal annually in various types of furnaces such as open-hearth, heating, puddling, soaking pits, continuous-heating, reheating, annealing, forging furnaces, and furnaces of practically every description where heat is required.

5 The copper industry is using between one and two million tons per year in ore-roasting furnaces, reverberatory and copper-melting furnaces of all types. Large amounts of pulverized coal are used in rotary kilns (other than the cement industry) for the desulphurizing and roasting of various grades of ores; for nodulizing blast-furnace flue dust so as to make available products heretofore very expensive to recover, for burning lime to oxide of lime for use in open-hearth furnaces, for burning dolomite for open-hearth

furnaces; and for the calcining (or driving off of CO_2 gas and water of crystallization) of various minerals, from which are obtainable such commodities as plaster of paris, stucco, potash, etc. A total of approximately ten million tons of pulverized coal are burned annually in the United States in the above industries.

6 A still further and very important development is now going on, which will, when it attains its growth, require more pulverized coal than probably all of the other industries combined, and that is in its application to locomotives, particularly in the West. This application is now being developed. There is still another field in which enormous quantities of this fuel will be used and a field in which we are all concerned, and that is in the generation of power in stationary power houses.

7 Practically any coal can be burned in pulverized form with a proper furnace and burning equipment. Each application however must necessarily be governed by the quality of the fuel available in the district in which it is made. Generally speaking, however, the coals which would give the most satisfactory results would be those in which the ash content would be less than 10 per cent, the volatile averaging between 30 and 40 per cent and the fixed carbon between 40 and 50 per cent. The sulphur content should be low, although coal with a sulphur content running as high as 4½ to 5 per cent is being burned in pulverized form under boilers without any detrimental results. The ash should have a high melting point. These statements, however, are tentative, as most excellent results have been obtained from all sorts of coals, differing widely from the ideal analysis stated.

8 It is very apparent that the development in this method of burning coal has brought coals, from which heretofore very inefficient results have been obtained, within reach of a great many consumers. For instance, from Texas to Edmonton, Alberta, the country is underlaid with various grades of lignites, low-grade mineral coals with high moisture content and of such a nature that the ash would melt or flow down on the grates, thereby preventing the highest efficiency from being obtained. They are of such a nature that their use in gas producers is not very satisfactory, so that until the development and burning of these coals in pulverized form was an assured success these coals were not used in as large quantities as is now possible. The largest deposits of lignite and mineral coals appear to be in the Northwest awaiting future development when proper means are at hand for obtaining the highest

possible economy from their combustion, and the location of these large deposits will now be of great value to the districts in which they are located.

9 Around steel plants there are large quantities of waste fuel such as coke breeze. This fuel is being used to a certain extent on some forms of grates, with forced draft, but it can be burned in pulverized form under boilers for the generation of power, and possibly in the open-hearth furnaces for making steel. In the anthracite field there are large quantities of coal daily pumped back into the mines, which coal is a result of the washing and crushing operation, for bringing the coal to commercial sizes. This silt or washery waste coal carries as high heat value normally as the coals which have been marketed. I understand that eight to ten million tons annually of this silt are allowed to be pumped back into the mines to fill up old workings.

10 A number of the coal companies are now carefully investigating the application of pulverized anthracite or low volatile coals with a view to using this waste coal in pulverized form so as to obtain power from its use, making available coals of higher grade for the market, which they are firing at the present time. A very successful installation of this kind is in operation at Lykens, Pa.

11 The above statements have been of a rather general nature so as to bring out forcibly the fact that coal in pulverized form is going to become one of the most important fuels. The results thus far obtained have shown that with installations properly designed and installed, that from an operating standpoint it is not only a desirable fuel but one which will eventually become necessary on account of its economy.

WHAT IS PULVERIZED COAL?

12 The average man will tell you that pulverized coal is coal ground to a powder. Any coal which is ground or powdered from his point of view is pulverized coal. From a technical standpoint pulverized coal is that coal which is properly dried, crushed and pulverized so that the product contains the highest percentage of impalpable powder. Merely powdering coal does not fulfill the requirements. Coal must be pulverized so that at least 95 per cent will pass through a 100-mesh sieve having 10,000 openings to the square inch, or in terms of dimension 95 per cent must be less than one two-hundredth of an inch cube.

13 The average person does not fully realize to what a high

degree of fineness it is possible to reduce the coal today by pulverization. The finer the coal is pulverized the more efficiently it can be burned and the more readily it will be diffused when mixed with the air for combustion and fed into the furnaces. A cubic inch of coal pulverized so that 95 per cent will pass through a 100-mesh sieve will contain over two hundred million particles, none of which will be greater than one one-hundredth of an inch cube, and a large percentage will be less than one six-hundredth of an inch cube. A cubic inch of coal has a superficial area of 6 sq. in., but the combined area of these multitudes of small particles shows that when the coal is ground to above mentioned degree of fineness the superficial area will increase to nearly 30 sq. ft. or an increase in area of approximately 700 times. This increase in area permits perfect and instantaneous combustion. Its rapidity depends directly upon the surface exposure. This is one of the reasons for the grinding.

14 Pulverizing certainly does not change the nature of the coal. We do however change the form of the coal to a certain extent in pulverizing it, in that we change it from a solid into a fuel having liquid properties. As the coal is pulverized it is mixed with air, and when handled in the conveyor it flows like water; when fed to the furnaces it is more or less like a gas, and the furnaces must be designed to burn a gaseous mixture.

DESCRIPTION OF A COAL-PULVERIZING PLANT

15 A pulverizing plant consists of three main units: a crusher, a crushed-coal drier and a pulverizer. The number of each one of these three main units will depend on the size of the plant. The coal is dumped in a track hopper and conveyed by either a belt or apron conveyor into hopper, feeding single-roll coal crusher, or where slack coal is at hand, direct to elevator pit. After being crushed to about one inch in diameter, it passes by gravity to elevator pit, where it is taken by bucket elevators to the drier storage bin. Automatic weighing scales may be installed if desired before the drier storage bin. Magnetic-separator pulleys are also installed, where a belt conveyor is used from the track hopper to remove iron or steel scrap in the shape of nuts, bolts, pick-points, wedges and such foreign matter, which would interfere with the pulverization.

16 From the drier storage bin the coal passes to a coal drier. This drier must be of a size to deliver the required quantity continuously, thoroughly dried. The drier is heated either by hand-firing on grates, or by pulverized coal, so arranged in either case to

avoid igniting the drying coal. The cylinder of the drier is rotated by power, either a small motor or line shaft being used. The dried coal falls from the drier through a chute into the pit of an elevator. In this chute the coal passes over another magnetic separator to remove any final pieces of metal which might be left in the coal and which were not caught on the magnetic separator spoken of above.

17 This elevator carries it to a storage bin set aloft for supplying the pulverizer. By spouts and gates the coal is permitted to enter the pulverizer as desired. This pulverizer grinds the coal to the fineness required. From the pulverizer the coal is conveyed in various ways to the pulverized-coal bins. With the type of mill used at our plant, it is carried by spouts from the mill to the pit of an elevator, which carries it aloft to the screw conveyor, which feeds the pulverized-coal bins. In another type of mill the fine pulverized coal is conveyed by suction fan from the mill to a cyclone separator, properly located over the pulverized-coal storage bins, or over the screw conveyor to the pulverized-coal storage bin. This separator will allow the coarser particles to fall back to the mill, to be reground, while the fine dust passes to the storage bin. No fine-dust elevator is necessary with this mill.

18 These pulverized-coal storage bins are of a capacity proportional to the service and hold a supply in excess of the amount required in the intervals when the grinding is not going on. Thus the mills may supply in eight to ten hours all that the furnaces may use in 24 hours, by making provision therefor.

19 If the pulverizing plant is located within 200 ft. of the furnaces and the furnaces are of large capacity, these storage bins are located directly at the furnaces. But if there are numerous small furnaces located at a considerable distance from the pulverizing plant, we then locate the above storage bins at some central point and convey from these points to the various furnaces by means of one of three methods: first, screw conveyor; second, in a mass by means of compressed air; third, in suspension in a current of air. We now have the pulverized coal at the furnaces ready for use.

COSTS OF PULVERIZING COAL

20 The cost of the operation of pulverizing coal depends upon four items: first, the amount of moisture that must be expelled from the coal before pulverizing; second, the cost of labor; third, the cost of coal delivered at the pulverizing plant; fourth, the cost of

electricity. Table 1, issued by a pulverized-coal engineering company, gives the cost of pulverizing plants, including buildings, and costs of pulverizing coal in plants of capacity from 10 to 250 tons per day. These figures include all costs, except interest and depreciation, in the pulverizing plant proper, and deliver the dust from the top of

TABLE 1 COSTS OF COAL-PULVERIZING PLANTS AND COSTS OF PULVERIZING COAL PER TON NET

Tons daily	Total cost pulverizing per ton, dollars	Cost of plant including building, dollars	Labor, hours	Labor, cost per ton, dollars
10	0.56	31,000	10	0.30
20	0.51	31,000	20	0.25
30	0.49	31,000	30	0.23
40	0.49	31,000	40	0.23
50	0.39	37,000	26	0.13
60	0.39	37,000	30	0.13
70	0.39	37,000	40	0.13
80	0.39	37,000	40	0.13
90	0.39	37,000	46	0.13
100	0.34	45,000	34	0.09
110	0.34	45,000	37	0.09
120	0.33	45,000	40	0.08
130	0.33	45,000	44	0.08
140	0.32	50,000	45	0.06
150	0.32	50,000	47	0.06
160	0.32	50,000	50	0.06
170	0.32	50,000	54	0.06
180	0.32	50,000	57	0.06
190	0.30	62,000	48	0.04
200	0.30	62,000	51	0.04
210	0.30	62,000	53	0.04
220	0.30	62,000	56	0.04
230	0.30	62,000	59	0.04
240	0.30	62,000	61	0.04
250	0.30	62,000	63	0.04

DATA ON WHICH TABLE 1 IS BASED

Labor rate: Millers, 30 cents per hr.; drier firemen, 20 cents per hr.; common labor, 20 cents.

Cost of drier fuel: 6 cents per net ton, based on 7 per cent moisture. Coal at $5 per ton delivered.

Evaporation: 6 lb. per lb. of coal burned or 26 lb. of coal per ton.

Repairs: 7 cents per net ton. This includes whole pulverizing plant, all machinery.

Power has been based on 12.7 cents per ton pulverized, and a consumption of 17 hp-hr. per ton pulverized at 1 cent per kw-hr. or about $54 per hp. per annum.

the last elevator to the screw conveyor, which feeds the pulverized-coal storage bins.

21 We have taken from Table 1 figures relating to those plants having daily capacities approximately the same as our plant, and

[illegible] of pulverized coal in this country [illegible] application in the cement industry [illegible] other types of metallurgical furnaces. [illegible] some cases increased production, have [illegible] coal within the last few years it has

[illegible] PULVERIZED COAL PER TON NET

[illegible]	[illegible]	Labor, hours	Labor, cost per ton, dollars
[illegible]	[illegible]	40	0.19
[illegible]	[illegible]	48	0.29
[illegible]	[illegible]	36	0.22

[illegible] Tons 2 in Series

[illegible] 35 cents per hr.; common labor, 25 cents. [illegible] 7 per cent moisture. Coal at $5 per ton

[illegible] lb. of coal per ton.
[illegible] pulverizing plant, all machinery.
[illegible] pulverized, and a consumption of 17 hp.-hr.
[illegible] per lb.
[illegible] to local conditions.

[illegible] various kinds of heating furnaces, [illegible] bushelling, puddling and open- [illegible] being used on continuous-heat- [illegible]. It is also being used as into consideration the knowledge obtained from the experience at the above-mentioned plant, and also that obtained from other installations, it has been found that furnaces can be successfully operated by various methods of applying this fuel. Each type of metallurgical furnace presents different requirements as to the kind

TABLE 3 COSTS OF PULVERIZING COAL PER NET TON AT THE ATLANTIC STEEL COMPANY

Daily Output 80 Tons per Day		
		Cost per ton
Labor		$0.22
Repairs		0.19
Power		0.134
Drier coal		0.0218
Total Cost		$0.5658
Daily Output 90 Tons per Day		
Labor		$0.195
Repairs		0.19
Power		0.134
Drier coal		0.0218
Total Cost		$0.5408
Daily Output 100 Tons per Day		
Labor		$0.176
Repairs		0.19
Power		0.134
Drier coal		0.0218
Total Cost		$0.5218
Total coal pulverized Jan. and Feb. 1919		5275 tons
Power: 17.9 kw-hr. per ton coal pulv. at ¾ cent		$0.134 per ton
Labor: 1 man 16 hr. at $0.40	$ 6.40	
2 men 16 hr. at $0.35	11.20	
Daily cost	$17.60	
(= $0.22 for 80 tons output, $0.195 for 90 tons, $0.176 for 100 tons.)		
Drier coal: Cost of drier fuel 2.8 cents per ton, based on 2.62 per cent moisture. Coal at $5 per ton, 6 lb. evaporated per lb. of coal burned or 8.7 lb. per ton of coal pulv.		
Repairs: Total repairs for Jan. and Feb.	$1412.16	
Credit	409.69	
Charged	$1002.47	(= $0.19 per ton pulv.)

of burners to be used. Probably the greatest recent development in its use has been as a fuel for open-hearth furnaces and boilers. Its application to boilers will be taken up later.

DESCRIPTION OF OPEN-HEARTH FURNACE USING PULVERIZED COAL

25 All open-hearth furnaces using pulverized coal as a fuel are of the reversing type. There has been only one exception to this,

as far as I know, in this country and that exception was at the plant of the American Iron & Steel Manufacturing Co., Lebanon, Pa., where they fired their open-hearth furnaces from one end only. On the other end they installed waste-heat boilers and economizers. As an open-hearth proposition this turned out to be a failure, but as a waste-heat boiler proposition it was a wonderful success. During 1918 they remodeled these furnaces and fired them from both ends.

26 The pulverized coal is delivered into storage bins located at each end of the furnace. On the bottom of these bins are screw feeders, driven by variable-speed motors for supplying the amount of coal desired. This carries the coal by gravity into the burner pipe. These burners are usually a combination of compressed air at from 60 to 80 lb. pressure and fan air at about 8 oz. pressure. In some cases compressed air alone is used as the medium for conveying this coal into the furnaces. The hearth of a pulverized coal open-hearth furnace is practically the same as the hearth of any other open-hearth furnace. The uptakes, slag pockets and checker chambers are entirely different from other furnaces. The uptakes are made as small as possible so as to hold the gases in the furnace as long as possible without blowing, and the slag pockets are made as large as possible so that the gases will have a slow velocity going through them, thereby depositing a large percentage of the heavy particles that are in the outgoing gases. On account of this heavy deposit, removable slag pockets, or very deep stationary pockets, should be used, so as to collect this accumulation over the run of the furnace. Where removable slag pockets are used, they are taken out and cleaned and replaced about every two weeks.

27 Only one checker chamber is needed on each end of the furnace. If the checker chamber is large enough, these chambers should be built up with large tiles and laid in such a manner as to form vertical flues, having openings of at least 6 x 9 in. or better 9 x 11 in. In some cases, no checkers at all are used but the chambers are filled with baffle walls with openings from the outside, so that the accumulation between these baffle walls can be raked out. All passages from slag pockets to stack must be as straight as possible and wherever any bends must be made, some agitating device should be installed at these points. The reversing valves are usually of the mushroom and damper-slide type.

28 Oil and gas are the ideal natural fuels. The increase in the price of oil has made its use as a fuel practically prohibitive. The natural-gas supply is rapidly being exhausted, even in those parts

of our country which have enjoyed its use for years. Consequently we will eliminate these two fuels from our discussion (except in Table 4) and compare the use of pulverized coal with producer gas.

29 The best coal for use in pulverized form in open-hearth practice is a bituminous coal as high in volatile matter as possible, and preferably low in ash. It should never contain below 32 per cent of volatile, nor more than 8 per cent of ash. For open-hearth furnace use it is necessary that the coal be as finely ground as possible and it should be so fine that about 97 per cent will pass through the 100-mesh sieve, preferably 90 to 93 per cent and not less than 85 per cent through the 200-mesh sieve, and from 70 to 75 per cent through the 300-mesh sieve.

30 This very fine pulverization is necessary for quick combustion and for the removal of sulphur in the coal. By this very fine pulverization we attempt to have complete combustion before the flame strikes the bath, thereby burning out the sulphur in the coal to SO_2 gas, which passes up the stack. In order to get this complete combustion before striking the bath, some 6 or 8 ft. are necessary from the end of the burners to the bath.

31 The advantages and disadvantages from the use of pulverized coal, as compared to gas producers, as a fuel for open-hearth furnaces, from observation of its use up to date are as follows:

a Since the coal is of a more even chemical composition all the heat units are consumed in the furnace, while in the case of the gas producer from 18 to 25 per cent of the heat units are lost in the producer itself when converting the coal into gas. This will result in a greater number of heats per week.

b Open-hearth furnaces using powdered fuel operate on a very low fuel consumption equal to the best producer-gas practice, and much better than the average of the older plants in this country; at our plant about 50 per cent less.

c Coal can be pulverized in plants of about 100 tons daily capacity and delivered to the furnace for approximately 50 cents per ton, which is about the same as the costs for gasifying coal in gas producers.

d Although the use of this fuel in metallurgical furnaces has been developed only about 75 per cent, we believe that this development is steadily increasing. In our plant the pulverized-coal open-hearth furnace has been shut down oftener than our producer-gas furnace of the same size, due to checkers and slag pockets filling up with cinders and slag, after about 80 heats; we are gradually overcoming these troubles by decreasing the size of our uptakes and enlarging the slag pockets, thereby holding the gases in the furnace longer and passing them slowly through the large slag pockets, so that the heavy particles can settle, and now only the fine particles are going to our checkers, which particles are being blown off daily by compressed air. By these means, we expect to get much longer life out of our checkers, and consequently longer runs out of the furnace, since the filling up of the checkers has always been the deciding factor in the length of run of the furnace. On account of this continued development, we believe that inside of six months we will show a 25 per cent increase in production over our gas-producer furnaces of the same size.

e Sulphur does not give us any trouble as long as we have good draft and the furnace is working hot, as we are now using over 1 per cent sulphur in our coal and getting good results, although when checkers get clogged up and the furnace begins to blow, due to lack of draft, we have trouble with the bath taking up sulphur. This takes place during the last week's run of the furnace, just before it goes down for repairs.

f The pulverized-coal open-hearth furnace is under complete control of the first helper as to the amount of coal being used at all times, air blast and temperature.

g The flame, using the same coal as on gas producers, is hotter, which allows us to use a greater percentage of scrap per ton of steel, thus reducing the consumption of high-priced pig iron.

h The finished steel is quieter in the molds, due to not being overly oxidized, as the coal coming directly in contact with the bath has a greater reducing action. We feel reasonably certain that the oxidation losses are less with pulverized coal than with producer gas, consequently the per cent of product is greater. All gas-house troubles are eliminated (cleaning fires, burning out flues, etc.) although the pulverizing plant must be given attention as to dryness and fineness of coal.

i Up to date, our refractory costs have been very much greater on our furnace using pulverized coal than on our gas-producer furnaces and was almost twice as great a year or so ago, although I believe on account of the steadily increasing development of the use of this fuel, these refractory costs will be steadily decreased. As a fair sample, we will consider the life of the roof. When originally installed, we got only about 100 heats per roof, but on the last run of this furnace we obtained 232 heats. I believe that, in time, the refractory cost will be nearly as good as gas-producer furnaces, but never any better.

32 Table 4 shows a comparison of fuel costs for all fuels now used on open-hearth furnaces and it will be seen that natural gas is not only the ideal fuel but is the cheapest fuel used. As a direct comparison in our plant, we will compare the figures shown in Table 4 under gas-producer and pulverized fuels, using all cold metal, or items 4 and 7.

33 These figures show 500 lb. coal per ton of ingots using pulverized coal as a fuel, against 739 lb. using producer gas. We have also used $3.40 per net ton as the price of coal for both kinds of fuel, while we invariably use a larger per cent of run of mine at a

TABLE 4 FUEL COSTS FOR OPEN-HEARTH FURNACES

Kind of fuel	Remarks	Amount per ton steel	Rate cost fuel, dollars	Cost of fuel and labor, dollars	Cost per ton steel, dollars
1 Natural gas..........	....	6000 cu. ft.	0.04 per M	....	0.24
2 Natural gas..........	....	6000 cu. ft.	0.12 per M	....	0.72
3 Producer gas..........	Hot metal	510 lb. coal	3.40	3.93	1.00
4 Producer gas[1]........	Cold metal	739 lb. coal	3.40	3.93	1.46
5 Fuel oil..............	....	40 gal.	0.02	....	0.80
6 Tar[2]...............	....	40 gal.	0.025	....	1.00
7 Pulverized coal.......	....	500 lb. coal	3.40	3.90	0.975
8 Electric power.......	....	500 kw-hr.	0.0075	....	3.75

Above includes handling cost
[1] Our plant
[2] Tar is a waste product at some plants and has to be burned.

cheaper price on the pulverized-coal furnace. The cost per ton of pulverizing coal is 50 cents, against 53 cents for gasifying. This gives us a total cost of fuel per ton of ingot steel of 97½ cents using pulverized coal against $1.46 using producer gas.

BOILERS

34 Many engineers who attempted to burn coal in pulverized form obtained unsatisfactory results, and concluded it was "impossible." In many of the earlier trials to burn pulverized coal under boilers the usual method was to install coal-feeding devices of some kind in the furnace as it stood, with the result that the fire bricks melted down and the tubes were plastered with unconsumed carbon, ashes and soot. So destructive were the results that we can

hardly blame those making the tests, from arriving at the conclusion they did. How close some were to success was not fully realized. Conditions were not ripe. Today, results are being obtained that are of sufficient importance to warrant careful investigation and consideration.

35 For the proper combustion of coal under boilers, there are five main points which must be given serious consideration, otherwise the burning of this fuel will not be a success. These five points are:

a Coal fineness
b Size of combustion chamber
c Necessary air opening
d Proper damper regulation
e Clean tubes

36 *Coal Fineness.* The pulverized coal should run about 96 per cent through the 100-mesh screen and about 85 per cent through the 200-mesh screen. If it runs below 80 per cent through the 200-mesh screen, we notice particles of carbon flying through the air inside of the combustion chamber, and these particles are not completely burned when the gases reach the tubes, and will then deposit themselves on the tubes. Also, these heavy particles in the pulverized-coal mixture will sometimes settle on the bottom of the combustion chamber and will soon build up in the shape of stalactites. This accumulation will continue to build until the bottom of the combustion chamber continues to be raised, until it comes in contact with the flame. These built-up particles will then fuse into a solid mass, which, in a very short time, will cause a shut down of the boiler to dig this fused mass out. It is, therefore, necessary, for the successful burning of this pulverized fuel under boilers, to have this coal as finely pulverized as possible.

37 *Size of Combustion Chamber.* Before installing, or considering the use of pulverized coal under boilers, we must first make a study of our boiler-house installation and know at exactly what rating we wish to operate these boilers, considering any peak load which may develop. We must then design our combustion chamber large enough to take care of the maximum loads which will be developed from our boilers at any time. After this maximum rating has been determined, we then figure our combustion chamber of a cubical capacity equivalent to approximately 50 cu. ft. per lb. of coal burned per min., or approximately 2½ cu. ft. per hp. developed by boiler.

If we decide to run this boiler at 150 per cent of this rating and design our combustion chamber accordingly, the efficiency will not be decreased perceptibly if the boiler is run under this 150 per cent rating, but if we develop over a 150 per cent of the rating, we run into serious difficulties. We have to admit more coal and air to the boiler to develop the greater rating, consequently we need more combustion space, but not having this combustion space, the flames impinge on the brick work and cut it away very rapidly. Also, combustion is not complete at the time the gases strike the bottom row of tubes and consequently the gases will pass up the stack unburned. Efficiency is then decreased. If the gases are not completely burned by the time they reach the first row of tubes, they will not burn later on in the boiler. The size of the combustion chamber should also be so designed that the velocity of the gases should not pass through this combustion chamber at a speed of more than 6 ft. per sec. The mixture of air and coal entering the combustion chamber, as stated above, should be at as low a pressure as is possible to bring this mixture in in suspension; or, in other words, breathe it in.

38 *Proper Air Openings.* The pressure at which the pulverized coal is admitted to the furnace is as low a pressure as can be used to carry this fine coal in suspension, and is, I should say, about half an ounce pressure at the nozzle. In some installations the coal falls by gravity from the variable-speed screw conveyor, located on the bottom of the pulverized coal bin, into a fan air line which carries it into the furnace and also supplies the necessary air for combustion. Some few openings are placed in the front wall of the boiler to give any additional air which may be needed, and a few may also be placed on the side walls of the boiler to protect the brickwork at times.

39 In other installations the amount of air necessary to convey the coal into the furnace varies, according to the rate at which the boiler is being operated. The balance of the air to burn the coal properly is admitted through adjustable air openings in the front, sides and bottom of the combustion chamber. These openings are made adjustable and are placed on all sides of the combustion chamber to take care of the various grades of coal which may be used in the boiler plant. By properly observing the combustion in this chamber, by a little experience the fireman knows at exactly what points to give more, or less air needed for combustion.

40 *Damper Regulation.* In order to give the proper velocities

of gases passing through the combustion chamber, it is necessary that we have very accurate damper regulation to take care of the various load conditions which the boiler is to supply. The damper should be so regulated that we should practically have a balanced draft inside of the combustion chamber and only a slight vacuum in the first pass, while at the damper itself we do not want more than 0.10 to 0.15 in. If we have more vacuum than this, it pulls our gases through the combustion chamber too fast, causing them to be unburned before reaching the first row of tubes and will then build up very fast on the outside of these tubes. This very small draft needed at the base of the stack will allow us to operate boilers using pulverized coal with stacks of about 30 to 35 ft. in height.

41 *Clean Tubes.* In order to get the maximum evaporation from any boiler it is necessary that the tubes be kept clean, both inside and outside. The keeping of the tubes clean inside is a question of the proper quality of water and has nothing to do with pulverized coal. The keeping clean of the outside of the tubes is very necessary with the use of pulverized coal as a fuel, and they should be blown by means of mechanical soot blowers at least every six hours, and oftener if it is necessary. Also, once every 24 hours by means of a hand-lance steam jet we should blow the bottom of the first row of tubes. These are the tubes in which the gases come into contact first after leaving the combustion chamber. This material can be blown off easily if the combustion chamber has been properly constructed and if they are blown regularly as needed. But if they are not blown regularly this material builds up and accumulates very fast and in time will become fused and cannot be blown off.

42 Another item which might have been included in the five main points above, and might have been called the sixth point, is the removal of ash which deposits at the bottom of the combustion chamber. As spoken of above, this ash should be removed at regular intervals, which intervals will be determined by the amount of ash in the original coal. If we do not remove this ash regularly, it will build up until it comes in contact with the flame, when it becomes fused and has to be dug out. But if removed at regular intervals, it can be easily raked out with the ordinary boiler-room ash rake and will not consume more than half an hour per 24 hours, and will not interfere with the operation of the boiler while this is being done.

43 The following are some advantages of pulverized coal as a fuel for boilers over stokers:

a Much wider variation in the quality of the coal usable is obtained when burning coal in pulverized form. Practically any and all grades of coal can be burned in this form with economy. No stoker will satisfactorily handle all grades of coal. Therefore the use of pulverized coal will largely overcome most troubles due to poor coal, and it is particularly desirable for this reason alone.

b The ability to take care of peak loads almost instantaneously. In other words, a pulverized-coal burning system is much more flexible than a stoker installation. Its flexibility approaches that of oil or natural gas.

c The amount of coal that can be burned per sq. ft. of grate surface on stokers is limited so that for increased capacity the boiler setting must be spread out to cover more area. When using pulverized coal, this condition does not exist, for proper furnace conditions can be obtained by increasing the height of the boiler setting or the depth of the combustion chamber.

d By throwing a switch the entire firing operation ceases; an advantage in case of accident or emergency.

e Ash is in much better condition to handle. The ash is in the form of a dust or slag depending upon its melting point. This helps to maintain constant furnace temperature as there are no interruptions to firing conditions on account of cleaning fires.

f Since there are no grates used when the fuel is burned in pulverized form, we experience no clinkering of grates, as in the case of stokers, particularly after operating at maximum rating.

g Pulverized coal is fired dry, containing less than 1 per cent of free moisture, whereas coal burned on stokers may vary anywhere from 1 to 10 per cent, of free moisture as fired.

h Considerable less excess air is necessary for complete combustion. This item is of the utmost importance when making comparisons. Less excess air means less power for furnishing air supply particularly where forced draft is used. With less excess air the stack losses are less. Lower grades of coal fired on stokers require more excess air as it is quite difficult for the oxygen to get in close contact with the combustible. An air supply sufficient to furnish all the air for combustion should be available, although at times only 50 per cent of the air is necessary to be injected into the furnace with the coal, the balance being supplied by the induction action of the burner or drawn in by the stack draft through the various adjustable openings in front and sides of combustion chamber. The air going into the furnace should be under control to permit close regulation under all conditions of firing. Less draft is required for pulverized-coal-fired furnaces.

i All the combustible in the coal is consumed when it is burned in pulverized form, providing the furnace capacity is not exceeded. None of the combustible goes out into the ash pile and therefore fires are eliminated in the ash pile.

j There is less erosion from sulphur on the boilers due to less moisture in the coal as fired, therefore high-sulphur coals can be burned more readily and without serious results.

k With furnace properly proportioned and with properly designed burning equipment smokeless operation may be maintained indefinitely. This is due to complete combustion of all the particles of coal before coming in contact with the cold surface of the tubes of the boiler.

44 The following few points must be kept in mind for the successful burning of pulverized coal under boilers:

a A boiler furnace using pulverized coal should have as few burners as is possible consistent with good regulation. The burners must be proportioned for the maximum rating of the boilers, and they must be adjustable. Simplicity of design is desirable. It has been found much more desirable to introduce coal into the furnace as far away from the side walls as possible so that the rapid continuous expansion of the gases will not develop high velocities in close contact with the furnace refractories. Furnaces under boilers should be proportioned so that the velocity of the gases should not be excessive, particularly at the smallest cross-sectional area of the furnace. Vertical baffles should replace all horizontal baffles.

b A boiler of any size can be fired successfully with pulverized coal. Various designs and makes of boilers can be readily arranged for pulverized-coal firing, but those containing the smaller percentage of space for the lodgment of ash are preferable.

c Feeders for regulating the flow of pulverized coal to the furnace must be designed so that at all times the variation in quantity will be directly proportional to the speed of the screw and no flooding allowed. The speed of the feeder should be so regulated that operating at its maximum r.p.m. the supply of pulverized coal to the furnace will not exceed the capacity of the furnace. Soot blowers should be installed in settings where pulverized coal is used.

d The equipment for using pulverized coal is standard for any grade of coal so far as handling, preparing and delivering to the furnace is concerned. Only a slight change is necessary in the furnace to take care of coals of very low volatile content, such as anthracite, culm and coke breeze, and increased drying capacity is desirable when lignite coal is used. With stokers this is not the case as the varying quality of coals require different type stokers to obtain highest efficiency.

e The labor required to operate a pulverized-coal installation may be of higher class but the number of men required will be less in the larger installations than that required for a stoker installation, thereby affecting a saving in the labor charge in favor of pulverized coal.

45 The following is a report of a test made on a 468-hp. boiler using pulverized coal as fuel. This installation is noteworthy not only by reason of the high efficiency obtained, but also because of the fact that it has made clear some of the conditions necessary for the successful operation of boilers utilizing powdered fuel.

46 When the boiler was first put into operation, a number of undesirable conditions resulted. An insufficient air supply caused high-furnace temperatures resulting in fusion of the ash particles and a consequent accumulation of slag between the tubes, on the furnace walls and in the ashpit. The removal of the molten slag presented considerable difficulty. It was also found that the combustion chamber was of insufficient size. High gas velocities resulting from insufficient air in the chamber tended toward destruction of the refractory surfaces of the furnace.

47 A new furnace was, therefore, designed. The combustion chamber was enlarged and a regulated air supply was provided for by means of a number of auxiliary air openings equipped with dampers. The accumulation of slag in the pit was prevented by raising the point of admission of the fuel into the furnace. As a result the flame path has been raised above the base of the pit, hence particles of ash dropping from the flame are not fused. The ash, therefore, can be drawn from the pit in the form of a powder and small slugs of slag. Analysis has shown that the ash contains practically no carbon.

48 Having established satisfactory furnace-operating conditions, a series of efficiency and capacity tests were conducted preliminary to proving the contract guarantees. The brickwork was then given a thorough trial by carrying the boiler at a continuous rating of 180 per cent over a period of several days. On August 12 and 13 a final efficiency test, the results of which are given below, was run. The boiler is a three-pass water-tube boiler, equipped with a superheater.

TABLE 5 LOG OF TEST OF A PULVERIZED-FUEL-BURNING STATIONARY BOILER. DATE AUGUST 12-13, 1918

Make of boiler				Edge Moor
Rated hp				468
Heating surface, sq. ft.				4685
Time fired or test started				11.15 a.m. 8/12/18
Time fire out or test finished				11.15 a.m. 8/13/18
Duration of test				24 hr.
		Maximum	Minimum	Average
Temperature of boiler room (deg. fahr.)		99	85	93.3
Temperature of feedwater		168	135	157.2
Temperature of steam (deg. fahr.)		477	427	448.7
Barometer in. of mercury		29.35	29.20	29.25
Temperature of flue gases (deg. fahr.)		515	455	495.3
Average boiler pressure, lb.				167.0
Atmospheric pressure, lb.				14.4
Temperature of steam, deg. fahr.				373.8
Superheat, deg. fahr.				74.9
Safety valve set for, lb.				175
Fuel fired per hr., lb.				1,990.6
Total fuel, lb.				47,775
Total water, lb.				393,168
Water apparently evaporated per hr., lb.				16,393.0
Water apparently evaporated per lb. of coal, lb.				8.23
Factor of evaporation				1.150
Water evaporated from and at 212 deg. fahr. per lb. of coal, lb.				9.47
		Maximum	Minimum	Average
Carbon dioxide (CO_2) per cent		15.4	12.2	13.85
Oxygen (O) per cent		5.6	3.2	4.38
Carbon monoxide (CO)				None
Fuel used				Bituminous screenings
Fuel analysis	No. 1	No. 2	No. 3	Average
Amount of coal represented by each sample, lb.	19,775	20,000	8,000	..
Per cent of total	41.3	41.1	16.9	..
Moisture (per cent)	10.3	11.0	9.7	10.49
Volatile (per cent)	33.81	36.96	38.77	35.96
Fixed carbon (per cent)	50.43	49.13	48.29	49.53
Ash (per cent)	14.36	13.91	12.94	13.93
Sulphur (per cent)	1.90	2.06	2.12	2.04
B.t.u. as received	10,600	10,763	11,263	10,779
B.t.u. dry	11,817	12,093	12,473	12,045
Vacuum in burner, in.				0.000
Vacuum under primary arch, in.				0.000
Vacuum in combustion chamber, in.				0.000
Vacuum in first pass, in.				0.000
Vacuum in second pass, in.				0.0057
Vacuum in breeching, in.				0.09
Feeder speed, r.p.m.				(No. 1), 53.6; (No. 2); 50.7
Coal per rev. of screw, lb.				0.318
Accumulation of slag on tubes				None
Flues blown during test				5 times
Operation of furnace				Very satisfactory
Pulsation				None
Condition of smoke				Light
Heat effect on brick				None
Back lash of flame in burner				None
Pounds of steam per hr. from and at 212 deg. fahr.				18,842.6
Horsepower				546.2
Per cent of rating				116.7
Boiler efficiency, per cent				85.22

Memoranda — Fuel-preparation deduction:[1]	
Coal used in drier, lb.	1,140
Motor operation,	449.3 kw-hr.
Coal equivalent at 3 lb. per kw-hr., lb.	1,348
Total deduction, lb.	2,488
Resulting net efficiency, per cent	81.1

[1] No deduction made for stand-by losses in drier.

49 At this same plant are other boilers fired by one of the most efficient types of underfeed stokers. A comparison is made between results of above test and tests made on the stoker-fired boilers.

PULVERIZED COAL VS. MECHANICAL STOKERS

50 Under this heading fuel-preparation costs will first be considered. In the case of powdered coal this can be classed under three general divisions:

a The cost of crushing the coal. This expense is the same for pulverized-coal equipment as for stokers.

b The cost of drying and pulverizing the coal. Although no cost records are available at present, it is estimated that 32 cents per ton will cover this preparation cost on a 200-ton-per-24-hr. plant using bituminous coal containing about 12 per cent moisture.

c The maintenance costs of the drying and pulverizing plant. This unit has not been determined from actual experience; however, it is estimated that 3 cents per ton will cover the maintenance. In stoker practice the maintenance cost per ton of fuel fired is close to 5 cents per ton.

51 Summarizing the above facts it is evident that, with fuel at $5 per ton, the gross efficiency shown by the pulverized-fuel boilers will have to exceed that shown by the mechanical-stoker-fired boilers by 6 per cent in order to offset coal-preparation costs. A 6 per cent deduction from a gross efficiency of 85.22 per cent results in a net efficiency of 79.22 per cent for the powdered-coal burner. In stoker practice the maximum attainable gross efficiency at any of our plants has been 80.54 per cent. Deducting the 2.5 per cent for auxiliary uses, the resulting net efficiency is 78.04 per cent, which is lower by 1.18 per cent than the figure obtained in pulverized-fuel practice.

52 Other advantages resulting from the use of pulverized fuel are summarized herewith:

a Continuous boiler operation at a uniform rating as well as a constant efficiency is made possible. At no time is there a loss in capacity due to the clinkering of coal on the grates or the cleaning of fires, as is the case in stoker practice.

b Heavy overloads can be taken on or dropped off in a very brief time through adjustment of the coal feeders and the furnace drafts.

c From 97 to 98 per cent of the combustible in the coal is utilized, regardless of the quality of the fuel.

d The ash-handling costs are reduced to a minimum due to the reduced volume.

e The banking conditions when operating with pulverized coal are somewhat different from those obtained in stoker practice. By stopping the fuel supply and closing up all dampers and auxiliary air inlets a boiler can be held up to pressure for about 10 hours. The furnace brick work having been heated to incandescence during operation gives off a radiant heat which is absorbed by the boiler rather than being sent out through the stack. The ease of controlling the fuel, feed and drafts, the ability to take on heavy overloads in a brief time, the thorough combustion of the coal and the uniform high efficiency obtainable under normal operating make pulverized coal a most satisfactory form of fuel for central station uses.

53 The full story of maintenance expense is only partly known as yet, however. Indications are that no unusual difficulties will be met. The cost of fuel preparation and labor for operating a boiler room fully equipped with pulverized-coal-burning boilers will be a question for the engineer to decide for himself according to his particular conditions. If properly installed with respect to capacity of storage, size of drier and pulverizers, and on a sufficient number of boilers to properly and fully employ the minimum number of men, the pulverized-fuel installation will undoubtedly be more advantageous. The main item that must be borne in mind by engineers is that the ease with which a high efficiency is obtained and the constant nature of that efficiency, as compared to the lack of constancy of efficiency in a stoker-fired boiler, unless very closely supervised, is the one factor about the burning of pulverized fuel which justifies its use. There is no doubt that with a well-equipped plant burning pulverized fuel, having all the necessary recording and indicating instruments to guide the operators in maintaining the proper conditions, a lower cost of generating steam will be possible than has heretofore been the case in any type of equipment.

No. 1702

PULVERIZED COAL FOR STATIONARY BOILERS

By Fred'k A. Scheffler, New York, N.Y.

and

H. G. Barnhurst, Allentown, Pa.

Members of the Society

It is the purpose of this paper to present those facts which the authors believe indicate the coming general adoption of pulverized coal as a fuel for boilers. Since stoker firing is the most efficient method when solid fuels are used, a comparison of stoker and pulverized-fuel plants is given, with particular reference to reliability, cost, adaptability and efficiency. The cost of pulverizing coal and the cost of stoker operation are discussed in detail and tables given showing results of tests on pulverized-fuel plants and data regarding boiler installations using pulverized coal as a fuel.

THE authors, being of the opinion that best present-day practice of firing boilers in power plants of moderate and large capacity has attained the maximum efficiency that might be expected, believe that if we are still further to conserve the coal fields of the country and in addition reduce the operating costs in the use of fuel and labor, some other method of firing boilers by coal must be adopted.

2 The principal increased costs of power-plant operation during the past three or four years have been due to the increased cost of coal and labor, which in many cases have alone added 100 per cent to the cost of operating the boiler plant. It is also safe to say that the improvements in the turbo-generators, condensing equipment and large-size steam units now being used with increasing boiler pressure and high superheat may possibly not permit of a still lower water rate. We must therefore turn to the boiler-furnace equipment for further reduction in operating costs, and the authors

---

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

respectfully submit the present discussion on the use of pulverized or powdered coal with a view of demonstrating a practical commercial solution of this highly important question.

3 Although a very great interest has been aroused throughout this country and abroad in the adaptation of pulverized coal to boiler furnaces, it is remarkable how little is known, in a practical way, of what is actually being done, how it is accomplished, and what results have been obtained. It is hoped, therefore, that the data and illustrations presented will prove of interest and will call forth discussion which will serve to develop further this study of the best method of reducing power-operating costs.

COMPARISON OF STOKER WITH PULVERIZED-FUEL PLANTS

4 Since the purpose of this paper is to present the facts which indicate the coming general adoption of pulverized coal as a fuel for boilers, the discussion is presented in the form of a comparison with stoker firing, the latter being the most efficient method in general use for burning solid fuel under boilers.

5 The ultimate adoption of a new method depends entirely on its overall commercial efficiency. In the generation of power, overall efficiency may be considered as composed of the following factors: reliability, cost and adaptability. A method may acquire a wide field if it shows improvement in any one or two of these points. Improvement in all three points leads to the general superseding of other methods.

6 *Reliability.* Let us compare the reliability of a pulverized-coal installation with that of stokers. This factor depends on two items: apparatus for preparing and presenting the fuel for combustion, and continuity of operation of the furnace itself. In a stoker installation the first of these includes the stoker itself. Neglecting the inherent defects of any system that presents a metal mechanism to the action of high temperatures, it may be admitted readily that the stoker system is satisfactorily reliable, with respect to its apparatus, for preparing and presenting the coal for combustion.

7 The corresponding mechanisms for pulverized fuel are equally reliable. This fact is proved by their widespread use for years in the cement industry and more recently in an ever-increasing variety of industries. It should be recognized that these mechanisms are not innovations, but are the result of years of development under operating conditions. Proper design of equipment by engi-

neers of standing who are specialists in this line has made negligible the danger of dust explosions, the occasional occurrences of which in years past have furnished ammunition to the opponents of pulverized fuel.

8 The second condition for reliability is the continuity of operation of the furnace. Here again we find an apparent balance between stoker and pulverized-fuel installations during operation. The advantage lies with pulverized fuel, however, for several reasons. The mechanism is altogether outside the furnace, hence cleaning and adjustment and the making of the few repairs required need not interrupt the operation of the boiler. In case of sudden necessity the fire may be ignited and quickly brought to full intensity, or it may be extinguished almost instantly. Greater uniformity of flame and temperature is conducive to longer life of the furnace lining in a properly designed furnace, and to the minimum variation in furnace efficiency. Finally, the pulverized-fuel installation relieves the power plant from dependence upon the availability of a certain grade of coal. Stokers will not handle all grades of coal.

9 *Cost.* This second factor refers to the cost per B.t.u. delivered to the boiler. The various items entering into this cost by the stoker system comprise power, repairs and maintenance, labor, interest on investment, depreciation, insurance and taxes.

10 With pulverized-coal equipment the cost of fuel for the drier should be added to the preceding items. A moment's consideration will show that this item must be taken care of in the furnace of a stoker-fired boiler and that it is clearly cheaper to remove the excess moisture content from the coal in a drier, from which the gases leave at very low temperature, than in the furnace itself, where the evaporation of the moisture damps the fire, increases the content of inert gases and at the same time carries off a very perceptible amount of heat.

11 Returning to the balanced cost items, it appears that these show a saving in favor of pulverized fuel in a large power plant and for the stoker in a small power plant. The figures in question are discussed further on in detail. It should be noted that when central pulverizing plants are built, they will relieve small power plants of the necessity for maintaining pulverizing equipment and make pulverized fuel considerably cheaper than stoker-fed fuel, regardless of the size of installation. This feature is already being carried out successfully.

12 The final factor of the cost, furnace efficiency, which gov-

erns all the others, results in all respects to the advantage of pulverized fuel for the following reasons:

13 First: The fuel enters the combustion chamber in a finely divided state, being introduced with air at low pressure, and is approximately perfectly mixed with the air for theoretically perfect combustion. Therefore no excess air is required for complete combustion. The units of heat taken up by heating excess air reduce the combined boiler and furnace efficiency. It is impossible to get a uniform fuel bed on a stoker or a grate and, therefore, impossible to approach complete combustion without introducing excess air. Should it be desired for other reasons to introduce excess air with pulverized fuel, it can be done in exact amounts, evenly distributed, without affecting the uniform nature of the flame and flue gases. This uniformity, which cannot be obtained in either grate or stoker installations, means maximum efficiency in all parts of the furnace and a maximum rate of heat transference to the boiler throughout its exposed area. It also means that flue-gas analysis gives an accurate determination of conditions in the furnace, and that control of coal delivered and air supply can be adjusted with great accuracy.

14 Second: In pulverized form all of the combustible is burned, a consummation certainly impossible in lump-coal firing by either hand or stoker. It is not unusual to find 20 to 30 per cent of carbon in ash refuse from grate- or stoker-fired boilers.

15 Third: With pulverized fuel there are no standby losses with change of load or when shutting down, such as banked fires, etc.

16 Fourth: With properly designed pulverized-fuel apparatus nothing of a mechanical nature takes place in the furnace. In stoker and grate firing not only is the mixing with the air done in the furnace, but the presentation of fresh surfaces of combustible to the air supply must take place by the removal of the ash and its discharge through the grate bars, or the pressure must be great enough to force the air supply through the ash bed.

17 *Adaptability.* Let us now consider the third factor of overall efficiency, which is adaptability. Pulverized fuel is here preëminent. The primary feature is the possibility of burning all grades of fuel without affecting the efficiency of the furnace. To burn anthracite and very low grades of fuel requires a furnace allowing a return flow of the flame past the incoming flame, to heat up the incoming fuel, and in a furnace of this type fuel containing over 50 per cent ash has been burned with high efficiency. The stoker is very much restricted in comparison.

18 The flexibility in the use of pulverized fuel is perfect, and the fire may be instantly adjusted to suit any condition of overload or lower load, including the cutting in and out of the boilers. The paramount importance of this feature and the utter impossibility of approaching it with stoker or grate firing is readily evident. Furthermore, the operation and the determination of conditions for complete combustion may be made automatic, the result being a smokeless and sootless boiler plant, which is essential in modern cities.

19 *Furnace Design.* A few words on the design of furnaces for pulverized fuel may be of interest. The primary requisite for good results is to maintain low velocities in the furnace. The combustion is no less perfect with high velocities, but this will result in damage to the linings and in their erosion. A furnace cubical in shape usually gives the most satisfactory results.

20 The burners should inject the coal under low pressure and should permit of varying the density of the mixture in the burner itself. Their location and number will depend upon the size of the boiler and rating required, and also may be varied to suit the grade of fuel. High boiler ratings such as are used in modern boiler practice can be obtained when desired, and such overratings should be predetermined and the furnace volume designed accordingly.

21 It will be noted that pulverized coal behaves more nearly like liquid and gas fuels than it does like lump coal and that it is in the ideal state for burning with the highest possible efficiency. It has been shown that it is superior to lump coal as regards all three factors of overall efficiency and these statements are susceptible of proof upon investigation. The novelty of the pulverized-fuel plant is rapidly beginning to disappear, and on account of the fact that all obtainable coals are apparently becoming more inferior in quality, the interest in the use of pulverized fuel is very general throughout the United States and other countries.

22 In Table 1 will be found an itemized statement of the costs of pulverizing coal, and in Pars. 34 and 35 some statements as to the cost of stoker operation for comparison purposes. Table 2 gives a list of boiler installations using pulverized coal and Table 3 reports of preliminary tests made on some of the pulverized-fuel installations now in operation. While these do not show the maximum efficiency to be expected with the further development of the art, they nevertheless indicate that the inherent difficulties have been solved and that at the present moment pulverized fuel is in a posi-

tion to compete advantageously with any other method of burning solid fuel under boilers.

COMPARISON ON AN EFFICIENCY BASIS

23 One of the most prominent engineers in this country, a member of the Society, has stated that the combined boiler and furnace efficiency by the month, day in and day out, of a modern stoker-fired power plant with the best average plant operation is

TABLE 1 COST OF DELIVERING PULVERIZED FUEL TO BOILERS

	100-ton plant, dollars per net ton	1000-ton plant, dollars per net ton
Power at ¾ cent per kw-hr. and 17 kw-hr. per net ton	$0.1275	$0.1275
Labor at 40 cents per hr.	0.14	0.04
Drier coal at $5 per net ton delivered	0.06	0.06
Repairs:	0.07	0.07
Total actual cost of pulverizing per net ton	0.3975	0.2975
Interest at 6 per cent	0.105	0.039
Depreciation	0.12	0.04
Taxes and insurance	0.035	0.013
Total cost per net ton	0.6575	0.3895

not better than from 63 to 65 per cent, although a carefully conducted test on one boiler and furnace might show during several hours' run 75 per cent efficiency. This statement has been confirmed by other engineers.

24 The results with pulverized fuel would be totally different. There is no apparent reason why a combined furnace and boiler efficiency of 75 per cent, and even higher, could not be maintained throughout the year, as the operation of the plant would be practically equivalent to that of a fuel-oil installation, in which stand-by losses, banked fires, etc., are almost entirely eliminated. Unquestionably there should be a saving, under these circumstances, of 12 to 15 per cent of the total coal consumption in favor of pulverized coal, and this reduction, on a basis of even a 2000-boiler-hp. plant, will show a very fair return on the investment, neglecting the fact that a lower-grade and cheaper coal could be used.

COST OF PULVERIZING COAL

25 The cost of pulverizing the coal is of prime importance as low costs are essential for success and are achieved when the quantity used per day of 24 hours exceeds 100 tons. The cost of pulverizing is made up of a number of items as follows:

Power	Interest
Repairs and maintenance	Depreciation
Coal for drying	Insurance and taxes
Labor	

26 *Power.* The power required in an up-to-date pulverized-coal plant is from 12 to 13 kw-hr. per net ton of coal crushed, dried and pulverized. The additional power required for transferring the coal to the point of use and feeding it to the boilers will vary considerably, depending upon the distance transported, the size and number of the boilers, and the conditions under which they operate. The power required for this latter purpose varies between 4 and 6 kw-hr. per net ton, so that the total power for the entire process from the track and storage delivered to the boilers is 17 or 18 kw-hr. per net ton. In the following paragraphs the cost of power has been assumed at $\frac{3}{4}$ cent per kw-hr.

27 *Repairs.* The item of repairs, including material, labor and general upkeep of the plant or maintenance, for the entire pulverizing plant and burning equipment will vary from 7 to 10 cents per net ton of coal handled. The figures depend upon local conditions, and the size and general arrangement of the entire installation.

28 *Drier Fuel.* The item of coal for drying depends directly upon the percentage of moisture and upon the price of coal. Ordinarily only from 1 to $1\frac{1}{4}$ per cent of the total amount of coal used is required for drying. Assuming coal to have an average of 7 per cent moisture as received and the cost to be \$2.50 per net ton, the cost per net ton of drying the coal will be 3 cents. At \$5 per net ton the cost of the drier coal will be 6 cents.

29 *Labor.* This item is the greatest variable in connection with the pulverizing of coal, due to the increased output that can be obtained in larger plants per man employed. It is also subject to local rates of wages. For example, assuming labor at 40 cents per hour, a plant of 100 tons daily capacity, properly designed and equipped, will require approximately 34 labor hours to prepare the fuel and deliver it to the conveyors, whereas in a plant having a daily

TABLE 2 BOILER INSTALLATIONS USING PULVERIZED COAL

Date of installation	Name of company	Location	No. of boilers	Horsepower rating and make of boilers	Furnace design, per cent of rating	Coal used
Aug. 1916	M. K. & T. R.R.	Parsons, Kan.	8	250 O'Brien	125–150	McAllister Cherokee slack, Kan. semi-anthracite, Texas lignites, San Bois coal, Oklahoma
Nov. 1916	American Locomotive Co.	Schenectady, N. Y.	1	300 Franklin	150	
June 1918	U. S. Verde Extension Mining Co.	Verde, Ariz.	2	439 Stirling	150	Gallup and semi-lignite
Feb. 1918	Ash Grove Lime & Cement Co.	Chanute, Kan.	1	371 Heine	150	Various grades of Kansas coals
June 1918	Garfield Smelting Co.	Garfield, Utah.	2	371 Stirling	150	Wyoming lignite, Wyopa, Wyoming lignite. Keystone, Utah, bituminous from various mines
Nov. 1918	Puget Sound Light and Power Co.	Seattle, Wash.	10	4–300 B. & W. 2–600 B. & W. 3–400 B. & W. 1–500 B. & W.	150	Renton buckwheat, Washington bituminous lignite and sub-bituminous
Nov. 1917	Siser Forge Co.[1]	Buffalo, N. Y.	5	250 Rust	125	Pittsburgh and Pennsylvania
Mar. 1919	British Columbia Sugar Refinery Co.	Vancouver, B. C.	13	2–504 Badenhausen 2–250 B. & W. 9–110 HRT	150	Vancouver, B. C., bituminous and lignite
July 1918	Milwaukee Electric Railway & Lighting Co.	Milwaukee, Wis.	5	468 Edge Moor	150	Indiana and Illinois bituminous, Pittsburgh and Youghiogheny

TABLE 2 BOILER INSTALLATIONS USING PULVERIZED COAL (*Continued*)

Mar. 1919	Allegheny Steel Co.	Allegheny, Pa.	1	333 Wickes	...	Pittsburgh coals
June 1919	Inland Steel Co.	Chicago Heights, Ill.	1	250 Heine	150	Illinois bituminous
June 1918	Pacific Coast Coal Co.	Seattle, Wash.	10	150 HRT	...	Renton buckwheat
			8	100 HRT	...	Washington bituminous, lignite and sub-bituminous
Nov. 1918	Susquehanna Collieries Co.	Lykens, Pa.	1	250 B. & W.	200	All grades of anthracite washery culm, mine dirt, No. 3 buckwheat, Lykens slush, Lytle slush
June 1919	Lytle Coal Co.	Lytle, Pa.	6	333 B. & W.	200	All grades of anthracite, washery culm, Lytle slush
May 1919	Garfield Smelting Co. (2d installation)	Garfield, Utah	4	371 Stirling	150	Wyoming lignite, Wyopa, Wyoming lignite, Keystone, Utah, bituminous from various mines
Sept. 1918	L. S. Smith Bldg.	Seattle, Wash.	2	1–300 B. & W. 1–200 B. & W.	... ...	Obtain and use coal from Pacific Coast Coal Co.
Sept. 1918	Crystal Natatorium	Seattle, Wash.	2	72 x 18 HRT	...	Obtain and use coal from Pacific Coast Coal Co.
Sept. 1918	Crystal Natatorium	Seattle, Wash.	2	72 x 18 HRT	...	Obtain and use coal from Pacific Coast Coal Co.
Sept. 1918	Pacific Coast Coal Co.	Seattle, Wash.	12	2–250 Wickes 2–125 Ames (72 x 16) 6–125 Chandler & Taylor 2–125 Casey-Hedges	...	Obtain and use coal from Pacific Coast Coal Co.

[1] Also use some waste heat from pulverised-coal-fired furnaces.

capacity of 1000 tons, approximately 115 labor hours are required. Therefore the labor cost would be 14 cents per net ton in a 100-ton plant, only 4 cents per net ton in a 1000-ton plant, and as low as 2½ cents per net ton in a plant of 5000 tons daily capacity.

30 *Interest.* The interest item is based on 6 per cent of the entire investment, and the cost of the pulverized-coal plant and

TABLE 3 REPORT OF PRELIMINARY TESTS MADE ON PULVERIZED-FUEL PLANTS

Date of test	Location of plant	Duration, hr.	Coal used	Efficiency maintained, per cent	B.t.u. per lb. of coal as fired	Ash, per cent	Rating, per cent
Apr. 16, 1917	Seattle, Wash.	14.5	Renton buckwheat	77	10,000	11.60	122
Dec. 4, 1917	Chanute, Kan.	5	Kansas bituminous	72	11,996	17.7	125
Dec. 12, 1917	Chanute, Kan.	5	Kansas bituminous	83.94	12,500	18.25	125
Jan. 28, 1918	Chanute, Kan.	(25 days)	Kansas bituminous	78.1	11,435	.....	100
Apr. 26, 1918	Parsons, Kan.	6	Kansas bituminous	80.3	12,900	17.49	....
Apr. 28, 1918	Parsons, Kan.	6	Kansas bituminous	80.9	12,289	17.49	130.8
June 14, 1918	Milwaukee, Wis.	12	Illinois and Indiana screenings	83.3	10,897	15.89	117.7
Nov. 5, 1918	Lykens, Pa.	10	Lykens No. 3 buckwheat anthracite	84.2	12,530	16.92	135
Nov. 15, 1918	Lykens, Pa.	5	Lykens slush buckwheat anthracite	81.2	13,653	11.09	142
Nov. 22, 1918	Lykens, Pa.	5	Lykens slush buckwheat anthracite	85	12,753	18.04	146
Nov. 23, 1918	Lykens, Pa.	5	No. 3 buckwheat anthracite	72.7	12,530	16.91	115
Dec. 2, 1918	Lykens, Pa.	5	Lytle slush anthracite	75.3	12,753	23.92	188
Feb. 1, 1919	Seattle, Wash.	24	Issaquah screenings	78.95	11,660	14.31	126
Feb. 2, 1919	Lykens, Pa.	4	No. 3 buckwheat anthracite	78.9	13,067	14.02	177
Apr. 7, 1919	Vancouver, B. C.	4	Nanaimo slack	83.3	9,364	28.4	125
Apr. 17, 1919	Vancouver, B. C.	5	Nanaimo slack	77.1	10,050	24.3	160
Feb. 3, 1919	Lykens, Pa.	5.5	No. 3 buckwheat anthracite	78.9	12,530	14.00	...
Sept. 24, 1918	Verde, Ariz.	(6 days)	Gallup, New Mexico	79.5	10,680	14.31	155

burning equipment will of course vary considerably with the conditions under which the plant is installed. Roughly speaking, however, the actual investment will vary from $12.80 per kw. output in a 5000-kw. plant down to $4.80 per kw. in a 50,000-kw. plant and $4.12 in a 100,000-kw. plant (assuming a turbo-generator water rate of 16 lb. and continuous boiler and furnace efficiency of 75 per cent).

31 All these figures in relation to cost are based on the present high prices. The investment required for a 5000-kw. plant using 100 tons of pulverized coal daily is approximately $64,000 and for a 50,000-kw. plant using 1000 tons of pulverized coal daily, approximately $240,000, so that, on a basis of 6 per cent and allowing for 365 days' continuous operation, the interest item will vary from

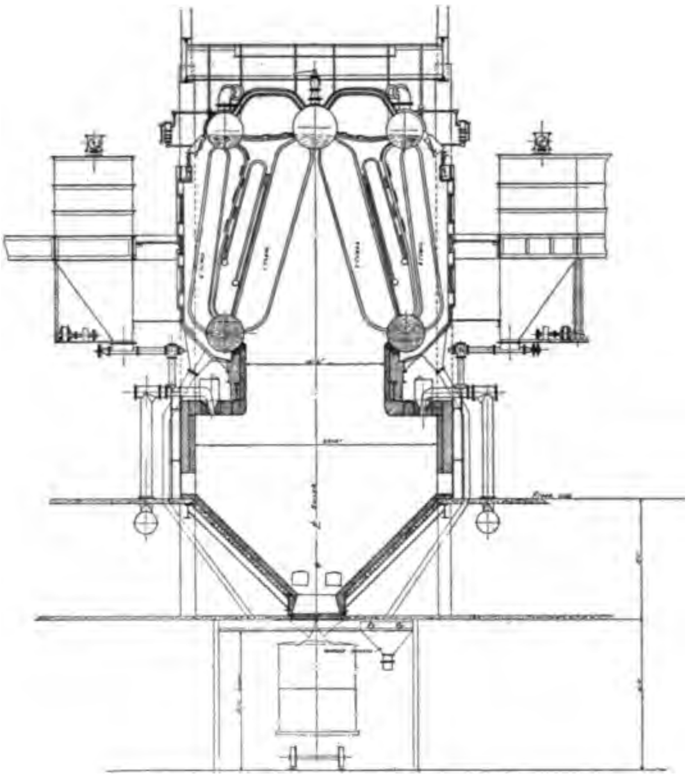


FIG. 1 2400-HP. DETROIT EDISON TYPE BOILER ARRANGED TO BURN PULVERIZED COAL AT 325 PER CENT RATING

10½ cents per net ton in a 100-ton plant down to 3.9 cents per ton in a 1000-ton plant.

32 *Depreciation.* Depreciation in a coal-pulverizing plant is usually calculated as follows: The life of the building is considered as 40 years, of the coal driers as 15 years and of the balance of the equipment as 20 years. With a 100-ton pulverized-coal plant and burning equipment the depreciation item will be approximately 12

cents per net ton, and in a plant of 1000 tons daily capacity it will be approximately 4 cents per net ton.

33 *Taxes and Insurance.* Taxes and insurance are based on 2 per cent of the entire investment and for a 100-ton plant this item is approximately 3½ cents per ton and for a 1000-ton plant, 1.3 cents per ton. Summarizing, the foregoing results show that the total cost of pulverizing and delivering pulverized coal to boilers is approximately as given in Table 1. The cost of the pulverizing equipment complete compares favorably with the stoker equipment when everything, such as coal- and ash-conveying machinery, etc., is taken into consideration, and in large plants it is considerably less.

COST OF STOKER OPERATION

34 The equipment required for a first-class stoker installation must necessarily be taken into consideration when making a comparison of the costs of the different installations. The cost of pulverizing is an item of expense which must be included with the cost of the fuel, and since it includes the complete handling of the coal, the expense of crushing, handling, power, repairs, maintenance, interest, taxes and insurance covering the stoker equipment must also be considered when making comparisons.

35 Stoker installations and operation are expensive and the investment is as great, if not greater, than that required for a pulverizing equipment in plants of 10,000 kw. and upward. For example, in a plant using 1000 tons in 24 hours the cost of operation will be approximately as follows:

Power for stoker, 2 per cent of the total boiler hp. developed..		$180.00
Power for fans, 2 per cent of the boiler hp. developed.......		180.00
Coal handling, 100 kw. at ¾ cent per kw-hr................		18.00
Labor for coal handling, 2 men per shift and 3 shifts at 40 cents per hour....................................		19.20
Repairs for stokers at 30 cents per boiler hp. per annum.....		17.50
Repairs for coal-handling equipment......................		10.00
		$424.70
Total cost per net ton......................	$0.425	

To this must also be added the cost of fuel used to heat the moisture in the coal, interest, depreciation, insurance and taxes, showing that even on a basis of equal efficiency the cost of operating a pulverized-coal equipment is considerably less than the cost of operating an equivalent stoker installation. It should be stated that the figures just given are based on present average results in both cases.

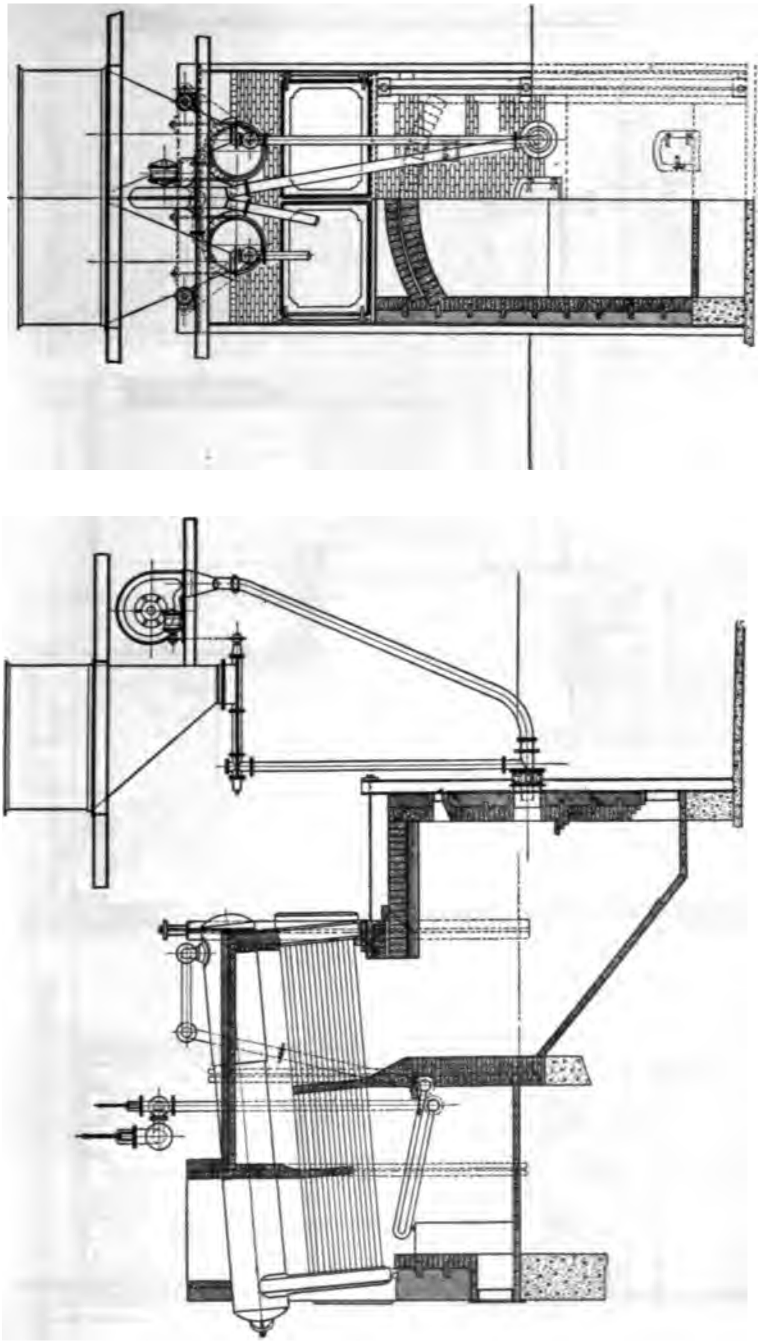


FIG. 2 425-HP. HEINE BOILER ARRANGED FOR PULVERIZED-COAL FIRING
(Designed by Fuller Engineering Co., Allentown, Pa.)

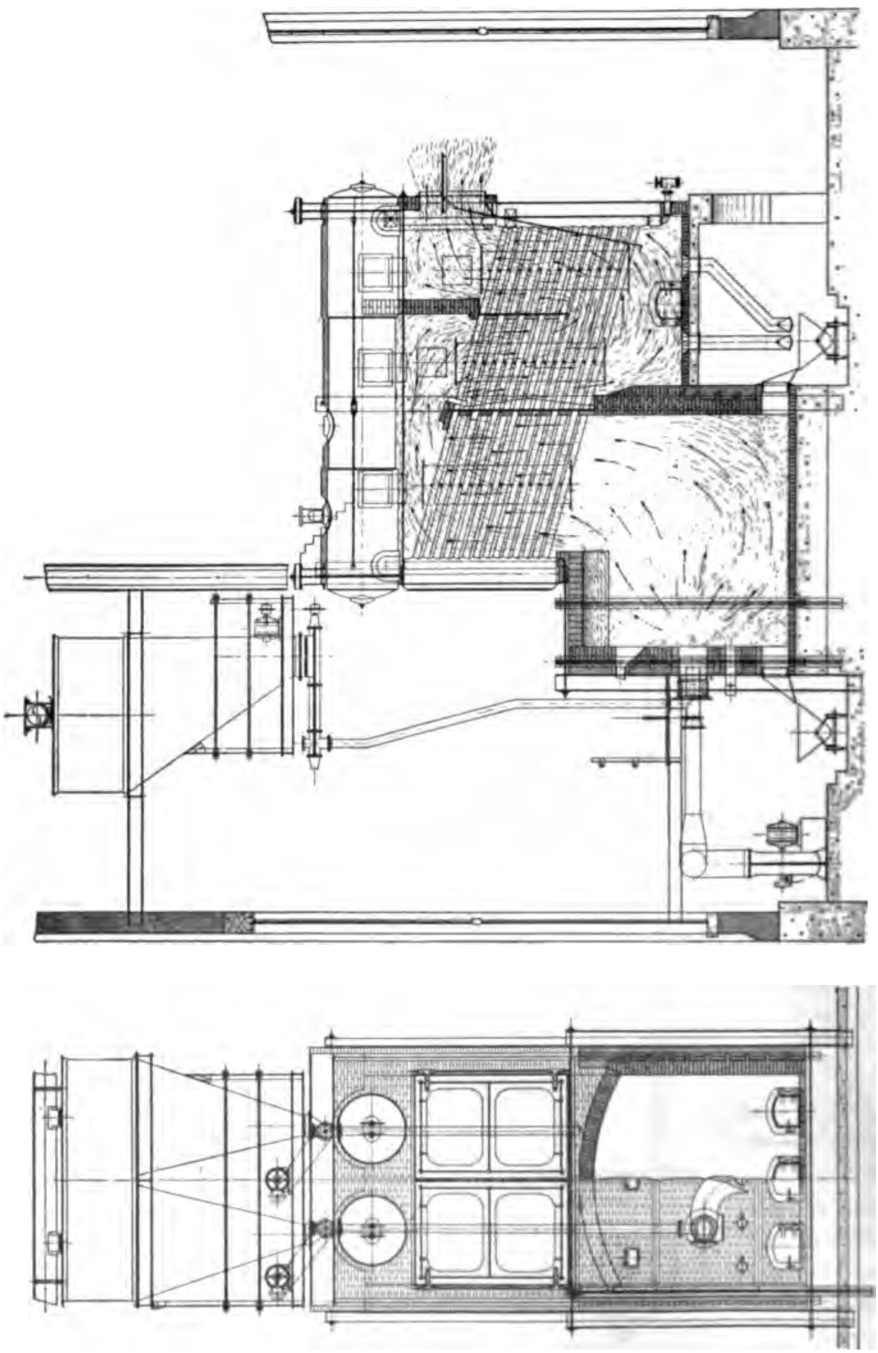


FIG. 3 500-HP. BABCOCK AND WILCOX BOILER ARRANGED TO BE FIRED WITH PULVERIZED COAL
(Designed by Fuller Engineering Co., Allentown, Pa.)

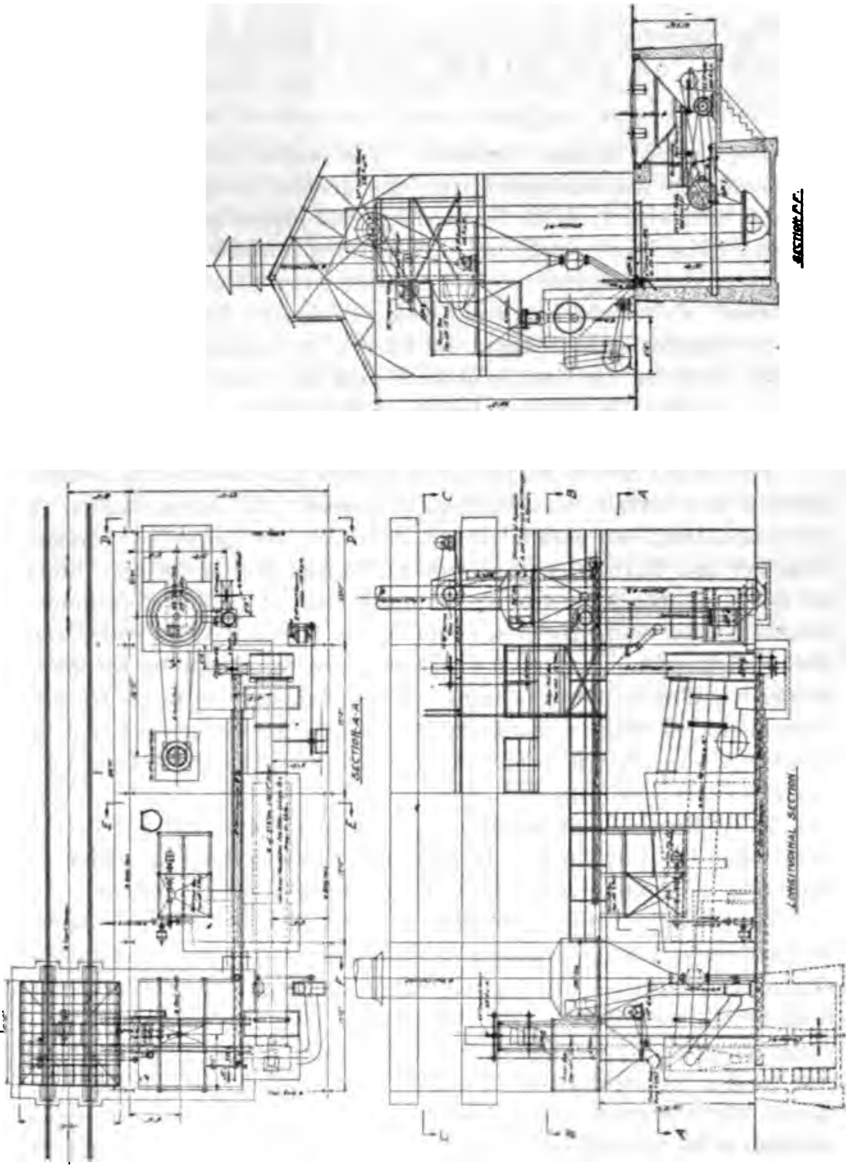


FIG. 4 STANDARD PULVERIZED-COAL PLANT WITH 42-IN. MILL AND 3½-FT. × 42-FT. DRIER

DISCUSSION

The following discussion on pulverized fuel applies to the preceding papers, Nos. 1701 and 1702, which were discussed jointly at the meeting.

Thomas A. Marsh (written). The writer would preface his discussion by the statement that the general proposition of taking raw coal with all of its by-products and preparing it to the most expensive size (pulverized) and then burning the raw coal and losing the by-products, appears to be a move very much in the wrong direction. Pulverized coal has a field, no doubt; for instance, such fuels as lignites which cannot be burned on stokers. This field is limited, however, for chain grates are now developed to burn lignites containing from 30 to 35 per cent moisture, which embraces practically all but those of the North Dakota field.

Pulverized fuel is adaptable to certain processes as the cement process and certain metallurgical processes. In substantiation of its adaptability for steam boilers, however, the paper by Messrs. Scheffler and Barnhurst represents pulverized fuel at its best, hints at not a drawback or trouble, submits tests of four or five hours' duration and evidently not made under the A.S.M.E. code, and contrasts stokers of the most expensive type to operate as to power to drive and as to power for fans. If it is necessary to resort to such extremities to make a comparable showing for pulverized fuel, it cannot be said that it is on a competitive basis with modern automatic stokers and furnaces.

The following discussion and questions are submitted in the hope that the full answers thereto will provide facts on which to base conclusions in accordance with the Society's past standards.

Reliability. Under the item of "Reliability" in the above-mentioned paper it is stated unqualifiedly that the mechanisms for pulverizing fuel are equally reliable with those for crushing coal for stokers. The extra equipment necessary for this exceedingly elaborate preparation of the fuel is another link to the mechanism (and a high-speed one at that), and I think no engineer will agree that the combination of a crusher system and a pulverizing system is as reliable as a crusher alone. Even, if this complex system were as reliable as a crusher, which it is not, the station dependent on pulverized fuel would suffer shutdowns unless the crusher and pulverizer equipment were duplicated. The writer understands that cement plants are usually equipped with duplicate pulverizing systems. This will all have a bearing on investment charges.

Further, it is stated in the paper that all grades of coal can be used. There are stokers which have limitations only on anthracite or coke breeze and in this connection the question arises as to whether a pulverized-coal furnace and burner designed for, say, bituminous 35 per cent volatile will burn anthracite or coke breeze. I ask this because in one of our other technical societies it is a matter of recent record that for burning low-grade anthracite or coke breeze in pulverized form, reversal of the flame is necessary, a firing method not suitable for the richer bituminous coals. It was also brought out that with coke breeze or anthracite culm the fire was unresponsive and the load had to be varied to suit the fire rather than the fire to suit the boiler load. What occurs when a rich high-grade fuel is used in a furnace and burner designed for low-grade fuels?

Adaptability. Under this heading the authors state that the primary feature is the possibility of burning all grades of fuel without affecting the efficiency of the furnace. In order for this to be true, it is necessary to burn the 50 per cent ash coal with no more excess air than the 3 per cent ash coal and suffer no more ashpit loss in the former case than in the latter. The writer therefore specifically asks if the 50 per cent ash coal requires in practice no more air than the 3 per cent ash coal. When 50 per cent ash coal is used the sensible heat of this amount of refuse leaving the furnace chamber is in itself an item which will affect the efficiency of the furnace. Can efficiencies be independent of these items? The tests show wide variation in efficiencies. What causes these wide variations?

Chain grates are burning coals containing 35 per cent ash and producing some excellent results, and have been for years. It would be desirable along with this statement by the authors regarding high-ash fuels for them to submit figures of tests with over 28.4 per cent ash, which is rather a low percentage as compared with chain-grate practice with western fuels.

As to flexibility with pulverized coal, the writer has seen cases where the furnace became unresponsive when forced. There is certainly no evidence in the paper to indicate that the pulverized-fuel burner is quickly responsive over a wide range of loads; in fact the evidence is much to the contrary, for of all the tests referred to, the maximum overload indicated is 188 per cent, which is not considered a high overload in modern specifications and practice.

Furnace Design. Under this heading the tests of Table 3 are referred to. Tests of four or five hours' duration should not be included in an A.S.M.E. paper. Efficiencies of from 70 to 80 per cent are shown here. Starting and stopping conditions, surging of water in the boiler and other items are so uncertain as to make these short tests absolutely unreliable. Under similar conditions, tests of 100 per cent efficiency and even higher are frequently made.

To be more specific, the writer would inquire if in the test of April 16, 1917, at Seattle, the heating value of the coal was determined by a calorimeter or was it assumed. The even figure of 10,000 simply suggests estimated figures.

What great difference brought about the marked increase in efficiency between the December 4 and December 12, 1917, tests at Chanute, Kan.? This wide difference is inconsistent with the statement of uniform efficiencies, regardless of operatives, coal or other items.

On the January 28, 1918, test at Chanute, Kan., of 25 days' duration, were the water and coal weighed? Did the boiler operate this long without the blow-off being opened, or if the boiler was blown down, how was the blow-off discharge measured?

Tests of April 26 and 28 at Parsons, Kan. Some tests were made at this plant simply by measuring the coal in a spiral conveyor. Was this done in this instance or were the coal and water properly weighed? The writer has a record of other tests made at this plant, as follows:

Horsepower Developed	Combined Boiler and Furnace Efficiency Per cent
180	53.9
191	73.5
230	67.8
247	67.4
506	61
356	59.0
230.5	71.5

These results indicate the possibility of low efficiencies on tests with pulverized fuel. What, then, may we expect in daily operation? Why should we pick only favorable tests for comparative figures?

On the test of April 26, should not the percentage of rating be given?

The writer is surprised at the high calorific value of the slush anthracite in all tests reported, as it has the highest heat value of any coal in the tabulation. We should not confuse this with low-grade anthracite, which contains less than 9000 B.t.u.

The complete data of temperatures, draft and CO_2, together with methods of measuring fuel and water, should be given for these tests.

Referring to the statement that a modern stoker-fired power plant with the best average plant operation obtains an efficiency of not better than 63 to 65 per cent, while carefully conducted tests show 75 per cent efficiency, the writer suggests that the authority for this statement be mentioned so that full credit can be given. This may be true with some types of stokers not completely automatic but dependent largely on the firemen's skill. With the chain-grate type, however, it is far from true, in fact the chain-grate plant operates within 2 or 3 per cent of the test figures, and test figures with modern chain-grate furnaces run frequently in excess of 75 per cent net efficiency. It has always been a feature of chain, grate performance that daily operating results approach very closely to maximum test efficiencies with no deductions to be made for fans or other auxiliaries. The fuel costs for modern chain-grate power stations may be adduced to confirm this fact.

The authors have selected for their comparison those types of stokers with which it is most difficult to get high operating efficiencies. whereas if the comparison were made with completely automatic stokers such as chain grates, the statement made of the operating advantage becomes void, as chain grates are evidently more capable of reproducing test results in daily operation than the pulverized fuel furnace.

Cost of Pulverizing Coal. Under the item of Drier Fuel the writer notes that all costs are based on only 7 per cent moisture and yet many of the fuels mentioned in the tests contain more than 7 per cent. Comparatively few fuels have as low as 7 per cent moisture, particularly west of Pittsburgh. The figures run from 10 to 40 per cent in the western section of the country, figures of 15 per cent predominating. The writer suggests therefore that the drying coal cost should be doubled.

Interest. As mentioned earlier, the pulverizing equipment should be duplicated to prevent shutdowns, which would, of course, raise the equipment's interest charge. There are some uncertainties regarding the cost of this equipment, as some of the plants mentioned are reported to have cost several times the figures quoted in the paper. The writer would therefore safeguard this figure by making it at least three times that used by the authors.

Depreciation. In figuring depreciation, everything should be limited to 10 years, inasmuch as we have all seen three or four complete changes in power-plant equipment in the last 30 years and are likely to see the same in the next 30 years. To figure that the fuel-pulverizing building has a useful life of 40 years is making a pretty strong effort to show up pulverized fuel in a favorable light.

The writer would therefore recalculate Table 1 as given below.

TABLE 1 COST OF DELIVERING PULVERIZED FUEL TO BOILER

	100-ton plant, dollars per net ton	1000-ton plant, dollars per net ton
Power at ¾ cent per kw-hr. and 17 kw-hr. per ton	0.1275	0.1275
Labor at 50 cents per hour (40 cents too low)....	0.175	0.05
Drier coal at $5 ton delivered (15% H_2O).......	0.12	0.12
Repairs (no data — accept authors')...........	0.07	0.07
Total cost of pulverizing per net ton[1]......	$0.4925	$0.3675
Interest at 6 per cent. (Triple the investment)..	0.315	0.117
Depreciation (10-yr. basis); Multiply authors' figures by 3...........................	0.36	0.12
Taxes and insurance (no data —accept authors' figures)...............................	0.035	0.013
	$1.2025	$0.6175

[1] In existing plants of 500 to 600 tons daily capacity this figure is reported to actually be $1 or more per ton, indicating that in actual practice and operation many incidental items of cost enter which are not shown in a theoretical estimate.

Cost of Stoker Operation. In making this comparison, the authors have again selected those types of stokers which show highest operating costs and cause the pulverized-fuel installation to benefit by contrast.

The power to drive some types of stokers may be as high as 2 per cent. Chain grates require but $\frac{1}{10}$ of 1 per cent. Power for fans with those types of stokers requiring fans may be 2 per cent of the power developed. Chain grates have no fans, so this item is eliminated. Coal handling would seem to be an item which would cancel out, as it should cost no more to deliver coal to the stoker hoppers than to the pulverizer. However, accepting these figures on the authors' basis, we have as follows per 1000 tons of coal:

Power for stokers, 1 hp. per 1000 developed; chain-grate practice, 20 cents per 1000 lb. of steam	**$ 4.00**
Power for fans, standard chain-grate practice	**0.0**
Coal-handling power (accept authors' figures)	**18.00**
Labor for coal handling (accept authors' figures)	**19.20**
Repairs for stokers (accept authors' figures)	**17.50**
Repairs for coal-handling equipment (accept authors' figures)	**10.00**
	$68.70
Total cost per net ton	**$0.0687**

We have, therefore, an additional cost of from 55 cents to $1.13 per ton against the pulverized-fuel plant, a figure that cannot be made up by increased efficiency, for it must be realized that many stoker installations have given performances within 5 per cent of the highest theoretically possible.

Now as to some of the items not mentioned by Messrs. Scheffler and Barnhurst in their paper.

Fine Ash. The writer understands that from 60 to 80 per cent of the ash carries over to lodge in tubes and combustion chambers, but mostly goes out of the chimney to scatter over the surrounding country. This may by comparison not be objectionable in a cement mill or in some of the localities mentioned in the list of installations, but it is evident that if the Commonwealth Edison Company of Chicago were to scatter fine ash from the 5000 or more tons of coal they burn daily, say 600 to 800 tons of ash, it would create a condition compared to which the worst smoke fog would seem like a paradise.

Slag. Some installations have encountered trouble due to the fine ash slagging in the tubes. The writer would inquire how prevalent this is, how it is overcome or prevented and what influence high boiler ratings seem to exert on it.

The refuse from pulverized-fuel firing contains some combustible. It has been reported that the slag containing this combustible causes damage to ashpits. Is this a serious item of maintenance?

It is reported that pulverized fuel if stored will pack and become difficult to handle. Where the container is jarred or vibrated this action is reported to increase. How serious is this action? Does it interfere with the use of stored powdered coal? Does it preclude its use on shipboard?

Turning now to Mr. Harrison's paper, the coals he states as most suitable seem to have (*a*) less than 10 per cent ash, (*b*) volatile

30 to 40 per cent, (*c*) low sulphur, and (*d*) high melting point of ash. This, of course, means a very choice bituminous coal. Such fuel commands a high price in any market, much higher per 1000 B.t.u. than the high-sulphur, high-ash fuels frequently competing. For instance, there are in Illinois and Indiana coals containing 18 to 20 per cent ash, 30 per cent volatile, 6 to 8 per cent sulphur, with the fusion point of ash at 1900 deg. fahr. The writer would inquire as to whether in the present state of the art this fuel could be commercially used to develop, say, 150 per cent rating from a boiler by means of pulverization.

It seems to be a prevalent opinion that lignites cannot be burned on stokers. The writer would rectify this at once. Lignites and sub-bituminous coals are being burned and producing high efficiencies and capacities. The following results of a few tests will substantiate this statement.

Fuel	Colo. Lig.	Tex. Lig.	Mont. Sub-Bit.	Mont. Sub-Bit.	Mont. Sub-Bit.	Mont. Sub-Bit.
Moisture, per cent	23.73	29.93	20.73	15.28	15 27	20 52
Volatile, per cent	35.25	30.96	33.44	37.56	41.24	31.23
Fixed carbon, per cent	33.23	25.45	32.50	38.07	30.91	39.00
Ash, per cent	7.79	11.66	13.28	15.09	12.58	9.25
B.t.u. (comm.)	8511	7124	8772	9118	9553	9096
B.t.u. (dry)	11,116	10,314	11,073	10,763	11,275	11,444
Type of stoker	Green	Green	Green	Green	Green	Green
Type of boiler	B & W	B & W	B & W	B & W	B & W	B & W
Coal per sq. ft. per hr., lb	28	32	28.91	38.06	30.46	35.31
Furnace draft, in	0.232	0.32	0.167	0.221	0.17	0.196
CO_2 at damper, per cent	12	11.81	10.36	12.97	11.77	12.62
Per cent, rating developed	128	.....	145.8	209.1	181.5	195.09
Combined efficiency, per cent	69 7	69	72	75 6	77.3	78.04

The ability to burn these coals is a development of the last few years, and has opened up a wonderful fuel field to chain grates. Some of these fuels contain as high as 30 per cent mositure, in which case the drying cost when pulverized fuel is used would be 4½ times those given in Table 1 and would make it necessary for the pulverized-fuel installation to obtain somewhere in the neighborhood of 100 per cent efficiency to be on a competitive basis.

Moreover, these fuels do not at this time command a price anywhere near $5 per ton, so that the cost of drying and pulverizing represents probably about 25 per cent of the fuel cost, a figure not to be regained by increased efficiency.

Reference is made to the use of low-grade refuse around the mines, as at Lykens, Pa. Mr. Sheffler and Mr. Barnhurst also refer to this same fact in their paper. The coals they mention, however, have calorific values not only far beyond the average idea of culm piles (12,500 to 15,400 B.t.u.), but are good enough to be used to advertise the coal. Certainly no western fuels and few mid-western fuels will equal these supposedly refuse coals.

It should be understood that when a chain-grate engineer refers to low-grade coal, he certainly has in mind fuel containing less than 9000 B.t.u., dry basis. No such fuels seem to be discussed even as refuse fuels in pulverized-fuel practice.

Mr. Harrison bases his costs of pulverizing on pre-war labor prices (30 cents per hour for millers, 20 cents for firemen and common labor), war coal prices (which is not consistent with 20-cent labor), and 7 per cent moisture in the fuel. His figures for labor should be doubled. For average fuels the moisture percentage should be doubled, and for the real field as mentioned by the author at least quadrupled for coal with 30 to 35 per cent moisture.

Referring now to (*a*) under Pulverized Coal vs. Stokers, the writer would inquire if a furnace suitable for lignite will handle anthracite. High overloads are spoken of freely and he would also inquire just what ratings have been sustained or even reached to confirm this.

Stokers are criticised in (*c*) as requiring more area for high overloads than pulverized-fuel burners. Has this been proved to be commercially true? With chain grates and coals such as mentioned in the average pulverized-fuel test, ratings of over 250 per cent are sustained for long periods, and peaks swung for short periods beyond this rating. Have any pulverized-fuel installations so far equaled this?

Referring to (*e*), the writer would state that not only is ash, more easy to handle, but there is only about one-fourth of the usual amount, the remainder going out the chimney. This is one item of decreased cost that pulverized-fuel exponents seem loath to claim. We certainly must overcome this difficulty if we are to make pulverized-coal plants commercially successful, as our city and health ordinances will soon put the ban on this wide distribution of fine ash.

The writer would correct item (*f*) to say that coal burned on stokers may contain from 1 to 35 per cent of free moisture as fired. As a point of interest, he would inquire as to how readily pulverized coal picks up moisture when stored.

Under (*d*) in the following paragraph certain changes in the furnace are mentioned to suit various coals, also rather extensive changes in drying equipment. This is contrasted with stoker practice. The author presents as a favorable argument for pulverized-fuel burners that, with only a slight change in the furnace and some increased drying capacity, the equipment may be changed from one suitable for anthracite to one suitable for lignite. The writer would inquire in what commercial fuel market or location is this a valid benefit. What market receives both anthracite and lignite as steam coals?

Referring to the comparison of costs given under the heading Pulverized Fuel vs. Mechanical Stokers, the writer would question the estimated figure of 32 cents and would state that since the installation had its acceptance test ten months ago, the costs should not be concealed, but should be made part of the paper.

Reference to Table 1 in the complete paper would indicate that pulverizing cost would be

(*a*) For pulverizing and drying 12 per cent moisture coal...	$0.516
(*b*) Labor (use 2 times tabulated cost)	0.08
	$0.596
Maintenance (use author's estimate).................	0.03
	$0.626

As a matter of fact, in existing installations of from 500 to 600 tons daily capacity, the cost of drying and pulverizing fuel is over $1 per ton. This makes pulverized fuel prohibitive for steam-making purposes.

From all the figures presented, some good efficiencies are indicated when high-grade coals are burned (no tests are presented on low-grade coals). The advantage in efficiency, however, is insufficient to offset the increased cost of preparing the fuel, even on a basis of $5 fuel. This causes the pulverized-fuel burner to operate at a loss, regardless of the efficiency obtained. Fuel costs will have to be from 50 to 100 per cent higher before the increased cost of pulverizing will be justified, unless pulverizing and drying costs decrease materially.

Further, no modern ratings are recorded in the paper; furnace and slag troubles are minimized; quick pickups of boiler load over a wide range of rating are not proved and seem very problematic; ash from chimneys is at present a prohibitive feature in cities.

Fred'k A. Scheffler (written). I cannot agree with Mr Harrison in his statement that when a boiler is being run with pulverized coal it is not necessary to have more than 0.10 to 0.15 in draft at the damper of the boiler, and consequently the boiler could be operated with stacks about 30 to 35 ft. in height. It is a well-known fact that it is necessary to have at least 0.10 in. in the furnace, and that the average water-tube boilers have a frictional or draft loss through the boiler of about 0.30 in. when run at rating, and when run at 200 per cent of rating this draft loss is almost double, or 0.60 in. Consequently, the stack would have to be at least 125 ft. in height in order to be sure that there is sufficient draft to overcome the frictional resistance through the boiler and allow a suction of at least 0.10 in. in the furnace.

W. N. Best (written). I should like to inquire how many open-hearth furnaces there are in the United States that are successfully burning pulverized coal, and for how long a period they have been operated.

The Boiler Test Code Committee of the A.S.M.E. would like to know of large power plants that are and have been successfully burning pulverized coal for a period of, say, 4 years. We have endeavored to locate such plants for some time in order to examine same and secure some data for the Society. I am aware that many plants have experimented with pulverized coal in the generating of steam, but results were very disappointing.

At the beginning of Mr. Harrison's paper, he states: " . . . and the shortage in the supply of crude oils which have become of too great value for ordinary fuel purposes." This, I think, is misleading, for statistics prove that the production of oil has never before been so great, and the demands for it never so great, as to-day. In Mexico alone there are many reservoirs as large as lakes filled with crude oil awaiting transportation, and hundreds of wells are capped awaiting to be turned on to meet the demand.

The increased price of coal and the liability of further increases in its price have compelled many boiler plants along the Eastern Coast to change to oil as fuel, and many more contemplate changing soon. The cost of oil is now very attractive in boiler plants owing to the fact that it only requires 147 gal. of oil to represent a long ton (2240 lb.) of bituminous coal, the coal having a calorific value of 14,000 B.t.u. per lb. One man can fire and water-tend twelve 300-hp. boilers. Oil is so attractive at the present time as a fuel along the

Atlantic Coast that I believe it will only be a matter of a year or a year and a half until all the larger power plants will use Mexican oil as a fuel in their boilers.

It is my opinion from close observation and study that the two distinct fields for pulverized coal are the furnaces of rotary kilns and copper matting furnaces. Both of these are constructed so that the building is quite open, and there is practically no liability of explosions being caused by spontaneous combustion. I believe the large quantities of poor coal and lignite that we have in our country can be successfully burned in combination with oil, the coal and lignite referred to being, of course, in the pulverized state. By this combination practically perfect combustion can be attained and maintained at all times, and this combination could be of value owing to the low calorific value of the coal and lignite, and the high calorific value of the oil. The process of burning this combination of fuels would not be by mixing them together, but each fuel should be delivered separately to the furnace.

W. G. DIMAN (written). So far as the better grades of bituminous coals are concerned, I think that the cost of drying, pulverizing, conveying, feeding and the first cost of the apparatus, together with the fact that all grades of fuel can be burned with a tolerable degree of smokelessness and efficiency in the regular stoker apparatus will restrict the use of pulverized coal for boiler purposes to special cases. If low-grade waste coal can be utilized for pulverized fuel there is a great field for its use. There must be some limit, however, in the use of low-grade fuel, for the cost of grinding would eat up any advantage in the economy. Where low-grade fuels are high in ash and slate there should be a limit to the proportion that one can afford to grind. The use of high-ash coal will be bad, as it will accumulate rapidly and be difficult to remove, especially if it slags to any extent. This would occur more where the ash deposits are within the limit of the flame. With anthracite, especially culm and of that nature, it might be difficult to pulverize and must be finely pulverized; it needs a higher temperature for combustion, burns more slowly, and in certain boilers like a H.R.T. it would require plenty of brickwork and a large combustion chamber to avoid the chilling effect and to maintain combustion. The best coal to use is one high in volatile and low in ash. Such a coal would not be a very satisfactory one for outside storage in large quantities due to the likelihood of spontaneous combustion. In order to successfully use the pulverized fuel, I am of the opinion that the best results can be obtained by designing the furnace and equipment to meet a specific grade of fuel. I do not think that the average cost per ton for getting the coal from the car into the boiler is as low as stated, and if the overhead is also taken into consideration it will run still higher.

E. H. Peabody (written). In Mr. Harrison's paper I note that among other important matters he mentions three points which occur to me as particularly significant in the use of pulverized coal for boiler purposes:

1 Very large furnace volume required, or, in other words, the fuel must be burned under conditions which imply a very low rate of combustion per cubic foot of furnace volume;
2 The continual effort necessary to keep the boiler tubes, and, to a less degree, the furnace itself, free from slag and clinker caused by the refuse in the coal; and
3 The very high and apparently inaccurate, boiler efficiency reported in the tests of August 12–13, 1918, which appears to be due to crediting the boiler with work done by the coal driers.

If, as seems to me proper, the boiler efficiency is figured on the basis of the heat value of the dry fuel, the result should be 76.3 instead of 85.2. The degree of efficiency obtained in the drier itself would in no way affect the results obtained in the boiler in actually transferring the heat in the fuel to the steam.

Oil fuel is undoubtedly superior to all others for boiler purposes. The furnace volume required for this fuel to give satisfactory results is about one-quarter the furnace volume specified in the paper for pulverized coal.

The similarity of action in many features between oil fuel and pulverized coal appears to me to constitute one of the principal attractions of the latter. It would seem, however, that the large furnace, with its extra first cost and extra radiation loss, together with the difficulties due to slag and their effect on operation, would offset the desirability of pulverized coal to a very large extent.

Albert A. Cary (written). To many who have not had occasion to keep informed concerning the previous use and past applications of powdered coal as a fuel, the idea seems to be prevalent that such fuel is a recent development, holding almost unlimited possibilities for all kinds of furnace applications and capable of easily and cheaply accomplishing phenomenal results.

As a matter of fact, inventors have been struggling for almost a century to make powdered coal a practical and successful fuel.

In 1831 an English patent was issued to J. S. Daws for a process for burning powdered coal, and this was rapidly followed by a large number of patents relating to this subject in England, the United States, Germany and elsewhere.

In 1881 a United States patent was issued to C. H. Palmer describing the means for feeding fine coal to a locomotive boiler with an air blast, while in 1870 Whelpley & Storer began to take out a series of patents for using pulverized coal in reverberatory metallurgical furnaces. In 1876 an elaborate series of tests was conducted by the Bureau of Steam Engineering of the Navy Department under the direction of B. W. Isherwood — using the Whelpley & Storer pulverized-coal system applied to a steam boiler.

Thus we are able to understand that the preparation and use of pulverized coal fuel is by no means a recent development, and notwithstanding the considerable expenditure of money, effort and ingenuity by many inventors, including men whose past experiences qualified them to carry on such work; most of these older productions have been scrapped so that today we have left merely the benefit of their varied experiences, which has, nevertheless, proved a valuable source of information for the more recent developments in the preparation and use of pulverized coal.

Turning now from this record to the more recent developments, it will require but little investigation to find that during the last decade (or even for a much shorter period) there have been many powdered-coal installations which have either been rejected or found to give poor satisfaction.

After the presentation of such facts, I may be accused of condemning the use of pulverized fuel as a practical and efficient fuel.

On the contrary, in the light of my experience and with the evidence presented by many satisfactory equipments now in operation, I am an unqualified advocate of the use of such fuel, but I must limit my endorsement to applications where a desirable grade of fuel is available — where proper preparation of the fuel and proper furnace conditions can be obtained and where other methods for burning the available fuel are inferior to accomplish the desired heating

About 24 years ago I was called upon to assist in the development of a pulverized coal equipment for a cement plant, and since that time I have been called upon to test, investigate and design a number of pulverized coal equipments for cement kilns, boilers, metallurgical and other industrial furnaces and thus have had an opportunity to follow the development of this art since the early days of its practical commercial adoption in this country.

Pulverized Coal in Cement Plants. Mr. Harrison has very properly given first place in his paper to the use of pulverized-coal fuel in the rotary kilns of our cement plants, as not only do we owe the present development of our pulverized-coal equipments to the pioneer work done in their development at such plants, but the very form of these long, cylindrical, refractory-lined furnaces furnishes us with the ideal construction for the use of this form and kind of fuel.

They permit the use of the long, air-projected current of finely powdered fuel, giving it ample time to complete its combustion while being held in suspension and without direct flame and ash impingement upon any impeding furnace wall.

The simplest form of burner and fuel-feeding device can be used with such furnaces, and the ash resulting from the combustion of this finely ground coal causes little or no trouble, providing the ash content and its quality remains nearly constant, as this ash drops down and mingles with the burned clinker without materially affecting the quality of the final product.

Pulverized Coal in Metallurgical Furnaces. Next in order of desirability in the way of furnace design for the use of pulverized coal, we have our reverberatory metallurgical furnaces of great length (100 to 150 ft.), such as are used for copper smelting, where the flame is projected from the front toward the rear of these furnaces over the charge of ore on the hearth.

Without the exceptional facilities for taking care of the ash resulting from the combustion of the pulverized coal as found in the rotary cement kilns, ash troubles proved to be a pretty serious matter with the early pulverized-coal installations applied to these furnaces about a dozen years ago.

The ash fused and formed a slag blanket over the top of the ore charge and stuck to the interior of the flue outlets in a way to block these passages; but these and other associated troubles were finally overcome by stopping the infiltration of large amounts of cold air, by stopping the practice of charging large quantities of cold ore into the furnace, by using a better design of burner and coal-feeding device, and by a more careful preparation of the pulverized coal.

By these means a much higher temperature was maintained in the furnace, a very much higher ratio of charge to fuel used was obtained, and a better and more constant regulation was secured at the burner. Then pulverized coal began to be recognized as a most excellent reverberatory fuel.

When pulverized coal was used as a fuel for shorter and smaller reverberatory furnaces and directly fired into their interiors, the above-named troubles were accentuated and due to the imperfect absorption of the heat in these furnaces, the use of waste-heat boilers generally became a necessity in order to obtain efficient results. Accumulations of more or less fused ash piled up rapidly on the heating surface of these boilers necessitating frequent cleanings. Experience has shown that the reversing type of furnace for the production of open-hearth steel, such as described in Mr. Harrison's paper, is the best design of pulverized-coal furnace for that purpose. Checkerwork and baffie walls must be specially designed for use of this fuel.

A modification of the above-described reverberatory furnace consisting of an extension furnace in front of the main furnace chamber, so proportioned as to retain a large percentage of the ash within this chamber, makes a highly efficient equipment. It requires a burner equipment permitting a close regulation of its coal and air supply, and one which will produce a short, brush-like flame close up to the burner, which means a very rapid combustion and a high temperature in the combustion chamber.

In some recently-built extension furnaces a large percentage of the ash is separated from the body of the flame and drops into a comparatively cool part of the chamber as a fine, dry ash mingled with but a small percentage of fused ash nodules, which do not interfere with the easy removal of the refuse from the furnace bottom.

With other recent designs of extension furnaces, where very high temperatures are maintained at the burner outlet, the rapid fusing of the ash is expedited, which refuse drops to the slag pit below in a molten mass and under proper conditions may be drawn off through tap holes from the bottom of the extension furnace.

Such a design of extension furnace has been installed by the

Lopulco Company in a steel plant near Pittsburgh, in connection with their heating furnaces, where it has given excellent satisfaction.

Application to Stationary Boiler Plants. The application of pulverized-coal fuel to steam boiler furnaces meets with many challenging difficulties, which, however, recent developments have done much to overcome. A number of such installations are in use, some of which are showing very desirable economy and producing a material increase in the steaming capacity of the boilers.

Great care must be taken to prevent any of the unconsumed coal from coming in contact with the chilling water-containing surfaces of the boiler, which, therefore, demands the use of an extension furnace in which the complete combustion of the coal is accomplished, and in which the proper handling of the fine ash (and its resulting slag) is provided for.

All that has previously been said concerning the requirements needed for extension furnaces applied to short reverberatory furnaces apply with equal force here.

With the intense heat generated in burning pulverized coal much trouble has been experienced in improperly designed boiler furnaces due to the rapid melting down of the refractory linings, which may also be subjected to the scouring or cutting action of impinging ash, thus requiring frequent and expensive relining.

Process of Combustion of Pulverized Coal. Pulverized coal *must* be burned while suspended in the current of air which accompanies it into the furnace.

As the finely divided coal enters the highly-heated furnace, any moisture contained by each minute particle is first driven out, and if this is excessive, it will form a steam cloud around the particle and suppress its further rapid combustion, thus defeating to a greater or less extent the purpose for which the coal was prepared.

One of the necessities for supplying the furnace with very dry pulverized coal is thus appreciated.

With little or no moisture present, the next effect of the very high surrounding temperature in the furnace is to almost instantly distil off the volatile matter occluded in the coal as this gas is liberated at a comparatively low temperature and thus, with properly designed furnace and burner, the dust cloud entering the combustion chamber suddenly becomes thoroughly saturated with a highly inflammable gas, mixed with an ample supply of air.

Under these conditions the gas is raised to its ignition temperature with great rapidity, and the flame produced is propagated with intense speed throughout the atmosphere of fuel and air entering the furnace; all of which is simultaneously subjected to the same action.

With the intense degree of heat thus generated, not only is the temperature of the furnace maintained to stimulate further combustion, but the particles of fixed carbon (or flecks of coke) left behind, after the volatile gases have been driven out of the coal, have their temperature raised with great rapidity to their temperature of ignition and so are most speedily burned, leaving behind the non-combustible matter (which we have called ash) to be more or less effectively taken care of in a properly designed furnace, or to give great trouble in furnaces where its importance has been ignored.

Mr. Harrison very tritely states that "The *finer* the coal is pulverized, the more efficiently it can be burned."

With the above analysis of the process of combustion before us, we can more readily understand the reason for this statement. Not only does the greatly increased fuel surface (presented to the highly heated interior of the furnace) facilitate the more rapid distilling off of the volatile gases, but with the volume of the particle itself greatly reduced, penetration of the heat to its interior takes place in a very much shorter period of time. Aside from these advantages the important fact stands out that the Law of Mass Action applies to such combustion (which is akin to the action of explosives) by which we find that the amount of the reaction in a unit of time is proportional to the active mass, or, in other words, the smaller the particle of coal, the smaller the number of gram-molecules of combustible matter found in the unit mass.

Under these most favorable conditions, the velocity of combustion is greatly accelerated, which is the principal object striven for when pulverized coal is chosen for a fuel.

A moment's thought will recall the fact that the volume of a mass varies as the cube of its diameter and thus a particle of coal $\frac{1}{100}$ in. in diameter has eight times the volume that is found in another particle that is $\frac{1}{200}$ in. in diameter.

The effect of the decreased surface in proportion to the mass results in a very great retardation of the combustion of the larger particles as is very plainly shown in the photographic study of the combustion of coal dust in Bulletin No. 102 of the Bureau of Mines, entitled The Inflammability of Illinois Coal Dusts on Plate IV. B and Plate V. B.

Uniformity in the size of pulverized particles is also very desirable, as we can readily understand that if we have mixed a considerable percentage of each of the above-named sizes of coal fed to our furnace, the combustion of the larger size will proceed much slower than the burning of the smaller size, which condition not only reduces the velocity of combustion of the total mass, but it prevents the production of that "gas-like flame," which Mr. Harrison refers to.

When such grades of coal, averaging from 30 to 40 per cent of volatile matter, as described above, are pulverized so that over 90 per cent will pass through a 200-mesh screen, the finer coal accompanying it does not seem to materially affect its even rapid burning effect in general furnace practice; and further, as Mr. Harrison has stated, with this very fine pulverization, which insures rapid and high temperature combustion, the sulphur in the coal burns rapidly to SO_2 gas, which passes to the stack without affecting the product of the reverberatory furnace.

When a coal, running high in fixed carbon, is used as pulverized fuel, we have a different and less desirable set of furnace conditions.

Lacking the production of an ample supply of volatile combustible gases given off by the coal at a comparatively low temperature, and also lacking the heating effect which the burning of these gases produces, to hasten the combustion of the associated fixed carbon, we must necessarily maintain a higher furnace temperature to ignite this fuel, and to obtain the best possible results, the coal should be ground finer, as it will be found that this fuel is slower to ignite. Much of such coal carries a higher ash content, which still further complicates matters and lowers the heat value of the coal.

Mr. Harrison gives a table purporting to give the cost of coal pulverizing equipments and the cost of preparing the coal per net ton. I question the reliability of these figures. Nothing is said of the pulverized-coal system upon which these figures are based nor the class of machinery or equipment included.

There is a considerable range in cost of equipments as furnished by different concerns and the figures presented in this table appear low for the better class of equipment.

Much depends upon the class of equipment one is willing to install, which has much to do with results obtained including cost of future upkeep.

There are many elements entering into the cost of such equipments. For example: should one be obliged to handle very wet coal in the coal driers, the normal capacity of the drier would be reduced and either larger driers or more driers would be required to obtain a fixed amount of dried coal, which would also require a greater building space.

Again, should one decide to grind the coal very fine to obtain maximum furnace results, the output capacity of the pulverizer would be materially reduced below that needed for coarser pulverization and more or larger mills would be needed, and thus I might continue this list so as to include most of the contained units.

Whereas, Mr. Harrison's total cost for pulverizing 80 and 90 tons of coal per day agrees pretty well with the cost estimate given in this table, his cost for producing 100 tons per day runs 21 per cent higher than the figure shown in the table.

In looking over the log of the boiler test quoted by Mr. Harrison I note what appear to be a few discrepancies, but lacking full and complete data, I have been unable to check them up as carefully as I would otherwise do.

Taking his analysis of furnace gases, he gives the maximum result obtained as 15.4 per cent of CO_2 and 5.6 per cent of C with no CO.

This is consistent and possible.

He then gives his minimum result obtained as 12.2 per cent of CO_2 and 3.2 per cent of O with no CO; the sum of these gases giving 15.4 per cent with 84.6 per cent of N by difference.

There is evidently some discrepancy here, as I do not see how it is possible to burn such a coal as he indicates so as to obtain such an analysis.

The percentage of boiler efficiency given in this table as 85.22 per cent seems entirely inconsistent and I cannot find sufficient indication contained in the other data submitted to assure me that the total losses in the performance of this test amounted to only 14.78 per cent.

One very interesting question in connection with this test is, What means were used for accurately weighing the pulverized coal used? As a matter of secondary importance I might also ask, What means were used for ascertaining the weight of water fed to the boiler?

J. E. Muhlfeld (written). Mr. Harrison's excellent paper reviews data that conform in general to my experience during the past few years in the development of systems for the more effective and economical utilization of coals and lignites in pulverized form, for stationary, locomotive and marine boilers, and for metallurgical and chemical heating furnaces, and the information that he has given will be of great assistance to existing and prospective users of this method of firing and burning solid fuels.

Until recently power-plant capacity and efficiency have been dependent upon combustion possibilities. Fortunately, this has now changed and the problem is in boiler design and construction.

In the log of test on a pulverized-fuel-burning stationary boiler given by Mr. Harrison, the operating capacity deductions on account of the power and other items necessary for the preparation of the coal in pulverized form should be 3.17 per cent, in place of 4.22 per cent, as used in arriving at the net efficiency. This figure of 3.17 per cent compares with the power necessary to operate mechanical stokers, and which, in an installation of this kind, would amount to from 2½ to 5 per cent, varying with the boiler load carried.

In connection with this log of test data it may be of interest to give the heat balance for the same 24-hour run, which is as given in Table 2.

Furthermore, to show the definite control over stand-by losses

TABLE 2 HEAT BALANCE FOR BOILER TEST IN TABLE 3 OF MR. HARRISON'S PAPER

Ultimate analysis of pulverized Indiana and Illinois screenings as fired, given in percentages:

1 Moisture	3.20
2 Carbon	55.66
3 Hydrogen	4.47
4 Oxygen	19.77
5 Sulphur	1.97
6 Nitrogen	1.45
7 Ash	13.48
	100.00

The distribution of the calorific value of one pound of coal as fired among the several items of heat utilized and lost is as follows:

	B.t.u.	Per cent
Heat absorbed by boiler	9,934	85.22
Loss due to evaporation by moisture in coal	41	0.35
Loss due to heat carried away by steam formed by burning of hydrogen	493	4.22
Loss due to heat carried away in dry flue gases	975	8.37
Loss due to carbon monoxide	0	0.00
Loss due to combustible in ash and refuse	51	0.44
Losses due to heating moisture, in consumed hydrogen, hydrocarbon, radiation and unaccounted for	163	1.40
Calorific value of fuel as fired	11,657	100.00

by the use of the same pulverized-fuel system, at the same plant, the data in Table 3 are of interest.

TABLE 3 SHOWING CONTROL OVER STAND-BY LOSS

Date	August 19, 1918
Boiler No. 5	Edge Moor, Rated 468 nominal hp.
Fuel feed shut off, uptake damper closed and auxiliary air inlets closed	9.00 p.m.
Boiler steam outlet to header closed and 175 lb. steam on boiler	9.20 p.m.
Safety valves released about one minute at the following times (p.m)	9.40, 9.55, 10.08, 10.15, 10.25, 10.38, 10.43, 10.52, 11.02, 11.09, 11.18, 11.28, 11.38, 11.48, 11.52.
Steam on boiler 158 lb. when fuel feed started and boiler steam outlet to header opened	7.00 a.m.
Drop of steam pressure in boiler, from 9 p.m. until 7 a.m., or during 10 hours while fuel feed was off and during which time safety valves popped 15 times, for one minute each, or a total of about 15 minutes	20 lb.
Time required to bring boiler from 155 to 175 lb.	4 minutes

To further show the positive control over the combustion, the performance of one of these boilers in Table 4 is of interest. During the 14-hour run, from 6 P.M., February 1, to 10 A.M., February 2, for which period the boiler was sealed and operated at about 150 per cent rating, it will be noted that there was a change of only 0.2 per cent in CO_2 and 0.4 per cent in O_2, and no slag was produced.

TABLE 4 SHOWING POSITIVE CONTROL OVER COMBUSTION

Date	Time	CO_2	O_2	CO
February 1, 1919	5.00 p.m.	13.00	6.00	0
	.....	13.6	4.2	0
	....	13.2	4.2	0
	6.30 p.m.	13.0	6.0	.....
	....	13.8	....	.....
	....	14.2	3.2	0.2
	.....	14.4	....	0
	.....	14.6	3.4	1
	9.00 p.m.	14.8	4.6	0
February 2, 1919	10.00 a.m.	13.0	5.0	0

Observations made of the stack from the outside of the power-plant building, when operating the pulverized-fuel and the stoker-equipped boilers in combination and independently have also demonstrated that no smoke was produced from the pulverized-fuel-fired boilers.

The four pulverized-fuel-equipped boilers at this plant were cut in on the main line over 90 per cent of the time during the months of February and March 1919, and when line banked no fuel was required.

Referring to various information as set forth in Mr. Harrison's paper, I would like to point out the following:

I can hardly agree with Mr. Harrison that the equipment for preparing pulverized coal has been developed past the experimental stage. There is still considerable to be done in this direction, particularly with respect to the elimination of the reabsorption of moisture and of entrained moisture, which results in condensation in the dry- or pulverized-fuel bins and the consequential trouble that it causes. The type and operation of drier and pulverizer has a great deal to do with these factors, and experience has demonstrated that certain fuels require special treatment in that regard or trouble is bound to occur.

From my experience, when pulverized fuel is used for steam generation the detrimental effect from the ash and sulphur content is nil, regardless as to the melting point of the former.

The economic use of coke breeze is at this time problematical, as the expense of preparing the fuel will, in a large number of cases, offset the fuel value of this by-product. A study should be made of each specific application before the use of coke breeze is gone into.

The utilization of the full thermal value of anthracite silt has been developed to a point where there are no commercial obstacles in its way, and this in itself will release an equivalent amount of commercial size of coal. The shipment of silt to a considerable distance from the mines, however, may not be economical, but the power consumption by the mining and nearby industries is sufficient for the full utilization of this by-product, which should make it unnecessary to ship the silt to other points.

The author's specific statement as to how the coal should be pulverized is not in conformity with my experience on this subject, as the degree of fineness should vary with the character of the fuel. The results of careful analyses and tests have demonstrated that this degree of fineness for effective and economical use cannot be arbitrarily set down, as it varies with the volatile content and the combustion characteristics of the fuel handled.

Commercially the expense of pulverizing increases with the degree of fineness, and consequently the coarser the particles of the fuel the less the cost chargeable against the pulverizing process, for which reason it is advantageous to use coarse pulverization whenever the character of the fuel is such that this is feasible.

From my experience the author's tabulated costs for pulverizing are somewhat misleading, as the cost for preparation will vary over wide ranges. While the information as set forth is valuable from a general standpoint, it must be used with care for the reason that it is not susceptible to specific adaptations.

In the case of open-hearth practice the system to be used for burning the pulverized fuel determines the degree of fineness. The high-velocity method of burning, such as described, will, of course, require much finer grinding than a low-velocity method such as obtains with other systems in which the time limit is increased, thereby permitting combustion to take place more slowly with the same thermal results.

By using a low-velocity flame the length of from 6 to 8 ft. mentioned by the author can be decreased.

Some of the steel companies have reduced their pulverized-coal consumption per ton of output to a little less than 450 lb. instead of 500 lb. as given in the paper.

It has been found in operating practice that pulverized-fuel-burning furnaces, under the best system of combustion, can be reduced to a little less than 1 cu. ft. of volume per b.hp. developed, and that the rating can be increased up to the capacity of the furnace to keep the lower tubes of the boiler free from honeycomb and still maintain effective combustion below the boiler-tube line.

The best system of combustion necessitates —

1 Proper preparation of the fuel
2 Effective means for feeding
3 Furnace properly designed and equipped with auxiliary air supply to insure a distinct hot zone and a distinct cold zone, and gas areas and baffles and draft coordinating with the foregoing.

The pressure at which pulverized fuel can be admitted to the furnace has been found, in service practice to be not more than plus 0.05 in. of water at the burner nozzle.

In my opinion a boiler cannot be successfully operated at any capacity with the natural draft obtained with a 30- to 35-ft. stack, the reason being that the friction losses of the gases passing through the boiler tubes, and the gas areas produced by the baffles, will reduce the draft, due to this low height. When operating at full capacity about 0.10 in. of water draft in the furnace, plus the friction losses through the boiler setting, is the minimum under which any pulverized-fuel boiler installation should be operated Anything less than this will limit the factor of regulation and will tend to cause constriction of heat to the furnace chamber and result in slag, which latter, in all cases, is to be avoided. Any pulverized-fuel-burning installation, in combination with a steam generator, in which the formation of slag is of any considerable consequence, can be taken as a failure from the standpoint of effective and economical results.

Under proper combustion conditions the use of a hand-lance steam jet to clean the bottom tubes as recommended by the author is unnecessary. The top of the tubes should, of course, be kept free from the accumulation of ash, which acts as an insulator and which tends to carry the heat in the ash and gases through the boiler to the stack.

While any ash accumulation in the bottom of the furnace should be periodically removed, at the same time with a proper design of hot-zone and cold-zone furnace any ordinary accumulation should not convert into slag.

In comparing pulverized equipment with stoker equipment, the author states that considerably less excess air is required with the former. The amount of air injected with the fuel into the furnace, however, should be a negligible percentage of that required for combustion, the only function of that air being a conveying and commingling medium. In the Edge Moor installation to which the author refers, the percentage of air entering with the fuel is about $1\frac{1}{2}$ per cent of that necessary for combustion, the remaining $98\frac{1}{2}$ per cent being induced by the stack.

Pulverized Fuel for Locomotives. From a study made of the coal measures in the southern part of Brazil during 1904–06, it was concluded that the native coal was unsuitable for economic use. Later, during 1915–16, at the direction of Dr. Miguel Arrojado Lisboa, then Director of Government-Operated Railways, an investigation was made of the use of pulverized fuel in the United States, with the result that the Central Railway of Brazil decided to install at 15-ton-per-hour capacity fuel-preparing and coaling plant and a stationary boiler equipment at Barra do Pirahy, an engine house and shop terminal about 65 miles north of Rio. Plans and specifications were prepared and installation was made of the "Lopulco" system by the International Pulverized Fuel Corporation of New York. Arrangements were also made for the equipping of 250 existing and new locomotives with the same system and by the same company, and since that time twelve ten-wheel-type and two Consolidation-type locomotives have been newly built and so equipped, by the American Locomotive Company, and put into regular use on the Central Railway of Brazil.

The first official run with Brazilian native coal, pulverized, was made on September 9, 1917, with a special train that transported Dr. Wenceslao Braz, President of the Republic of Brazil, and his staff. Ten-wheel-type locomotive No. 282 handled the President's special train from Barra do Pirahy to Cruzeiro, a distance of about 90 miles, and during the greater part of the trip President Braz remained in the locomotive cab and fired the locomotive, on which the steam pressure was fully maintained throughout, without any smoke.

As the result of this performance, President Braz sent a telegram to the Minister of Public Works, as follows:

> From Barra do Pirahy to Vargem Alegre, I traveled on ten-wheel locomotive No. 282, fitted for the use of pulverized fuel, with excellent results. The trip was made with a velocity of 63 kilometers per hour, having a train of 210 units behind it. I take great pleasure to give you this communication which I am certain will be received by all Brazilians interested as solution of one of your most important national problems. Salutation.
>
> WENCESLAO BRAZ.
>
> Barra Mensa, September 9, 1917.

The Central Railway of Brazil locomotives equipped with pulverized-fuel-burning equipment are operating in fast-passenger, mixed-passenger and freight, and freight service, and all are giving excellent results.

In tests recently conducted with Brazilian native coal and lignite the distance traveled during trials was 118 miles and the boiler pressure remained almost constantly at 175 lb., which is working pressure. The results of these tests may be found in Table 5.

The only difficulty met with has been in instructing the enginemen, who were not acquainted with this method of combustion, and for this purpose an illustrated instruction book has been issued to each man.

Only by adopting this pulverized-fuel system has the problem of the utilization of Brazilian fuel, which cannot be burned practically or economically on grates or in retorts, or utilized to good advantage for the production of producer gas, been solved, and the development of the native coal fields of the country is now in process through the establishment of steamship and railway means of transportation from the mines, and in the actual mining developments.

In the United States the development work in connection with the use of pulverized anthracite and bituminous coals and lignite for steam locomotives has been carried out by making applica-

PERCENTAGE ANALYSES OF FUELS USED

Name	Kind	Moisture	Volatile	Fixed Carbon	Ash	B.t.u. per lb.
Jacuhy	Bituminous	6.10	22.40	45.70	19.80	10,851
Santa Catharina	Bituminous	12.60	36.00	42.10	9.00	10,259
S. Jeronymo	Bituminous	3.00	31.00	39.30	26.70	9,565
Cacapava	Lignite	19.00	36.00	19.20	25.80	5,249

PERFORMANCE DATA

Fuel		B.t.u. per lb.	Quantity burned per trip, net tons	Evaporation, lb. water per lb. of fuel	Ash found in firebox and pan, lb.
Name	Kind				
Jacuhy	Bituminous	10,851	4.19	7.2	176
Santa Catharina	Bituminous	10,259	3.5	7.1	176
S. Jeronymo	Bituminous	9,565	5.419	7.1	198
Cacapava	Lignite	5,249	4.41	7.3	220

tion to single locomotives of different types which were distributed on five different railroads of the country in order to determine upon a composite and interchangeable pulverized-fuel-feeding, burning and furnace equipment that would be adaptable to any kind or size of steam locomotive, as well as to all possible fuels or combination of fuels locally available, and which at the same time would permit of the quick conversion from pulverized fuel to fuel oil, and vice versa.

When it is taken into consideration how many modifications of firebox, grate, ashpan, brick arch, smokebox draft appliances, exhaust nozzle and stack designs and equipments are required to adapt steam locomotives to the various anthracite and soft bituminous coals and lignites as used for fuel, even on a single railway, it can readily be imagined what the development of a single pulverized-fuel-firing mechanism and furnace arrangement for the entire United States has involved, particularly to make it adaptable to existing as well as new designs of locomotives. For example, the time required for the development and the practical use of fuel oil and of a satisfactory superheater is comparable.

During the past year the financial, labor and material conditions on steam railways, brought about by the war, have prevented any appropriations being made for the equipping of operating terminals and divisions in the United States for the extended use of pulverized fuel, but the result of what has obtained may be summed up in the following data applying to The Delaware and Hudson Company and the Atchison, Topeka & Santa Fé Railway:

On The Delaware and Hudson Company a newly built Consolidation type of freight locomotive, No. 1200, with a tractive power of from 61,400 to 64,000 lb., was equipped for experimental purposes, from March 1916 to August 1917, and operated in road freight service between Carbondale and Plymouth, Pa., and Oneonta, N.Y., on runs of from 37 to 94 miles one way. Pulverized fuel was supplied from The Hudson Coal Company's stationary-boiler experimental pulverizing plant at Olyphant, Pa.

This locomotive was designed for a working steam pressure of 195 lb., but the boiler was designed to carry 215 lb. steam pressure. With 195 lb. working pressure the cylinder horsepower rating is 2368 and the boiler horsepower rating 2540, giving a 107.2 per cent boiler.

Pulverized-fuel tests were made with the following adjustments:

Adjustment	Boiler pressure, lb.	Tractive power, lb.	Factor of adhesion	Results
Original	195	61,400	4.36	O. K.
First change	200	63,000	4.24	O. K.
Second change	205	64,600	4.14	O. K.
Third change	210	66,200	4.03	O. K.

The raw coal which was supplied for these tests analyzed about as follows:

Page 418, bottom table should read:

Content	Anthracite Slush	Anthracite Birdseye	Bituminous Slack
Moisture	1.30	0.50	1.67
Volatile	6.34	8.44	23.63
Fixed Carbon	65.70	62.66	65.16
Ash	27.96	28.40	13.21
B.t.u	10,500	10,100	13,671

This raw coal was mixed in the proportion of 60 per cent anthracite and 40 per cent bituminous, which, after drying and pulverizing, produced a fuel of from 15 to 20 per cent volatile content. This was entirely satisfactory for locomotive purposes and yielded an average of one boiler horsepower for each 1.4 sq. ft. of combined firebox and tube heating surface.

Dynamometer-car tests conducted to determine sustained pulling capacity on heavy grades and at starting gave the following results:

Maximum dynamometer drawbar pull, lb.	Speed miles per hour	Reverse lever cut-off, per cent	Throttle opening, per cent	Boiler pressure, lb.	Grade on line, per cent
64,000	At start	Full	75	200	1.65
59,000	6	66	Full	205	1.65
58,000	8	66	Full	205	0.72
56,000	10½	66	Full	205	0.72

During these tests a fuel mixture of 60 per cent anthracite birdseye and 40 per cent bituminous slack was used, and the apparent evaporation ranged from 7.3 to 9.3 lb. of water per lb. of coal consumed. The coal fired per 1000 ton-miles averaged 202 lb.

In heavy-tonnage-service runs — over ruling grades of from 0.72 to 1.65 per cent — for a distance of 37 miles the following data show typical performance:

Item	Trip No. 1	Trip No. 2
Miles run	37	37
Speed, average, miles per hour	14.5	13.1
Ton-miles, actual	83.147	85.758
Ton-miles, adjusted	88.553	90.113
Coal consumed per 1000 ton-miles	186	202
Steam pressure, average, lb	199	200

When in heavy-mine-run service between Carbondale and Olyphant, Pa., for the three months' period, March 13 to June 12, 1917, the performance of the No. 1200 was as follows:

Period		Days in road service	Hours in road service
From	To		
1917	1917		
March 13	April 12	28	301 hr. 3 min.
April 13	May 12	27	301 hr. 30 min.
May 13	June 12	25	273 hr. 10 min.
Total....	....	80	875 hr. 43 min.

After the day's work, upon arrival at the Carbondale engine terminal, the locomotive would be run directly into the house, no fire, track or ashpit delays or work being required.

On the Atchison, Topeka & Santa Fé Railway an existing Mikado type of freight locomotive, No. 3111, with a tractive power of 59,600 lb., was equipped for experimental purposes — from May 1917 to July 1918, and operated in road freight service between Fort Madison, Iowa, and Marceline, Mo., on runs of 112.7 miles one way. Pulverized fuel was supplied from the company's experimental pulverizing plants at these points.

Dynamometer-car tests were run with the following average results, using Frontenac, Kan., run-of-mine bituminous coal, averaging in analysis when pulverized as follows:

Moisture, per cent	1.05
Volatile, per cent	32.67
Fixed carbon, per cent	51.57
Ash, per cent	14.71
Sulphur, per cent	3.95
B.t.u. per lb	12,022
Per cent through 100-mesh	97.8
Per cent through 200-mesh	82.6

The general performance of the locomotive equipped with the Lopulco pulverized-fuel system was as follows:

Date of runs	Mar. 4 to Mar. 22, 1918
Total trips run (112.7 miles each)	14
Total miles run	1578
Average running time	5 hr. 6 min.
Average speed, miles per hour	22.3
Average train tonnage, net tons	2273
Average gross 1000 ton-miles	256.5
Average coal per gross 1000 ton-miles, lb.	82.4
Average water per gross 1000 ton-miles, lb.	566
Average boiler pressure, indicated, lb	188
Average feedwater temperature, deg. fahr	48
Average flue-gas temperature, deg fahr	553
Average smokebox draft, inches of water	11.3
Average firebox draft, inches of water	1.3
Average quality of steam, per cent dry	96.0
Average superheat in steam, deg. fahr	233
Average lb. of coal per hour of running time, per equivalent sq. ft. of grate area	71.3
Average lb. of coal per hour of running time, per sq. ft. of boiler heating surface	1.01

FUEL PERFORMANCE

Equivalent evaporation, lb. of water from and at 212 deg. fahr. per lb. of coal for boiler and superheater	9.22
Per boiler horsepower for boiler and superheater	11.15
Combined efficiency for boiler and superheater, per cent	74.5
Thermal efficiency for locomotive, per cent	4.19

An actual evaporation (not corrected for the quality of steam) showed at the rate of 8.46 lb. per sq. ft. of boiler heating surface.

The combined boiler and superheater efficiency showed a gain of 23.2 per cent for pulverized fuel as compared with hand-firing.

Based on the hand-firing performance, the use of pulverized fuel showed a saving of 22.5 per cent in fuel. The combustion was practically smokeless and the pulverized-fuel operating mechanism gave no trouble.

Fig. 1 shows a typical application of a pulverized-fuel-burning furnace as adapted to a modern steam-locomotive type of boiler for the use of any solid fuel having 15 per cent or higher volatile, or for fuel oil, regardless as to the other chemical characteristics.

Pulverized Fuel in Marine Service. During September 1918 quite a number of tests were made on the USS *Gem* (SP-41) on Long Island Sound to determine what results could be obtained from the use of Navy Department specification coal in pulverized form as compared with oil and other fuels.

One of the two Normand type of water-tube boilers was equipped with two pulverized-fuel feeders and burners, the furnace being fitted up with firebrick in a manner that would enable the use of either pulverized coal or fuel oil, or a combination of both. On account of the boiler not being equipped with induced draft the pulverized-fuel induced-air burners were connected to the regular air-blast system instead of being open to the atmosphere.

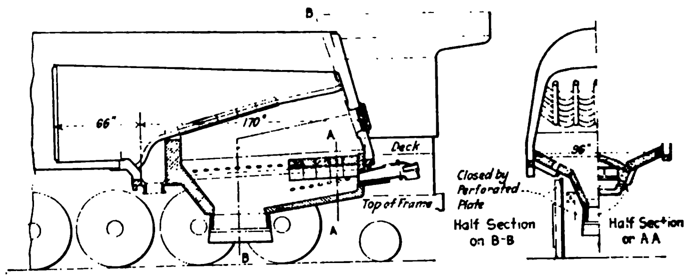


FIG. 1 TYPICAL APPLICATION OF A PULVERIZED-FUEL-BURNING FURNACE TO A STEAM-LOCOMOTIVE BOILER

The characteristics of the coal used, after pulverization, were as follows:

Moisture, per cent	1.02
Volatile, per cent	18.70
Fixed carbon, per cent	75.10
Ash, per cent	5.18
Sulphur, per cent	0.65
B.t.u. per lb	14,975
Fineness:	
Percentage through a 200-mesh screen	83.6
Percentage through a 100-mesh screen	92.0

On September 18, 1918, a four-hour run was made, from 8.30 A.M. to 12.30 P.M., during the last two hours of which the most economical speed for the ship, i.e., 16 knots per hour, obtained, and which enabled the engines to take the steam as fast as generated. This speed was maintained for the two-hour period, and then, as during the entire four-hour test run, the furnace operation was good — there was no heat effect on the refractory — and there was either no smoke, or it was very light; there was no accumulation of slag or ash on the boiler tubes or settings, and had the boiler been equipped with induced draft the efficiency results would have been better and the light smoke that was produced would have been entirely eliminated.

The log of the test for the last two hours of this four-hour continuous run was as follows:

TEST NO. 23, SEPTEMBER 18, 1918

Time fired up	8.30 A.M.
Time of test	10.30 A.M. to 12.30 P.M.
Duration of test, hours	2
Average speed, knots	16 approx.
Average boiler pressure, lb.	210
Average superheat, deg. fahr	67.5 (max. 86)
Average flue-gas temperature, deg. fahr	541 (max. 570, min. 500)
Average indicated horsepower	751
Average revolutions per minute	261.5
Total pulverized fuel fired, lb	3,235
Coal per i.hp.-hr., lb	2.15
Total water evaporated from and at 212 deg. (on basis of 25 lb. per i.hp.-hr.), lb	37,550
Water evaporated per lb. of coal from and at 212 deg., lb.	11.6
Boiler efficiency, per cent	75 approx.
CO_2 per cent	Avg. 13.5, max. 14, min. 13
Coal	Pocahontas Bituminous
Lepulco system equipment operation	Good
Lepulco system furnace operation	Good
Smoke	Light
Brickwork heat effect	None

This same boiler when using fuel oil of about 18,500 B.t.u. and when operating at a speed of 14 knots, develops an indicated horsepower-hour on 1.68 lb. of such fuel. Therefore, with coal of 14,975 B.t.u., in order to give equivalent results it should use 2.08 lb. per i.hp-hr.; whereas the performance at 16 knots shows 2.15 lb. per i.hp-hr.

A comparison of superheat as obtained with straight pulverized fuel and with fuel oil, respectively, on various test trips follows:

Pulverized Fuel			Fuel Oil		
Trip No.	Average knots	Average superheat deg. fahr.	Trip No.	Average knots	Average superheat deg. fahr.
22	10	43	14	10	45
17	12	73	16	12	70
23	16	66	24	14	58

Edwin Lundgren, who opened the oral discussion, called attention to the large combustion space specified in Mr. Harrison's paper for a boiler burning pulverized fuel as compared to a stoker-fired boiler, especially when the boiler was to be operated at, say, 300 per cent of rating. In the matter of draft, he thought Mr. Harrison had not taken the resistance of the boiler into consideration in specifying but 0.10 to 0.15 in. For the ordinary type of boiler a draft of about 0.15 in. was required at 100 per cent rating, which increased to about 0.7 in. for 300 per cent rating. A stack 100 ft. high would give about 0.6 in. draft, so that a height of 35 ft. was out of the question.

As to the low efficiency (63 to 65 per cent) of stoker-fired plants mentioned in the paper by Messrs. Scheffler and Barnhurst, he would cite the Detroit Edison plant where an average efficiency of

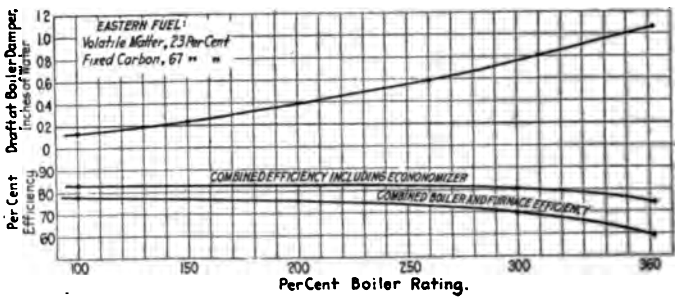


Fig. 2 Boiler Efficiency Curves, Cincinnati Gas and Electric Co.

76 per cent was maintained throughout the year, and where Dr. Jacobus' series of 24-hour tests showed efficiencies of 80 per cent or better. Also, the Cincinnati Gas & Electric Co. maintained a combined boiler and furnace efficiency of about 80 per cent in their plant constructed to maintain 365 per cent maximum capacity and 300 per cent continuous capacity.

The efficiency curves are shown in Fig. 2.

With regard to combustible in the ash, he said, the plant of the Boston Edison Co. had a record of as low average percentage of combustible as 8 and 10 per cent and the Detroit Edison plant as low as 11 per cent, under daily operating conditions.

There was no question but what pulverized-fuel equipment would be useful in burning some of the cheaper grades of fuel, like some of the western fuels, for which it seemed no real type of stoker had been designed. He did not believe, however, that it would be generally applied to modern central stations, for one reason because the pulverized-fuel-equipment buildings, to judge from the slides shown, appeared to be about as large as the power stations proper.

C. F. Hirshfeld said that in modern boiler plants the chief losses were due to carbon in the ash and to heat escaping up the stack. The Detroit Edison Co. operated under good conditions with 10 per cent of carbon in the ash, or a very small per cent of the fuel originally fired, and with but 10 to 12 per cent of excess air. If use of powdered fuel would make it possible to burn all the carbon in the coal — which was contrary to his experience — and with no greater amount of excess air, the saving effected would still be of little consequence. He believed in using powdered coal where conditions were such that it was distinctly economical to do so, but these did not appear to obtain at the Detroit plants. In his opinion there was greater chance for improvement in the turbine end of the plant than in the boiler room.

As to greater uniformity of flame and temperature, which conduced to longer life of the furnace lining, he believed that the advantage was with the stoker with its smooth bed of incandescent fuel. While powdered fuel might be more adaptable when a wider range of fuels had to be used, the stoker seemed just as adaptable so far as the art had developed.

In the paper by Messrs. Scheffler and Barnhurst the power for operating a stoker was stated to be 2 per cent of the total boiler hp. In the Detroit Edison Co.'s plant, however, it was much less than 1 per cent with the boilers operated at 200 per cent of rating.

H. Wade Hibbard said that in a larger plant he had in mind an increased efficiency resulted from adding moisture to coal that had dried out too much before it reached the stokers. He therefore asked whether drying was necessary in order that the coal might be thoroughly pulverized, and also whether there had been any experiments performed in introducing steam along with the powdered coal. Affirmative replies were respectively made to these queries by Messrs. Scheffler and Barnhurst, the latter adding that he did not know whether the introduction of moisture had increased the efficiency.

P. A. Poppenhusen, referring to a statement made by Mr. Lundgren, said that his company had successfully adapted the chain-grate stoker to the burning of lignites of all grades and containing as high as 30 per cent of moisture, and that 24-hour tests had shown efficiencies of from 76 to 78 per cent. The power to operate a stoker was insignificant — less than 0.5 hp. per unit, whether a 200-hp. or a 600-hp. unit. He thought Mr. Scheffler's figure of $64,000 low for the cost of a pulverizing plant of 100 tons daily capacity, and asked how large a plant should be in order that this cost would not be prohibitive.

JOHN VAN BRUNT, calling attention to Mr. Harrison's reference to coke breeze, said he could state that over 75,000 hp. of boilers equipped with traveling-grate stokers were now satisfactorily operating on this waste fuel in the large steel plants and gas works of the country.

GILBERT A. YOUNG spoke of successful experimental work carried on for the past two years at Purdue University on a furnace for burning crushed, undried coal containing as high as 20 per cent moisture. The particles after crushing ranged from the size of a match head down to powder passing a 200-mesh screen. He hoped to submit a paper later giving comparative tests of the crushed-coal furnace and several types of stokers, all operating under the same conditions.

W. P. FREY[1] said that his company burned 1,000,000 tons of coal a year for steam and electric power, and that he was interested in pulverized fuel as a possible means of utilizing the twenty-odd million tons of small coal lying on the ground in the anthracite regions. But as it was said to take twice as much power to pulverize anthracite as bituminous coal, and as the minimum wage for the anthracite union laborer was 43 cents per hour, this would make the cost of pulverizing the $1 coal in question about 75 cents, and he did not see how pulverizing would make up for this enormous difference. Firing a boiler at a high rating with anthracite required far more draft at the damper than the 0.10 to 0.15 in. mentioned in Mr. Harrison's paper, and in consequence stacks much higher than 40 ft. The fire-brick problem was a very important one, and he would be obliged if Mr. Scheffler or Mr. Barnhurst would give specifications for bricks that in the long run would stand the steady temperature of over 3000 deg. that they would have with their

[1] Fuel Engineer, Lehigh Coal & Navigation Co., Lansford, Pa.

low supply of air, unless they reduced the temperature in the combustion chamber.

W. L. Wotherspoon referred to tests he had made ten years ago in South Africa on a Bettington boiler fired with pulverized fuel and the comparative results obtained between it and a B. & W. boiler with a chain-grate stoker. With boilers of about 1000 hp. the overall efficiencies obtained were practically equal. Low-grade fuels of about 9500 B.t.u., of which large quantities were available in the Transvaal, were used, and the results obtained were such that some of the Bettington boilers installed at that time are still in use.

There are several in use in England, and two or three have been in continuous use in Nova Scotia, Canada, for about five years.

More recently he had been paying attention to the use of pulverized fuel in connection with smelting in blast furnaces, where combustion takes place under pressure. Development work had reached a state where it had been able to carry on a continuous test for eight days and smelt the refractory copper-nickel ores of the Sudbury district with 50 per cent of the normal coke used replaced by pulverized coal.

At the smelter where these tests were made coke cost approximately twice that of bituminous coal. He also referred to tests at The Tennessee Copper Company's smelter, where pulverized coal had been used in the smelting of copper ores in the standard blast furnace.

Continuous tests had been made for periods of ten to twelve days with practically all the coke replaced by pulverized coal and, in addition, it was found the total weight of fuel used was about 25 per cent less.

Regarding the cost for drying and pulverizing coal, the prewar cost at The International Nickel Company's smelter in Ontario, Canada, was 40 cents per ton, but owing to the increased cost of labor, repairs and supplies due to the abnormal conditions now existent, the costs were practically double.

Referring to the transmission of pulverized fuel, the screw-conveyor had been commonly adopted in the past, but compressed air was now being used to a greater extent, and he had recently carried out experiments in Canada on the transmission of fuel by compressed air and found it possible to convey 2½ tons in five minutes through a 3-in. pipe, over a horizontal distance of 1200 ft., and then elevated 60 ft. These experiments were made in order to prove the practibility of utilizing compressed air for transmitting about 50 tons of fuel per day from the pulverized fuel plant at the reverberatory furnace department to the blast furnace in a separate building at considerable distance.

The air pressure used was 70 lb. and the results were quite satisfactory and were in use for several months. This method might prove advantageous for the transmission of pulverized fuel from a central pulverizing plant to several isolated power plants by means of underground pipe lines.

W. E. Snyder felt that while pulverized coal was an ideal fuel in the cement plant, there were, nevertheless, many problems to be solved in regard to its use in steel furnaces and under boilers. The best modern stoker-fired plants showed very high efficiencies, and with the same expenditure of intelligence the pulverized-coal plant might be brought to equal them.

He had experienced difficulty in obtaining definite facts regarding performances of pulverized-fuel installations, and was skeptical as to the accuracy obtained in measuring the coal by observing the revolutions of the feed screw, for it did not always give the same quantity per revolution. Apropos of accurate measurements, Mr. Snyder caused considerable amusement by telling of the man in charge of a small power plant who had succeeded in overcoming troubles he had experienced with a venturi meter by merely boring out the narrow part!

Narrowing down the comparison of the two methods of getting coal into the boiler furnace, he said, it resolved itself into a very simple matter of reducing waste in the ashpit and up the stack, and a possible saving in repairs and maintenance of the means employed to handle the fuel.

W. F. Verner gave estimates that had been prepared covering the comparative costs of operating six 2400-hp. boilers at the Ford Motor Company's blast-furnace plant with pulverized fuel and with stoker fuel. The estimated cost for the pulverizer equipment and buildings was $691,000 and for a corresponding stoker plant $475,000. The cost of pulverizing per ton, including fixed charges, power, maintenance, lubricants, etc., and labor (at $8 per day) was $0.76; for transmission from pulverizing building to boiler room, $0.25; and for boiler room, $1.13; or a total of $2.14. For a corresponding stoker plant the figures were: for transmission from breaker building, $0.24; boiler room, $1.66; total, $1.90. For a plant with twelve 2400-hp. boilers the total for the pulverized-coal installation was $1.63, and for the stoker equipment, $1.49. These figures, in connection with the higher estimated efficiency of the pulverized-fuel plant, indicated for it a saving of 4 per cent over the stoker plant.

One important point in favor of pulverized-fuel plants was that the stand-by losses were reduced to a minimum as compared with stoker installations, where the fires had to be banked over the shut-down periods. An additional reason for installing the former was that blast-furnace gas would be available for use under the boilers and in quantity sufficient to carry the light loads.

John A. Stevens thought that the matter of stand-by losses brought out by Mr. Verner a very important one. In New England factories running on one shift of 9 hours a day, the stand-by coal required in the best hand-fired plants ranged from 20 to 25 per cent of the daily consumption. This, of course, decreases with greater number of shifts, or with continuous runs of say 24 hours, as in paper mills. With stoker fired boilers there is a loss in running full load right up to quitting time, and also in burning a bank of coal.

R. Sanford Riley said that the papers and discussion were valuable in directing attention to the problem involved in all combustion—the carburization of air. The final question regarding the two systems of burning fuel under consideration was which would prove the more satisfactory under the long, hard test of average conditions, and he felt that in the long run the simpler apparatus would win out.

Edward N. Trump spoke of experiments he had made about 25 years ago in burning powdered fuel under a 300-hp. B. & W. boiler. The furnace used had too small a combustion chamber, and the ash accumulated and blocked the passages between the tubes in less than a week. This ash had taken up tar from the combustion products and clung to the tubes, and it proved very difficult to scrape off. It was important, therefore, to see that the combustion chamber was large enough, and, as noted in Mr. Harrison's paper, that the boiler tubes and surfaces were carefully and frequently blown.

H. G. Barnhurst, in closing the oral discussion, said that there were certain arguments in favor of pulverized coal for certain localities and for certain grades of fuel which could hardly be overcome by any other method of burning. One point which had not been brought out was the fact that by pulverizing coal and burning it in a furnace the boiler installation was made entirely independent of any particular quality of fuel. In the course of experimenting his company had burned coal with as low as 2 per cent of volatile matter and up to 40 per cent with an ash content running as high as 51 per cent, just to show that the percentage of ash did not affect or interfere with the combustion conditions. Mr. Trump's main troubles, he thought, were due not only to the small combustion chamber used, but also to the fact that his coal was not finely pulverized. He wished again to emphasize the fact that the coal must be pulverized to a very high degree of fineness in order to get good results — two or three hundred million particles to the cubic inch. This increased the surface exposed to the air about 700 times over that which it was in the form of lumps, and made it very much easier to effect combustion. Moreover, various grades of coal could be burned which could not be handled on stokers.

As to the volume of a furnace per pound of coal burned per minute, he would not care to assign any particular value, because all combustion chambers had different shapes. The essential feature was low velocity of the gases, for the brick became plastic at high temperatures, and if subjected to the action of gases at a high velocity, the under rows of lining would be destroyed.

The question of furnace firebrick was, of course, very important, and the higher the grade used the better would be the results. The ash accumulation in the bottom of the furnace by further refinements might be cared for by movable hearths so that it would not build up in the furnace and interfere with the operation. The ash going out of the stack, in some cases, was now being recovered to a certain extent, and there was certainly a field for recovering ash not only from pulverized-coal-fired furnaces, but also from stoker-fired furnaces, because the percentage of ash from many stoker installations, under forced-draft conditions, rose very high.

Fred'k A. Scheffler and H. G. Barnhurst. Mr. Marsh prefaces his remarks with a reference to the desirability of utilizing the by-products from coal. It appears to the authors that this is not pertinent to the discussion and that it will not require consideration until it becomes common practice to remove the by-products from coal before firing on hand grates or stokers. The inference is that pulverized coal cannot be burned after removing the volatile constituents, a supposition which is not borne out by the facts. His statements that pulverized coal is restricted to a field comprising forms that cannot be burned on stokers is an admission that pulverized fuels increase the latitude of combustibility and is not supported by evidence.

In reply to Mr. Marsh's objections to the tests submitted, it should be noted that with pulverized fuels uniformity of operation is obtained and maintained in a short time after firing the furnaces, and the variation in the figures need not be expected by prolonging the tests. In particular in the test made May 18, 1919, on one of the 600-hp. B. & W. boilers at the plant of the Puget Sound Traction, Light and Power Co. at Seattle, during which both the coal and water were weighed, the duration of the test being 13¾ hours, the efficiency obtained was 78.99 per cent with coal containing 10,106 B.t.u. per lb. as fired.

In regard to the figures assumed for stoker efficiency, it is submitted that the best information we have been able to obtain shows that an average efficiency of 65 per cent may be expected in modern power houses. Where power is not produced for sale this figure appears to be too high. Mr. Marsh does not substantiate his objections to these figures.

In regard to reliability, it is customary to duplicate a sufficient proportion of the pulverizing equipment to obviate irregular operation in pulverizing plants. This is taken care of in the figures on investment charges given in the paper and makes unnecessary Mr. Marsh's assumption that the investment charges quoted are too low.

In reply to Mr. Marsh's question as to the latitude of pulverized-coal furnaces, it should be stated that a furnace built for high volatile bituminous coal will not handle anthracite or coke breeze. Where a considerable variation in the grade of available fuels is to be expected, however, the furnace may be designed to burn anthracite or coke breeze and will then be able to handle any grade of bituminous fuel with slight changes in the arrangement of the burners. It is a matter of surprise to the authors that the tests mentioned by Mr. Marsh show that the fire was unresponsive with low-volatile coals, since it has been their experience that the flexibility of all grades of fuels when pulverized exceeds that obtained when burned in lump form. The burner for pulverized fuel is very simple and has a wide range of adjustments, easily meeting any requirements made upon it by a change in the fuel.

In regard to adaptability, the effect of ash in coal is to carry away a small amount of heat as the percentage increases. The excess air required to burn fuels of the highest ash percentage is, however, negligible. It may be stated in regard to the varied efficiencies shown in the tests that pulverized-fuel furnaces require proper design as well as any other type, and that tests in many instances were a part of the experimental development and variations were to be expected.

In regard to the burning of high-ash coal on chain grates, it should be stated that the 28.4 per cent ash fuel mentioned by the authors was tried out on chain grates and was a failure. The stokers were taken out after they had been proved incapable and return was made to hand firing.

In regard to flexibility it is readily admitted that pulverized fuel requires large furnace volumes to burn successfully. It should be noted, however, that other types of fuels, including oil, are burned more efficiently with large furnace volumes, and that the best new stoker installations have furnace volumes as large as required for pulverized coal. The amount of heat that may be developed in a furnace with pulverized fuel is restricted only by the resisting powers of the refractories up to this maximum. The flexibility of control far exceeds that obtained with any other method of burning solid fuels. The upper limit is merely a matter of design and rating.

Under the head of furnace design Mr. Marsh again protests against the inclusion of tests of four or five hours' duration. This has been answered above.

Mr. Marsh objects to the figures used for moisture percentage in coals. It may be stated that our tables were based on a general average moisture content for the entire country. Further, the figures given for pulverizing cost are the result of present information and the best obtainable. They will not vary greatly except under unusual conditions.

We must take exception to Mr. Marsh's statement that three-quarters of the ash goes out of the chimney. Generally speaking, 20 to 25 per cent will remain in the combustion chamber and 30 to 40 per cent in the second and third pass, and the balance in the flues and out of the stacks. Very little settling takes place after the gases leave the boiler. With coal containing 10 to 15 per cent ash the stack discharge is not objectionable, even in cities. With higher ash content it might be necessary to provide dust collectors n the flues in city power plants.

In answer to Mr. Marsh's question, pulverized coal is stored in closed tanks, absorbs moisture very slowly, and under the methods of installation in use does not stay in storage for any considerable period.

Mr. Marsh criticizes the authors' claims for range of fuels, and follows by asking of what value the capacity to handle both anthracite and lignite may be. It is self-evident that with such a range as this many of the intermediate fuels may be burned. His statement that the cost of drying and pulverizing fuel in installations of from 500 to 600 tons daily capacity is over $1 per ton is at variance with figures obtained from hundreds of plants and upon which our estimates are based.

Mr. Diman is correct in stating that there is a limit to the quality of low-grade fuels that can be used in pulverized form. There is no doubt that if the coal is too low in combustible the cost of grinding would overcome any advantage obtained in economy of burning these coals in pulverized form. Success in burning anthracite depends upon the form of the furnace. As the coal lacks, volatile, increased temperature must be generated near the burner, and this is accomplished by the use of some of the products of combustion from the furnace itself. This means designing the furnace so that there is a returning flame to ignite the incoming fuel. With this arrangement very satisfactory operating efficiencies are being obtained, and there is no trouble in starting a furnace so designed in but a little more time than that required with pulverized bituminous coal.

In Mr. Peabody's discussion of Mr. Harrison's paper he states that oil is undoubtedly superior to all other fuels for boiler purposes, and that the furnace volume required for oil fuel is only about one-quarter of the furnace volume specified in the paper for pulverized coal. Mr. Harrison recommends a larger combustion chamber than we know is actually required, but we have found that where oil is fired in pulverized-coal furnaces much better efficiency is obtained than is obtained in furnaces designed for oil alone. Larger combustion chambers are constantly being installed under all boilers for the purpose of obtaining better combustion conditions. This is due to the fact that the best mixtures of the air and carbon or coal cannot be made with the present method of oil or lump-coal firing.

The slight extra cost of the larger furnaces required is more than overcome by the increase in efficiency obtained over other methods of firing. It has been mentioned previously that combustion chambers recommended for pulverized coal are no larger than those now being used in a great many stoker installations.

Mr. Hirshfeld has probably not examined very many pulverized-coal-fired furnaces, or he would have found that there is hardly any trace of combustible in the ash in the furnace. We do not agree with his statement that a more uniform flame and temperature can be obtained with stoker practice due to the smooth bed of incandescent fuel. A fuel bed is a very uncertain quantity in the average furnace. It cannot remain of constant thickness or of constant quality. The conditions of the fuel bed are beyond control, and the coal must frequently be redistributed by the fireman. Certain types of stokers must be watched constantly and the holes in the fire bed kept covered with fuel.

In reply to Mr. Hibbard, the authors would state that drying of the coal is necessary for the purpose of properly pulverizing it. The injection of water or steam into the firebox is a dead loss, and is resorted to only as a means to overcome defects in the method of burning fuels. There is no occasion for this with pulverized fuel.

The experiments mentioned by Professor Young have no connection with the subject of pulverized-coal burning, because the quality of the material with which he has been experimenting is thousands of times coarser than the standard pulverized coal as now being used for commercial purposes.

Mr. Frey takes up the cost of pulverizing coal, basing his statements on data obtained from a first-class power house. We are doubtful, however, even in this power house, as to whether the evaporation from the coal that he is using would average 6 lb. of water on the average over a year's operation, and if the evaporation was increased only 50 per cent, certainly the increased efficiency obtained from the fuel would more than counterbalance the extracost due to pulverizing unless the coal is obtained for practically nothing.

It is evident that the lower the price of the coal the larger must be the plant and the lower the cost of preparation to show economies warranting its application, but the cost of coal is going up steadily on account of the increased demands for fuel for manufacturing purposes.

Mr. Wotherspoon's remarks had no bearing on the subject of the application of pulverized coal to boiler-firing work, with the exception of his reference to the Bettington boiler experiments which were made a number of years ago, and which were an entire failure so far as practical operation was concerned.

Mr. Snyder's remarks are noted with interest. The question of the adoption of pulverized coal to power houses is only warranted where the cost of operation would show a fair interest on the investment. We admit that stoker plants are operated with high efficiency under careful management, but we believe that pulverized coal will show increased efficiency and lower cost of operation than the average stoker installation, particularly in plants using a large enough quantity of coal to permit obtaining a low cost of preparation.

Mr. Riley's remarks are particularly gratifying in that he fully realizes that the question of the adoption of pulverized coal is a matter of dollars and cents, and that its application in any installation is a question of which system will give the best returns for the money invested.

Mr. Trump spoke of experiments made in past years, and the writers particularly wish to point out that the furnace was too small for the boiler which he used, or else the quantity of coal fired to the furnace was beyond its capacity for satisfactory service. Furthermore, in the earlier days the fineness of pulverization was so much coarser than that now used in standard practice that there is no comparison, and the results obtained could not possibly have been satisfactory enough for commercial purposes.

Mr. Cary is correct in his statement that the idea of the use of pulverized coal has been in the minds of engineers for many years. Many experiments and thousands of dollars have been spent in an endeavor to accomplish this purpose. It has, however, been an evolution, and the earlier disastrous results were strictly due to the fact that sufficient knowledge of the subject was not at hand. The engineers did not realize that it was not the pulverizing of the coal that caused the trouble, but the fact that they tried to burn this kind of fuel, which is nearly in the form of a gas, in a furnace designed for lump fuels.

We would like to mention in regard to Mr. Cary's statements that today the fineness of pulverized coal is such that it practically all passes through the 50-mesh sieve which means that the largest particles are less than 0.01 in. in diameter. The velocities are slow enough to insure the complete combustion of the particles of this size, hence it is not necessary to go to the extra cost of pulverizing to any higher degree of fineness than is now practically the standard, which equals 95 per cent through a 100-mesh sieve and 85 per cent through a 100-mesh sieve, all passing through the 50-mesh.

In closing, the authors wish to express their appreciation of the attention and interest accorded them, as evidenced by the discussion. It is not their wish to present the pulverizing of fuels as a modern panacea, but merely to lay before the Society a statement of its possibilities and to remove some of the fallacies in regard to it which have had more or less general circulation.

N. C. Harrison. I note that Mr. Marsh states in his discussion that I have used pre-war prices for the cost of pulverizing. He has cited the cost used in Table 1, whereas he should have used the cost shown in Table 3, or rates of 40 cents for millers and 35 cents for laborers and $5 a ton for coal. These are not pre-war prices, but are the present-day prices as paid by the Atlantic Steel Company.

Mr. Marsh also speaks about pulverized coal picking up moisture when stored. I thoroughly agree with him on this, but every one who uses pulverized coal has found by experience that this fuel must not be stored. It must be kept moving at all times in the bins, consequently there is very little moisture picked up by the pulverized coal from the air.

I note in several discussions that considerable comment has been made regarding the height of stacks necessary for the use of pulverized coal on boilers, as spoken of in my paper. During February I visited one plant installed in Kansas, and saw boilers working there very satisfactorily with a stack of about 35 ft. in height.

Most of the comment has been that we should have a draft of 0.10 in. in the combustion chamber and a draft of from 0.20 in. to 0.60 in. at the damper. This has not been my experience for the best burning of pulverized coal under boilers. I believe that we should have practically a balanced draft in the combustion chamber and first passage through the tubes, and from 0.10 in. to 0.20 in. at the damper. These figures are based on 150 per cent of rated capacity. Of course, if the boiler is intended to be worked at a larger rating than this, it would be necessary to have larger stacks.

No. 1703

ECONOMY OF CERTAIN ARIZONA STEAM-ELECTRIC POWER PLANTS USING OIL FUEL

By C. R. Weymouth,[1] San Francisco, Cal.
Member of the Society

In certain steam-electric power plants the combination of load factor and high fuel cost not only necessitates but also makes possible the attainment of high economies in operation. These particular conditions exist in three Arizona plants known and located as follows: Inspiration Consolidated Copper Company, Inspiration, Ariz.; New Cornelia Copper Company, Ajo, Ariz. and Arizona Power Company, Clarkdale, Ariz. All of these plants embody many similar features and in this paper the author discusses the economy of operation of each. Tables and curves of operating characteristics are also given and these show that the best economy per bbl. of oil obtained at the Inspiration Plant was 294.5 kw-hr., at the New Cornelia Plant 326.2 kw-hr., and at the Arizona plant 333.3 kw-hr. All these values were obtained during the winter months when conditions were most favorable. The paper concludes with a section dealing with the personnel responsible for the installations.

IN certain Arizona steam power plants the combination of favorable load factor and high fuel cost has not only necessitated but has also made possible the attainment of high fuel economy, and even in plants where cooling ponds are used for condensing purposes. This paper refers to the performance of three such power plants, namely, those of the Inspiration Consolidated Copper Company, Inspiration, Ariz.; the New Cornelia Copper Company, Ajo, Ariz.; and the Arizona Power Company, Clarkdale, Ariz. These plants embody many similar features. They differ, however, in methods of condensing, as cooling ponds are used at the Inspiration and New Cornelia plants, and water from the Verde River at the Arizona Power Company's plant.

2 The Inspiration plant was designed in the winter of 1913–1914. The International Smelting and Refining Company's smelter adjoins the Inspiration power-plant site, and the steam generated in waste-

[1] Chief Engineer, Chas. C. Moore & Co. Engineers.

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

heat boilers in the smelter is utilized for the operation of the reciprocating blowing engines, which are designed for about 175 lb. steam pressure. These blowing engines are located in the same power-plant building as the steam turbines, and since the waste-heat boilers are connected to the same steam header as the oil-fired boilers their steam pressure, and the economy of the turbine plant have been limited by the common steam pressure of from 175 to 185 lb. The New Cornelia Copper Company's plant was designed in the winter of 1915–1916, and being independent of blowing engines, the boilers were selected for 250 lb. pressure. While a higher steam pressure would have been possible, the remote location of the plant and experience at the date of design led to the lower boiler pressure being selected. The Arizona Power Company's plant was designed in the winter of 1916–1917, and has boilers for 250 lb. pressure.

3 The Inspiration plant is 25-cycle, 3-phase, 6500 volts. The New Cornelia plant and the Arizona Power Company's plant are both 60-cycle 3-phase, 2300 volts. The maximum load at the Inspiration plant was estimated to be 12,000 kw.; three 6000-kw. Curtiss turbines were therefore selected, thus giving one spare turbine. For the New Cornelia Copper Company's plant the load was estimated to be 7500 kw., and this led to the selection of two 7500-kw. turbines, one unit being a spare. The Arizona Power Company's plant was designed as an auxiliary to a hydroelectric system, and intended to carry a peak load of 5000 kw. Owing to the quick shipment required, however, a turbine previously ordered for another company, and rated at 6000 kw., was installed, whereas the remaining equipment was selected only for a 5000-kw. load.

4 All three plants have Stirling steel-encased boilers, with Peabody-Hammel oil furnaces, Green fuel economizers, Moore automatic fuel-oil regulating systems, Wheeler surface condensers, Wheeler dry vacuum pumps, centrifugal hotwell pumps, direct-connected exciters, steam-driven boiler-feed pumps, etc. Superheaters are installed in all plants, specified to give 100 deg. superheat for the [illegible] and 150 deg. superheat for both the New Cornelia [illegible] Power Company's plants, all measured at the boiler. [illegible] All the plants also have [illegible] pumps, [illegible] being turbine- [illegible]

[illegible] Copper [illegible] com- [illegible] arrangement.

The actual vacuum shown by the operation of this plant, however, has been a disappointment, as a pond of insufficient area was installed. Since the design of the Inspiration plant this detail has been corrected by the addition of cooling towers. The condensate is returned to the oil-fired boilers, and make-up water for the pond is purified by a Booth water softener. A Cochrane hot-process purifier purifies the make-up for the oil-fired boilers if needed, but it is primarily for purifying boiler feed for the waste-heat boilers installed at the smelter. The blowing engines operate at slightly lower vacuum than the turbines, and advantage is taken of this fact to heat slightly the turbine condensate by passing it through the Volz heaters in the surface condensers of the blowing engines.

6 Cole-Bergman water weighers and Lea recorders are installed for feedwater measurement, for computing the steam supplied from the waste-heat boilers and the steam required by blowing engines, as well as the steam consumption of the turbines. Steam-flow meters are also used for checking purposes. By this means a separate record is kept of the economy of the turbine plant on the basis of operation independent of the blowing engines.

7 The feedwater for the New Cornelia plant contains from 30 to 50 grains of impurities per U. S. gallon. The water is low in calcium sulphate and carbonate, high in sodium sulphate, and very high in sodium chloride. It is not practicable to purify this water by chemical treatment. The condensate from the condensers is returned to the boilers, and raw make-up water is used for the pond and boiler-feed make-up purposes. Frequent blowing down is required for the cooling pond, as well as for the boilers. Scale is also formed both in the condensers and the boilers, and this requires frequent cleaning, as condenser scale tends to impair the vacuum.

8 The Arizona Power Company uses Verde River water for condensing purposes, this being taken through a flume at such a point that the pumping head is reduced by gravity flow. The condensate is returned to the boilers, and the raw make-up water is purified in a Cochrane hot-process purifier.

ECONOMY OF INSPIRATION CONSOLIDATED COPPER COMPANY'S PLANT

9 Upon the completion of the Inspiration plant it was placed in regular service. The performance of the individual pieces of apparatus was investigated in order to make certain that everything was working to the best advantage, particular attention being

paid to the efficiency of boilers, the adjustment of burners, furnaces, and the automatic firing system. A number of uniform load tests were made of boilers for checking purposes, and the results obtained are given in Table 1 and the curves of Figs. 1 and 2. These tests show a fairly high efficiency at rating, and, rather contrary to the usual results obtained with non-casing-set boilers, a higher efficiency at fractional loads than at rating. The higher efficiency is due to the tightness and the insulating efficiency of the steel casing. As a result of these tests, instructions were given the operators to divide the load equally among all boilers, and this, of course, was done automatically by the firing system. The operators, however, were instructed to keep as many boilers on the line at light loads as could be properly fired, maintaining a fire in each of the three

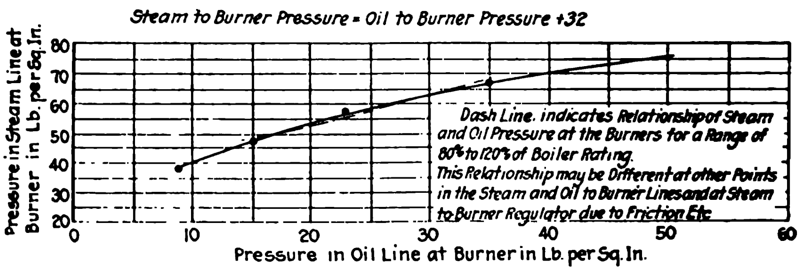


FIG. 1 STEAM AND OIL PRESSURES AT BURNER LINES

burners per boiler; below this load boilers were cut off the line, and only refired when the load again increased.

10 The curves of Figs. 1 and 2 show the relation between the steam pressure of the atomizing steam and the oil pressure, both measured in the supply pipes at the individual burner, between the throttle valve and burner. From data obtained from these curves the steam-to-burner regulator was set to give the proper pressure of atomizing steam, based on the momentary oil pressure. A complete description of the automatic oil-firing system in use in these plants will be found in a paper by the writer entitled Unnecessary Losses in Burning Oil Fuel and an Automatic System for Their Elimination, page 797, Vol. 30, of TRANSACTIONS, and the system used in these plants differs from that described in the paper merely in the use of a diaphragm pump governor to maintain a constant predetermined maximum pressure at the oil pumps; the oil-to-burner regulator then operates a throttle valve to give the desired pressure at the oil burners.

11 Under variable load the damper controller has been able to maintain CO_2 readings varying from 12 to 14.5 per cent CO_2, for which the corresponding excess air for normal conditions is 28 per cent and 6 per cent, respectively.

12 After instructing the operators, and while the plant was still under the control of the engineers, an average economy was obtained for the month of September 1915, as follows:

Average number of turbine units in operation	1
Average daily load, 24-hour basis, kw	5980
Average steam pressure at boilers, lb. per sq. in	178
Average vacuum in condensers, in. of mercury, absolute	2.66
Average rating on boilers, per cent	95
Gross boiler efficiency, per cent	80.9
Average economy, kw-hr. per bbl. of oil as fired	289
Average economy B.t.u. per kw-hr	21,500

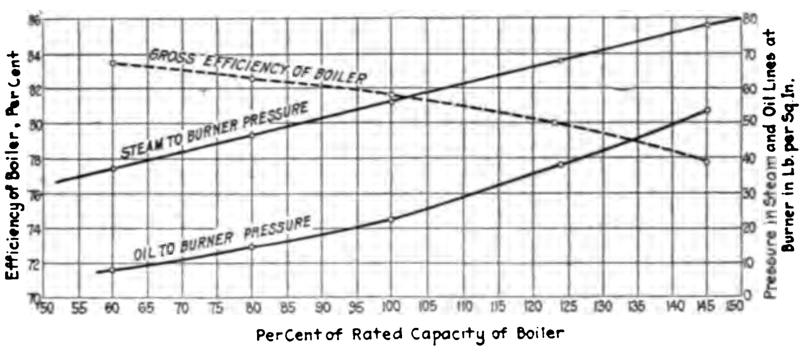


FIG. 2 PRESSURES AND EFFICIENCIES AT VARIOUS LOADS

In the subsequent operation of this plant by the owners the economy has been maintained practically equal to that shown under the direction of the engineers, but during the winter, due to colder circulating water, the economy is even better than indicated above. On the other hand, during the summer months, with the warmer circulating water and falling off in vacuum, the economy naturally drops to a lower figure than that given for the month of September.

13 This plant operates in conjunction with the hydroelectric plant at the Roosevelt Dam, and at periods of the year preference is given to hydroelectric power. As a result, there is a fractional load, or partial shutdown of the steam plant, and for certain months this in turn has naturally resulted in a reduced economy.

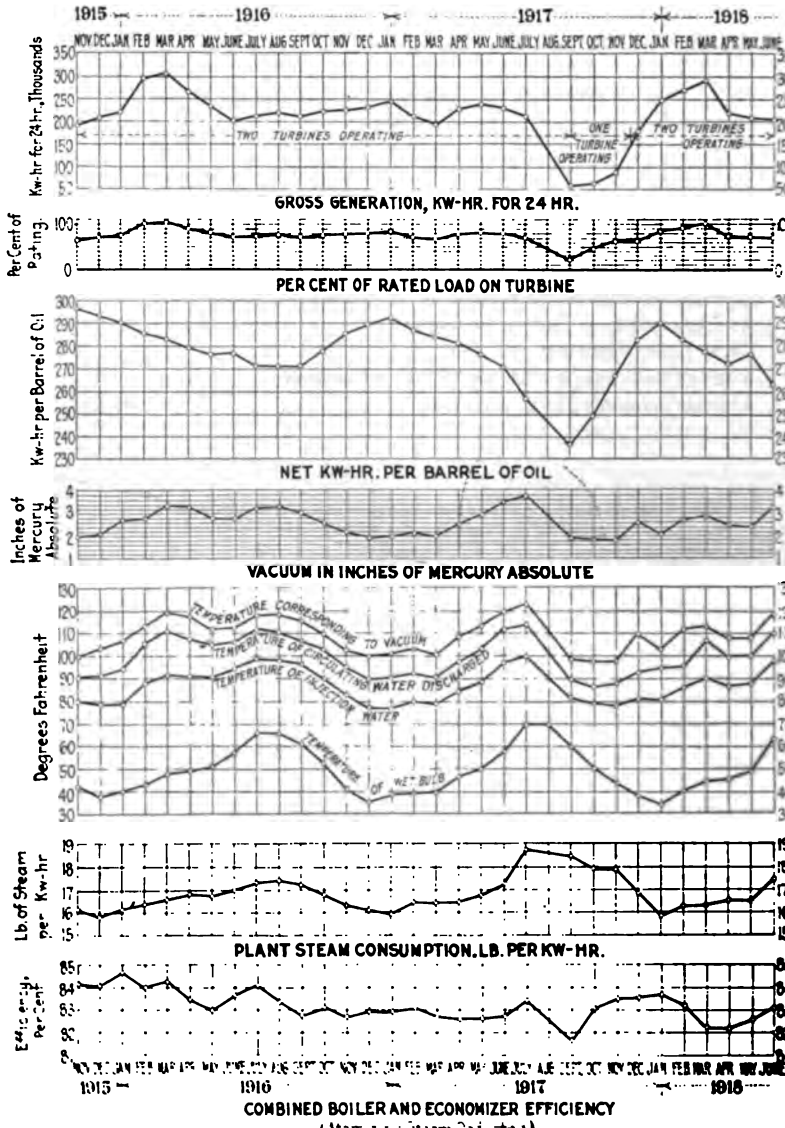


FIG. 3 OPERATING CHARACTERISTICS OF THE INSPIRATION POWER PLANT

14 The operating results for the plant, furnished by Mr. W. W. Jourdin, Mem.Am.Soc.M.E., Chief Engineer of the Inspiration power plant, are given by the curves of Fig. 3. It will be noted that the best monthly economy for winter conditions has been 294.5 kw-hr. per bbl. of oil, or 20,910 B.t.u. per kw-hr.; and the poorest economy for summer conditions for the normal load has been 257.5 kw-hr. per bbl. of oil, or 23,700 B.t.u. per kw-hr., although for the month of September 1917, due to the very light load, the economy was only 237 kw-hr. per bbl. or 25,970 B.t.u. per kw-hr. As previously stated, this plant, in comparison with non-cooling-pond plants, is subject to an accumulation of scale in the condensers, and as a result there is a slight loss in vacuum. Since purified feedwater is used, the plant is not subject to troubles from scale formation in the oil-fired boilers, nor does any loss of fuel result from shutting down boilers for cleaning or boiler blow-off.

15 The curves of Fig. 3 also give the operating records for combined boiler and economizer efficiency, atomizing steam deducted, this being the form in which the power-plant records are kept. A comparison of this result with the boiler efficiency tests would seem inconsistent without the explanation that the economizers at this plant heat the feedwater through a temperature range of from 40 deg. to 45 deg., whereas it will be noted that results for the Arizona Power Company's economizers, given in Table 2, indicate a temperature rise in the economizer of 91 deg. fahr. In proportioning the Inspiration economizers with reference to the investment and fuel saving it was assumed that the average period of operation would be less than half the year, owing to the use of hydroelectric power; it was also assumed that the fixed charges would be very high. This combination of circumstances, together with high freight rates and construction costs in Arizona, resulted in the selection of an economizer of comparatively small surface, giving a favorable return on the investment but a result considerably less favorable than that attained in the average economizer with oil fuel, when measured only by temperature rise.

ECONOMY OF THE NEW CORNELIA COPPER COMPANY'S PLANT

16 Following the installation of the New Cornelia plant in November 1917, an attempt was made by the designing engineers to check the economy of the station, but owing to the war and a scarcity of labor this work had to be abandoned before completion. While it has never been possible to show the best performance of

TABLE 1 RESULTS OF EFFICIENCY TESTS ON STIRLING WATER-TUBE BOILER IN INSPIRATION CONSOLIDATED COPPER COMPANY'S PLANT

Class M No. 26 battery Stirling water-tube boiler with steel casing. Heating surface per boiler, 7129 sq. ft. Rated boiler hp., 712.9 (based on 10 sq. ft. per hp.). Boiler-room floor elevation, 3615 ft. Normal barometer, 26.13 in.

		Test at 100 per cent rating [1]	Test at 80 per cent rating [2]	Test at 60 per cent rating [2]	Test at 125 per cent rating
Date of test, 1915		May 29	June 1	June 2	June 4
Number of test boiler in plant		5	5	5	5
Duration of test, hours		10	6	6	10
Temperature of feedwater entering boiler, deg. fahr.		210.15	200.0	191.0	209.5
Temperature of superheated steam, deg. fahr.		503.9	502.4	491.0	517.3
Deg. fahr. superheat		122.1	122.0	111.0	135.5
Temperature of steam to burner, deg. fahr.		504.2	502.7	..	..
Temperature of oil to burner, deg. fahr.		187.8	186.9	186.0	185.0
Temperature of flue gases, deg. fahr., measured across breeching outlet and in rotation	No. 1	466.5	423.7	389.0	495.0
	No. 2	469.0	427.5	394.0	502.0
	No. 3	470.1	434.7	389.0	507.5
	No. 4	473.3	434.7	394.0	508.0
	No. 5	472.5	432.5	391.0	507.5
	No. 6	470.8	429.0	390.0	510.5
	No. 7	471.3	427.0	392.0	506.0
Temperature of outside air, deg. fahr.		95.0	90.2	86.5	69.0
Temperature of air entering ashpit, deg. fahr.		93.1	100.3	95.0	75.9
Steam pressure, lb., gage		187.4	184.03	182.3	187.3
Pressure in oil line before burners, in lb., gage		22.0	15.0	9.+	36.8
Pressure in steam line to burner, lb., gage	Before burner valves	56.6	46.8	38.0	67.0
	After burner valves	..	..	..	..
Draft, inches of water	Top of first pass No. 1	0.014	+0.016	0.013	0.005
	Bottom of 2nd pass No. 2	0.062	0.080	0.032	+0.076
	Front of damper No. 3	0.076	0.070	0.020	+0.091
Draft, inches of water (power-plant instrument), front of Damper		0.065	0.074	0.033	0.110
Analysis of fuel oil by Smith Emery & Co.	Water, per cent (by centrifuge)	0.480	0.490	0.490	0.49
	Sand	Trace	Trace	Trace	Trace
	Sulphur in per cent of dry oil	1.06	1.06	1.06	1.06
	Carbon in per cent of dry oil	85.58	85.58	85.58	85.58
	Hydrogen in per cent of dry oil	12.87	12.87	12.87	12.87
	Net B.t.u. per lb. of oil as fired	18,540	18,540	18,540	18,540
	Sulphur corrected in B.t.u. per lb. oil	85	85	85	85
Total water actually evaporated, lb.		232,736	104,394	76,857	269,963
Lb. water actually evaporated per hour		23,273.6	17,399	12,809	26,996.3
Factor of evaporation		1.124	1.134	1.137	1.131
Total water evaporated from and at 212 deg. fahr., lb.		261,594	118,383	87,386	305,328
Lb. water evaporated from and at 212 deg. fahr. per hr.		26,159	19,730.5	14,564	30,532.8
Total oil fired, lb.		16,743	7,498.5	5,458.5	19,907
Oil fired per hour, lb.		1,674.3	1,249.7	909.5	1,990.7
Boiler horsepower developed		758	572	422	885
Per cent of rated capacity, based on work done by water-heating surface		106.7	80.3	59.2	124
Lb. water actually evaporated per lb. of oil		13.90	13.92	14.08	13.56
Lbs. water evaporated per lb. of oil from and at 212 deg. fahr.		15.62	15.73	16.00	15.34

TABLE 1 RESULTS OF EFFICIENCY TESTS ON STIRLING WATER-TUBE BOILER, INSPIRATION CONSOLIDATED COPPER CO.'S PLANT (*Cont.*)

		Test at 100 per cent rating[1]	Test at 80 per cent rating[2]	Test at 60 per cent rating[2]	Test at 125 per cent rating
Efficiency of boiler based on gross evaporation per cent		81.76	82.64	83.74	80.29
Per cent CO_2 in flue gas at top of first pass		15.5	15.17	15.1	15.1
Percentage analysis of gases at bottom of second pass	CO_2	15.6	15.41	15.0	14.4
	O	0.8	1.11	1.4	1.94
	CO	Trace	0	0	0
	N	83.6	83.48	83.6	83.66
Percentage analysis of gases at front of damper	CO_2	14.5	..	14.5	14.3
	O	2.3	..	2.1	2.11
	CO	Trace	..	0	0
	N	83.2	..	83.4	83.59
Per cent excess air over chemical requirements at front of damper		4.19	..	4.19	6.14

[1] Apparent discrepancy in the draft readings of power-plant instrument and thermometers are due to the different location of the nozzles and to slight leaks in flue-gas piping.

[2] On account of the waste-heat boilers and intermittent firing of the other boilers the load was very jerky. On this account the damper was fixed to a low point on the draft and the ashpit door was regulated during the test.

which the plant is capable, the operating crew have, for the most part, been very efficient in handling it, except during an illness of the chief engineer, when the economy of the plant fell off to a disappointing figure. This occurred during the summer and fall of 1918, and the figures for economy for that period are thus hardly fair to the plant. It should also be borne in mind, in connection with performance data, that the feedwater condition at this plant is such that the frequent shutdowns for boiler cleaning and the large amount of hot water which is blown off from the boilers affect to an appreciable extent its economy.

17 This plant, due to its location in the southwestern portion of Arizona, is subject to more intense summer heat than probably any other power plant in the western territory, and this in turn gives rise to considerably less favorable cooling-pond and condenser performance, with respect to vacuum, for the summer months than for the winter months. The vacuum performance is also influenced by the accumulation of scale within the condensers between the periods of cleaning condensers, due to concentration of salts in the cooling pond.

18 The best economy for this station, known to the writer, is for the month of January 1918, the average performance for four successive days being as follows:

Date 1918	Average load, kw.	Economy, kw-hr. per bbl. of oil	B.t.u. per kw-hr.
Jan. 26	7921	321.7	18,975
Jan. 27	7830	324.1	18,833
Jan. 28	7725	324.2	18,829
Jan. 29	7800	326.2	18,711

19 The poorest economy for this station during the summer of 1917 was for the month of July, namely 293.5 kw-hr. per bbl. the average vacuum was 1.66 in. absolute and the average load 550 kw. The poorest economy during the summer of 1918 was also for the month of July, or 287.5 kw-hr. per bbl. The average vacuum was 2.14 in. absolute, the average load 5850 kw., and consequently the economy for the summer of 1918 was abnormally low.

20 The monthly report for December 1918, gives the following: Average load, 7800 kw.; average economy, 312 kw. per bbl. and 19,913 B.t.u. per kw-hr. The monthly report for January 1919, gives: average load, 7790 kw.; average economy, 317.9 kw. per bbl., and 19,535 B.t.u. per kw-hr. The improvement for the month of January over December is due to a straightening out of the aforementioned difficulties experienced in the summer of 1918, and it is the writer's belief that the station will soon be operating at its best previous economy for the corresponding season.

21 It will be noted that the economy of the New Cornelia plant is materially better than that of the Inspiration. This is due somewhat to the larger turbine units installed, but in the main to the higher steam pressure and to the improved design of cooling pond, which results in better vacuum. In comparing the economy of these cooling-pond stations with that obtained in tidewater plants, allowance should be made for the size of turbine units, the obtainable vacua under operating conditions, the increased head on circulating pump due to the greater quantity of water which must be handled through the condensers, and the increased pumping head due to the cooling-pond nozzles and longer lengths of circulating-water line.

ECONOMY OF THE ARIZONA POWER COMPANY'S PLANT

22 This plant was completed in September 1917, and due to the war conditions in the mining region it was difficult to assemble a skilled operating crew. The plant was furnished under a contract

covering a complete plant-economy guarantee at 5000-kw. load, and the final test, covering 48 hours' operation in regular commercial service, under variable load, was concerned mostly with the economy at this load, although a run was made at 6000-kw. load, which is the rated capacity of the turbine The results for the final test are

TABLE 2 RESULTS OF TEST CF THE ARIZONA POWER COMPANY'S STEAM PLANT AT TAPCO, ARIZ.

Duration of test, hours	6
Boiler pressure, lb. per sq. in., gage	250
Steam pressure at turbine throttle, lb. per sq. in. gage	238.5
Avg. temp. of superheated steam at boilers, deg. fahr	546
Temp. of superheated steam at turbine, deg. fahr	516
Temp. of feedwater entering boilers, deg. fahr	207
Temp. of water leaving feedwater heater, deg. fahr	116
Temp. of circulating water from condenser, deg. fahr	62.4
Temp. of circulating water to condenser, deg. fahr	45.7
Room temperature, deg. fahr	83.2
Barometer, inches of mercury	26.773
Vacuum in condenser, inches of mercury	25.783
Absolute vacuum, inches of mercury	0.990
Temp. of flue gases leaving economizer No. 3, deg. fahr	276
Average load, kw. per hr	5,815
Power factor by power-plant indicator	0.93
Electrical Output by Integrating Meters:	
Gross kw. generated	34,890
Gross kw-hr.	5,815
Auxiliary power, k.w	277
Net kw. output	34,613
Net kw-hr. output	5,769
Oil Measurements:	
Total oil weighed, lb	34,516
Correction due to diff. in temp. at start and finish, lb	46
Oil actually used, lb	34,470
Average gravity of oil (analyzed by Smith Emery & Co.), deg. B.	17.65
Weight of oil per bbl. of 42 gal., lb	332
Heat value of oil (analyzed by Smith Emery & Co.), B.t.u. per lb.	18,703
Economy:	
Fuel used per gross kw-hr., lb	0.988
Fuel used per net kw-hr., lb	0.996
Kw-hr. per bbl. of oil, gross	336.0
Kw-hr. per bbl. of oil net	333.3
B.t.u. per kw-hr. gross	18,478
B.t.u. per kw-hr. net	18,628

given in Table 2. At all times during this test the plant was subject to a variable load, due to the regulation of the hydroelectric system. The oil was carefully weighed and the electrical output was measured by calibrated meters. The electrical output given is the net useful output for the station at the 2300-volt bus, deduction having been made for the power consumption of electric auxiliaries, including lighting for the operators' cottages, circulating water pump, deep-well pump and air washer. The average electrical auxiliary load was 46 kw., which is somewhat smaller than would have been the case had the entire head on the circulating water been overcome by pumping; against this condition is the fact that during the test the quantity of circulating water was somewhat less than specified, so that, roughly speaking, the one condition nearly offsets the other.

23 It is not possible to give daily operating results for this plant, at the load for which it was designed, for since its installation it has been maintained only for reserve purposes, carrying occasional peaks but the majority of the time a very light load, and for a number of hours during an average day, with the turbine at standstill. Of course, favorable economy is not possible under such conditions, as the fuel losses due to keeping hot boilers, piping, etc., the dead-load losses for the operation of auxiliaries, and the zero-load steam consumption of turbine result in a zero-load fuel consumption of the plant which is an appreciable percentage of the full-load fuel consumption. With the foregoing explanation, the results given below for the month of February 1918, are as follows:

Total kw-hr	1,235,580
Total bbl. of fuel oil used	4783.9
Total hours of operation	444
Average kw. for operating period	2800
Kw-hr. per bbl. of oil (delivered to lines, net)	258.50
Operating period load factor	0.467
Average kw. for monthly period	1838
Monthly period load factor	0.306

24 It is of interest to note that the turbine at the Arizona Power Company's plant is practically a duplicate of those at the Inspiration Copper Company's plant, having the same number of stages, the difference in the operating economy of the turbines being largely due to steam pressure. Here, again, is a marked increase in plant economy due to an increase in steam pressure, and also by reason of the colder river water at the Arizona Power Company's plant as compared with the cooling-pond water at the Inspiration plant and the consequent improvement in vacuum. All economy figures given for these plants are based on oil as fired, without deduction or correction for moisture, sulphur or silt.

25 The engineers have endeavored to instill in the minds of the operators of these plants and to show by example that high

economies need not merely be looked for during the test period, but can be maintained during the operating period. The Inspiration plant is large enough to permit the employment of a boiler-room engineer, but, due to their smaller size, such an engineer is not maintained at the other two plants. This plant was designed on the assumption that the average load would be maintained for a period of six months only during the year, and that oil delivered would cost about $1.45 per bbl. The New Cornelia plant was designed on the assumption that oil would cost $1.25 per bbl., but since the date of design of these plants the cost of oil has materially increased.

PERSONNEL

26 The selection of the principal equipment for the Inspiration plant and its general layout were made jointly by John Langton, Consulting Engineer for the Inspiration Consolidated Copper Company, and Chas. C. Moore and Co. Engineers, which firm was also responsible for the detailed designs, installation and tuning up of the plant.

27 For the New Cornelia plant the entire work was in the hands of Chas. C. Moore and Co. Engineers, with the approval of A. G. McGregor, Consulting Engineer for the New Cornelia Copper Company.

28 The Arizona Power Company's plant was designed and built by Chas. C. Moore and Co. Engineers, with the approval of R. S. Masson, Chief Engineer for the Arizona Power Company.

29 A considerable portion of the testing work on the Inspiration plant was handled by A. G. Budge, under the writer's direction, and for the New Cornelia and Arizona Power Company plants by T. B. Paulson, also under the writer's direction.

DISCUSSION

C. H. Delany (written). The Pacific Gas and Electric Company is operating at Sacramento, Cal., an oil-burning plant which is similar in some respects to the three plants discussed in Mr. Weymouth's paper. This plant was built in 1911 and is equipped with one 5000-kw. Curtiss turbine, Stirling boilers, Peabody Hammel oil furnaces, and a surface condenser taking water from the Sacramento River. This plant was designed for 175 lb. pressure and 100 deg. superheat at the turbine throttle, and is therefore similar in this respect to the Inspiration Copper Co.'s plant. The plant, however, being essentially a stand-by plant, is not equipped with economizers, steel casings for the boilers, or automatic fuel-oil regulators.

In comparing the operation of different steam plants operating at variable loads, it is convenient to plot on a diagram the kilowatt-hours generated against the total oil burned, each point plotted representing the average results of a month or a day as the case may be. If a sufficient number of points at different loads is obtained, it is possible to draw a line through them representing their average, and this line in most cases is found to be practically a straight line. Such a diagram for the Sacramento station for the year 1918

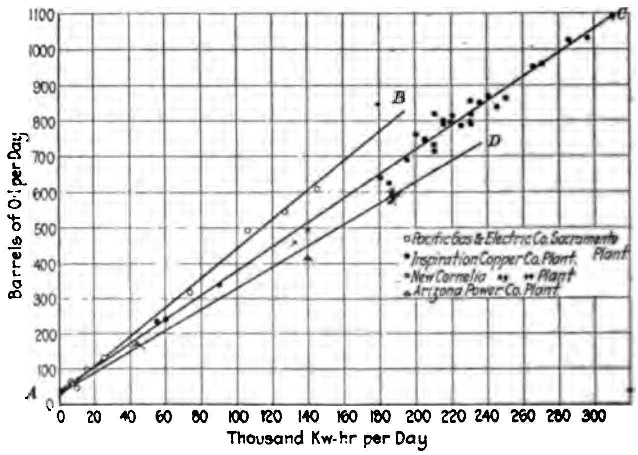


FIG. 4. DIAGRAM SHOWING RELATION BETWEEN BARRELS OF OIL BURNED PER DAY AND KILOWATT-HOURS GENERATED

is given in Fig. 4 and to it have been added points taken from Mr. Weymouth's paper for the three Arizona plants. The points for the Inspiration Copper Co.'s plant are sufficiently numerous to enable the line AC to be drawn through them, corresponding to the line AB for the Sacramento plant. It will be noted that these two lines intersect the ordinate of zero load at almost the same point, A, indicating about the same quantity of oil (that is, in the neighborhood of 25 or 30 bbl. per day) required to operate either plant at no load.

There are not sufficient points to enable a similar line to be drawn for either the New Cornelia Copper Co. or the Arizona Power Co. plants. However, assuming the point A for zero load to be the same as for the Inspiration Copper Co., the line *AD* has been drawn as an approximate average for these two plants.

This diagram brings out more clearly than a mere statement of the kilowatt-hours per barrel of oil, though perhaps somewhat crudely, the essential differences between the various plants. Thus a comparison of the lines *AD* and *AC* shows at a glance the effect of the higher steam pressure, higher superheat and better vacuum carried at the New Cornelia Copper and Arizona Power Companies' plants. The line *AC* compared with the line *AB* shows the economy resulting from the use of economizers, steel casings, fuel-oil regulators, and other minor differences such as the greater age and smaller size of the turbine in the Sacramento plant.

W. W. Jourdin (written). Referring to Par. 4 of the paper, the Inspiration power plant is equipped with engine-driven circulating units and barometric hotwell, with sealed sump and lift pumps located on a hillside at a sufficient distance below the condensers to secure positive drainage of the condensate therefrom.

The high-pressure cylinders of the blowing engines are being replaced with poppet-valved cylinders good for 250 lb. pressure, and all new equipment, including boilers recently installed, is designed for the higher pressure. However, until such time as the original boilers may be disposed of, a maximum of only 200 lb. pressure will be available at the throttles.

The cooling tower was not put into service until November 1918, therefore the plant operated under the handicap of poor vacuum, due to insufficient cooling capacity, throughout the period covered by the curves in Fig. 3.

Figs. 1 and 2 do not represent present practice — in fact, there has been a very considerable departure in numerous operating features from the original instructions, although these in principle were good. Owing to labor troubles, the plant was shut down from July 2 to September 20, 1917, inclusive, and in the absence of detailed explanation concerning conditions, the curves would better be broken between June and October 1917.

The heat recovery is low in the economizers not so much on account of inadequate heating surface as because the recovery by the boilers is so high, compared with coal-burning practice, that little heat available for warming feedwater is contained in the gases to the economizers.

The Inspiration plant is jointly owned by two companies, and the money involved is so large, due to high fuel costs, that it is essential that the measurement of all quantities shall be so made as to secure the highest possible accuracy. The metering equipment is quite elaborate, and the accounting is on the basis of continuous overall plant test; consequently the operators are enabled to check economic performance more closely than is usually possible even in the most highly developed plants.

The economy reported is based on net electrical production, and no deduction is made for fuel consumed in warming up and stand-by due to changes in electrical load.

W. D. Ennis (written). The maintenance of high overall efficiency in regular operation is a feature to be expected with well-designed plants using oil fuel. In discussing the reasons for the superiority of the New Cornelia plant over the Inspiration the author omits mention of the higher superheat used. Allowing for differences in pressure and superheat, the steam temperatures in the two cases are 471 deg. and 551 deg. The load factor may have been of weight also, but there seems to be nothing in the table dealing with this plant to show whether or not both turbines were running. If they were, the New Cornelia load factor was around 0.50 as compared with a value of about 0.70 for the Inspiration plant.

A most interesting feature of this paper is the indication of maximum boiler efficiency at about 0.60 rating (Fig. 2). The author attributes this to the tightness and insulating efficiency of the sheet-metal casing. It is regrettable that the boiler proportions are not given in some detail. The curve of boiler efficiency against rating cannot be considered a normal curve. The (actual) rate of evaporation at normal rating works out 3.26 lb., or at 60 per cent rating, less than 2 lb.

W. N. Best (written). There is one particular point to be emphasized in the use of oil fuel in power plants, and that is the importance, wherever possible and wherever plants are of sufficient size, of the company's employing an engineer whose duties will be to see that the oil is properly burned in order that the strictest economy and highest efficiency may be attained and maintained at all times. The writer has always recommended this in power plants and in forge shops, and he is glad to learn that there is one plant in the United States that really has placed a man in charge, whose duty it is to see that the oil is properly burned. He can save his wages many times by the economy effected in fuel by his attention to his specific duties.

W. J. Davis, Jr. (written). We have to thank Mr. Weymouth for presenting us with some very interesting and useful information on the possibilities of obtaining unusually high economies from comparatively small steam power plants operating in districts where lack of sufficient circulating water for the condensers makes necessary the use of cooling ponds.

The monthly average performance of the Inspiration Consolidated Copper Company's plant of 21,500 B.t.u. per kw-hr. is especially commendable when we consider the plant conditions, namely, units of 6000 kw. capacity, steam pressure of 178 lb. gage at boilers and vacuum of 2.66 in. absolute back pressure. If correction is made for the low vacuum at the Inspiration plant assuming 1 in. average back pressure in the condensers, a condition frequently obtained in modern Eastern plants, the average economy would be reduced to less than 20,000 B.t.u. per kw-hr.

This figure is very good and compares favorably with the economy of plants of modern design using much larger units and operating at higher steam pressures and superheats.

While the performance of the Inspiration plant may be held to be unusually good, that of the New Cornelia is considerably better due mainly to the remarkably high vacuum conditions resulting from the improvements in the design of the cooling pond and also to the use of higher steam pressures and superheats.

These plants were both designed to meet unusual conditions of high cost of fuel and an inadequate supply of cooling water. The high economy maintained as shown by the monthly plant records should prove highly gratifying to the designers of the plants as well as to those in charge of operating them.

A questionnaire on power conservation was undertaken by the Chairman of the Engineering Committee of the Pacific Coast Section of the National Electric Light Association, the results of which appeared in the *Journal of Electricity* for April 15, 1918, and which showed the economies obtained in steam plants where oil was the fuel. The subject of power plant losses was also undertaken by R. J. C. Wood, General Superintendent of the Long Beach Plant of the Southern California Edison Company and the results of his investigation were published in the *Journal of Electricity* for April 1, 1918.

Robert Sibley (written). In the Spring of 1918 the questions of the conservation of fuel oil and of its utilization at the highest degree of efficiency became matters of prime importance throughout the nation and especially west of the Rocky Mountains where there was a shortage of hydroelectric power and an urgent necessity for the economical operation of steam plants.

In both of these articles the average economy of the great steam-electric power plants of the West was shown to be from 200 to 240 kw-hr. per bbl. of oil. There was widespread interest among engineers, therefore, when it became known that Mr. Weymouth had secured such results as are mentioned in his paper. An average economy of 289 kw-hr. per bbl. of oil as fired was attained for the month of September 1915, at the Inspiration plant while it was still under the control of the engineers. At the New Cornelia plant the monthly report for September 1918, showed 312 kw-hr. per bbl. of oil, and in January 1919, the average economy rose to 317.9 kw-hr. per bbl. of oil. At the plant of the Arizona Power Company, during September 1917, the final test covering 48 hours' operation in regular commercial service under variable load due to the regulation of the hydroelectric system showed the remarkable record of 333.3 kw-hr. per bbl. of oil.

Such instances as these indicate the remarkable attainments that are possible in modern high-pressure steam generation and the economic performance of automatically-controlled oil-burning furnaces.

The only adverse comment which can be made of the excellent results obtained by Mr. Weymouth is that the tests were conducted under abnormal conditions wherein engineers, expert in this particular field, were employed so that possibly under ordinary operating conditions these efficiencies could not be obtained. The fact remains, however, that a high goal of accomplishment has been established which shows clearly that the engineer, properly trained, can increase efficiencies far beyond those now prevailing in our power plants. It would also indicate that our power plant managers should give more attention to this important subject and see that men who have in charge the operation of the plant are qualified to develop the highest efficiency attainable.

Mr. Weymouth's paper brings to our attention again the advisability of the establishment by the Society through its Power Test Codes Committee of standard test codes for oil-fired boilers. While it is true that the present boiler test code may be adapted to this purpose, many definite rulings should be made by the Committee. The subject of definition of efficiency, for instance, is one of misunderstanding where oil is used as a fuel, due to the fact that a certain amount of the steam generated is used for atomizing purposes.

W. L. Du Moulin (written). Results of tests on power plant equipment are always of interest. They show the performance of the equipment under the most favorable conditions. Information of this nature is particularly of interest from an engineering standpoint, and desirable to have. However, test conditions are usually special, and consequently, actual average operating results over a period of time are of more practical benefit. The data given of the economy of the power plant of the New Cornelia Copper Company are an indication of what may be attained under operating conditions by a plant subject to the extreme summer heat of Southwestern Arizona, and located where circulating water for condensing purposes is a serious problem. The operating records of this plant demonstrate that economies can be obtained by a well-designed plant operating under favorable conditions approaching those obtained under test conditions.

The operating conditions of the New Cornelia plant are exceptionally favorable, both because of the high power factor and the relatively high and steady load factor. About 72 per cent of our total power generated is converted from alternating to direct current by four motor-generator sets with synchronous motors. These motor-generator sets supply the direct current required by the electrolytic tankhouse, and their load is, therefore, uniform.

During the last eleven days of January 1919, the average net economy in kw-hr. per bbl. was 323.1; average load 8085 kw.; average vacuum 1.28 in. absolute. January is the most favorable month in the year, due to the better vacuum obtainable on account of the cooler condenser circulating water, and July is about the most unfavorable month, because of the poorer vacuum. The average net economy in kw-hr. per bbl. for July 1919, was 302.4; average load 7587 kw.; average vacuum 2.08 in. absolute. On January 26 and 27, 1919, the net economy in kw-hr. per bbl. was 327.3 and 326.5; average load 8091 and 8012 kw. respectively. While the records of such individual isolated days are interesting, yet the average records covering longer periods of time are the ones that will be of more practical value, as the influence of the personal equation is reduced to the minimum. In this connection, therefore, the average net economy for 1918 of [illegible] kw-hr. per bbl. of oil, average load 7250 kw., will be of interest. With careful operation this yearly average economy may be improved.

The superheaters installed were specified to give 150 degrees superheat. The average superheat has been 1[illegible]0 to 110 deg. fahr.

The boiler plant consists of five 522-hp. Class M, No. 30, steel-incased Stirling water-tube boilers equipped with attached type of superheater. Steam is taken from the rear drum. One boiler was equipped with a mechanical soot blower. Some of the blowing elements of this equipment interfered seriously with the oil firing when the soot blower was operated. These elements were removed and installed in the remaining boilers so as to remove the soot from the superheaters, which resulted in increasing the superheat about 15 degrees. It is the writer's opinion that mechanical soot blowers properly installed in all boilers would materially aid in maintaining the best economy. In boilers of the size contained in this plant, the removing of soot effectively from boiler and superheater flues cannot be accomplished with a hand lance. The writer desires also to emphasize the value of operating the economizer tube scrapers in connection with oil-fired boilers in order to maintain good economy. Also, the installation of feedwater regulators will tend to a reduction in operating difficulties.

The automatic firing regulator has been an appreciable factor in maintaining our economies. A steam pressure has been maintained with a variation of not over 3 lb., and the amount of steam required for atomizing the fuel oil very closely regulated.

While the exceptionally favorable operating conditions have a very considerable bearing on the good economy being obtained, yet the boiler room plays a most important part, especially in as isolated a place as Ajo, and in a country where fuel is costly, as large losses may occur there. Too much care cannot be exercised to determine the proper routine to establish in order to obtain the best results under the operating conditions. The boiler room requires constant attention and close application to detail to maintain good economy.

Following is the average boiler room data covering a representative period:

Temperature of feedwater entering economizers, deg. fahr.	175
Ave. temperature of feedwater leaving economizers, deg. fahr.	234
Temperature of superheated steam, deg. fahr.	508
Degrees of superheat, fahr.	100
Temperature of oil to burners, deg. fahr.	190
Steam pressure, lb. gage	255
Pressure in oil line to burners, lb. gage	10 to 15
Pressure in steam line to burners, lb. gage	30 to 45
Lb. of water actually evaporated per lb. of oil	13.7
Factor of evaporation	1.097
Water evaporated from and at 212 deg. fahr. per lb. oil	15.02
Boiler horsepower developed	3500
Per cent of rated capacity developed	106½
Boiler efficiency, Gross, per cent	78.4
Net (deducting steam to burners) per cent	77.6
Boiler and economizer efficiency, per cent	82.8

By consistent work, the boiler room performances as indicated may be improved.

All economies given in Mr. Weymouth's paper and in this discussion are net.

D. S. Jacobus (written). Information respecting plant efficiencies such as that given in the author's paper and by those who have discussed it is of great value in placing on record the progress in the art of power generation. Saving of fuel was brought prominently before the country during the war and is still prominently before the public in view of the increased cost of fuel and the general movement that has been organized to avoid excess waste. Higher and higher boiler-room and power-plant efficiencies are being obtained and reliable tests are most useful in a study of the problem and in inspiring those in charge of power plants to secure the best results.

Mr. Sibley states that the only adverse criticism that can be made is the fact that the author's test was, perhaps, conducted under abnormal conditions, in that engineer experts were employed and that under ordinary operating conditions the efficiencies which the author gives could not be attained. The tests given in the paper bear on the results secured in the boiler room and not in the plant as a whole. All of the plant economies given in the paper are operating figures and Mr. Sibley's statement therefore applies only in part to the figures given in the paper. The general idea involved in Mr. Sibley's statement is certainly correct for most cases, but there is no reason why there should be any difference between test and operating results with the very best operation. In some plants every detail is attended to in the same way as in a continuous test — in fact the plant is supervised by high-class experts and subjected to a continuous test; and there are instances where the operating results are as high as any test results. More and more attention is being given to securing engineers of the right type for supervising the operation of the boiler and engine room with the result that higher and higher efficiencies are being obtained. It is a mistake to assume that as high a type of man is not required in the boiler room as in the engine room, for as I have said on many occasions, there is more to be lost or gained through the operation in the boiler room than in any other part of the plant.

The results secured at the Inspiration plant, taking into account the conditions outlined in Mr. Jourdin's discussion, show what can be done with high-class attendance and operation. The same applies to the figures for the New Cornelia Copper Company plant which are amplified in Mr. Du Moulin's discussion. To obtain high efficiencies of the sort there must be a proper equipment, but aside from this there must be the highest class of operation.

It is interesting to compare the efficiencies secured in the oil-burning plants with those that have been obtained with coal fuel. The best maintained economy for the Inspiration plant is given as 20,910 B.t.u. per kw-hr. and the best performance of the New Cornelia plant is about 18,000 B.t.u. per kw-hr. Mr. Alex Dow in his discussion of a paper presented by Mr. Richard H. Rice on Recent Installations of Large Turbo-Generators at the New York meeting of the American Iron and Steel Institute, 1917, gave the following figures for the results secured from the Conners Creek Power House of the Detroit Edison Company.

	12 months ending June 30, 1916	12 months ending Dec. 31, 1916	3 months ending March 31, 1917
Kw-hr. output	125,158,800	162,117,600	54,654,900
Maximum demand (30 minutes)	35,000	36,000	45,000
Average load	14,300	18,500	25,300
Load factor	.409	.514	.562
Coal per kw-hr., lb.	1.44	1.45	1.56
B.t.u. per kw-hr.	19,700	19,800	20,300

20,000-kw. turbines are employed at Mr. Dow's plant.

Mr. Dow states that the difference between the July to June, 12 months, and the January to December, 12 months, was due to disturbances of coal supplies and that the same cause, together with the increased use of heat in the buildings during the winter months affected the three months period given in the final column.

Mr. Dow estimated what might be done in securing efficiency through making certain changes in the plant, his idea in this connection being as follows:

> Were we designing today for fuel at $5.00 — which seems to be the probability — we would buy turbines of still higher rotative speed, which would require about 9 per cent less steam than our Conners Creek turbines, and which would be *nearly* as reliable; we would install economizers, for which we have room, but which we have not heretofore thought desirable, and thereby bring our maintained boiler-room efficiency up from 76 per cent to, say, 81 per cent; and we would make certain other refinements in our heat balance which might save 1 per cent of our total fuel. The result of these changes would be a reduction from a normal use of 19,700 heat units per kw-hr. to something like 17,000 per kw-hr. of net output.

The high efficiencies secured in Mr. Dow's plant are an instance where the operating results are as high as the test results, as the plant is fitted with all the apparatus necessary for making a continuous test, and a continuous test is actually conducted with the plant operated with the highest degree of intelligence.

Regarding the limit which will be reached in the economy of large power plants, the writer recently made the following statement:

> A large steam-turbine plant of the best modern design can be built to generate a kw-hr. with a heat consumption based on the heat in the fuel of 17,000 B.t.u. per kw-hr. This is a round figure for plants of the best modern construction throughout with a load factor of, say, 60 per cent and steam pressure of 300 lb. per sq. in. By increasing the steam pressure and raising the superheat the figures could no doubt be reduced to the neighborhood of 15,000 B.t.u. per kw-hr.

To secure these higher efficiencies we must install plants involving more complication than the older plants. The day is fast passing where simplicity is considered of first importance irrespective of the economy. Where modern power plants have been installed with the more complicated apparatus it has been found that with the right sort of instruments the same class of men that operated the more simple plants, after proper training, effectively operated the more complicated plants and experience has demonstrated that the saving in fuel far offsets the increased cost of investment and the cost of providing better expert supervision.

THE AUTHOR. The straight line diagram representing barrels of oil and kilowatt output, submitted by Mr. Delany, is very interesting, and the writer has used this extensively when investigating the economy of single-unit plants at fractional loads. It is a fact that a line of the character shown is very nearly a straight line throughout its rated capacity for a single-unit turbine plant, but when two or more units are in operation at the peak load, and with a reduction in the number of units in operation at the lighter loads, that is turbines, boilers and auxiliaries, the line becomes a jagged line, instead of a straight line, and the line which Mr. Delany has shown for the Inspiration plant is, therefore, approximate only. Further, as the Arizona Power and New Cornelia Co.'s plants have different characteristics, and different rates of economy, it is not accurate to draw one line as Mr. Delany has done, to represent the performance of the two stations.

The comparison of lines is likely to lead to serious errors. For instance, the highest load recorded for the Sacramento plant is 145,000 kw-hr. per day, corresponding to which the average load was 6041 kw. and the economy 238 kw-hr. per barrel of oil, yet Mr. Delany has extended this as a straight line corresponding to average loads of about 8000 kw. with a higher apparent economy. This overload, on a 5000-kw. unit, with boilers and connected auxiliaries, if such a load were possible, would result in an upward turn of the line beyond the economical rating of the plant.

The graphic method, when plotted with respect to the total load, gives a distorted comparison. For example, a plant of best modern design, having a 30,000-kw. unit, operating with 300 lb. pressure boilers, 150 deg. superheat, and tidewater vacuum, would give, at full load, a test economy of 369 kw. per barrel of oil, or 16,772 B.t.u. per kw-hr. If the characteristic line for this plant had the same ratio of dead-load to live-load fuel consumption as the line for the Arizona Power Co., as plotted by Mr. Delany, it would show an economy, at 5000-kw. load, of about 245 kw-hr. per barrel of oil, and when plotted on the same sheet with line "*B*" shown by Mr. Delany, would indicate about the same economy at the 5000-kw. load; and yet, comparing the economies at rated loads, there is a great superiority for the larger plant. The comparison of zero-load economy is inaccurate since the Inspiration plant, with two units in operation at zero load, would show practically double the fuel consumption for a single unit.

Although a single line gives an approximate idea of the performance, at various loads, of a single-unit tidewater plant, a family of lines is necessary to represent the performance of a plant such as that of the Inspiration Copper Co., having a cooling pond for condensing purposes, due to the effect, on the economy of the plant, of the large variation in vacuum for winter and summer conditions. While it is necessary to point out these limitations with the graphic method of comparison, we are indebted to Mr. Delany for pointing out the usefulness of this method in checking up the day-to-day economy of a given plant, at various loads.

Referring to Mr. Jourdin's discussion: The boiler efficiencies given in Figs. 1 and 2 were the result of a series of uniform load tests of boilers. Mr. Jourdin does not state whether these results have been improved in subsequent operation, and no doubt he means that in variable load operation the efficiencies are necessarily somewhat different than under constant load test conditions. It will be noted that Fig. 3 gives the combined boiler and economizer efficiency, with atomizing steam deducted, taken from the power plant logs, this data being supplied to the writer by Mr. Jourdin, and showing a very favorable performance. Referring to the performance of economizers: This is relatively poorer than in other stations operated with equal boiler efficiency, partly because of the less economizer surface per boiler horsepower; this is shown by a comparison of results at the Inspiration and Arizona Power Co.'s plants, and the writer is unable to agree with Mr. Jourdin's explanation of this detail.

Mr. Ennis states that maintenance of high boiler efficiencies is to be expected in regular operation with well designed oil-burning plants, but he apparently is not aware that many well designed plants are not showing as high overall efficiencies as those given in the writer's paper. The operating efficiencies for the New Cornelia plant were based on one unit only, and full information is given to determine the load factor. The curve of boiler efficiency against rating is a normal curve for an oil-fired steel-incased boiler. If the writer understands it correctly, the inference from Mr. Ennis' discussion is that if these oil-burning plants had been burning coal, a considerably poorer economy would have resulted. It is regrettable that no economies have been given by Mr. Ennis for record performances of modern coal-burning plants, having units of the size given in the writer's paper. Happily the conversion of a number of eastern coal-burning plants to oil will soon afford comparative data. From the information as to operating boiler efficiencies given for the Inspiration plant, and in Mr. Du Moulin's discussion for the New Cornelia plant, also in Dr. Jacobus' discussion for coal-burning plants, it will be seen that the operating efficiency with oil fuel is practically the same as in the best fired stoker plants. It may be interesting to point out the fact that in oil-burning plants there is a deduction for the amount of steam for atomizing the oil, for heating, and for pumping oil. While it is true that good firing with oil fuel permits a very small percentage of excess air over chemical requirements, the large hydrogen content of the oil decreases the actual efficiency of the boiler. Were it possible to burn liquid carbon with the same minimum excess air as with oil fuel, boiler efficiencies would be obtained of nearly 90 per cent; and the difference between this figure and the actual efficiency is mainly due to the large hydrogen content, and the formation of steam from the combustion of hydrogen, with its large latent heat, superheated to the temperature of the escaping gases. In coal fired plants, having economizers, there is much greater heat recovery in the economizers owing to the greater weight of gases per boiler horsepower, and in turn the higher temperature of escaping gases. Therefore, it is a fair statement that the net combined boiler and economizer efficiency, after deducting steam for atomizing, heating, and pumping oil, is but very little better in oil-burning plants than that obtained in the best coal-burning plants, taking as a basis for coal the figures given in Dr. Jacobus' discussion, which the writer understands were obtained from actual performances at the plant of the Detroit Edison Co.

Mr. Sibley, no doubt in error, has referred to the economies given in the writer's paper as test economies under the care of experts, instead of operating economies. A more careful reading of the writer's paper would have indicated to Mr. Sibley that the performances for the Inspiration plant, given in Fig. 3, are strictly operating records, under the care of the regular power plant operators, extending over a number of years, and not in the hands of the engineers. Also, as the writer stated, the engineers never developed the economy of the New Cornelia plant, and the figures given are strictly operating results, no complete plant tests ever having been made by the engineers. The economy for the Arizona Power Co.'s plant is admittedly a test result, but one which can be practically maintained, since it has been proven that in the operation of the Inspiration plant, subsequent to the engineers' tests, actual operating efficiencies were secured fully equal to the test results. As has been pointed out by Dr. Jacobus, modern power plant operation is only at its best when each day's run is regarded by the regular operators as a test performance. While engineers have frequently excused the rather ordinary performances of their stations by claiming that superior results were obtained under test conditions, the writer is quite sure that this is not the spirit of Mr. Sibley's discussion. The men responsible for the economy of the Inspiration and New Cornelia plants are very clever men, but no better than the type of men that should be employed in every modern steam-electric power station. Mr. Sibley's statement of economies of other plants on the Pacific Coast is hardly fair, since many of these operate in conjunction with hydro-electric plants and are not on as favorable a load factor basis as the Inspiration and New Cornelia plants. While the writer does not know of any performances of other Pacific Coast plants equal to those given in his paper, there are several plants which have shown very high efficiency, higher than indicated by Mr. Sibley. The writer is pleased to endorse Mr. Sibley's recommendation that the Power Test Code Committee give especial attention to codes for oil fired boilers.

The figures given by Dr. Jacobus afford an interesting comparison of the efficiencies of coal-burning and oil-burning plants, but allowance should be made for the fact that the plants referred to in this paper are those having much smaller units than the coal-burning plants quoted by Dr. Jacobus. Dr. Jacobus' statements as to the ultimate economies now possible for large plants are of great interest to engineers.

When considering the boiler efficiencies given by Mr. Du Moulin for the New Cornelia plant, and comparing these with the efficiencies given by Mr. Jourdin for the Inspiration plant, due allowance should be made for the fact that in the New Cornelia plant there is a considerable heat waste due to the necessary, but abnormal amount of boiler blow-off, and as well the slightly higher temperature of escaping gases, due to the higher steam pressure of boilers and higher temperature of water within boilers.

No. 1704

ELEMENTS OF A GENERAL THEORY OF AIRPLANE-WING DESIGN

By Walter C. Durfee, Boston, Mass.
Member of the Society

This paper presents in brief outline form ten subjects which have reference to the theory of fluid motion around the wings of airplanes. These are: the vortex theory of lift; the theory of initial motion around wings; vortex theory of shape; hydrodynamic-electromagnetic analogy; action of vortices with reference to each other; action on vortices with reference to their images; influence of the local wind; laws of energy content in trailing vortex; friction and head resistance; and explosion of eddies. These various subjects are not discussed but are merely brought forward for the purpose of providing a starting point for discussion.

THERE are several subjects which seem so interesting in connection with a study of the action of wings upon the air that the writer has thought it valuable to the Society to place them on record, in brief outline and in such a way as to provide a starting point for discussion and the addition of any data which members of the Society may wish to contribute. These subjects which have reference to the theory of fluid motions around the wings of airplanes are as follows:

a The vortex theory of lift, which states that the air which passes the wing of an airplane, or the blade of a propeller, contains a component of circulation around that wing or blade, in such a direction that there is a comparatively high velocity and low pressure on the upper surface of a wing; and a comparatively low velocity and high pressure on the under surface.

b A theory which states that an imperfect fluid will act like a perfect one momentarily; from which it may be inferred that the circulation around a wing cannot exist at the first moment or beginning of its motion of advance but must develop at some time after the first beginning

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

of the motion, since there is no circulation in the beginning.

c The vortex theory of shape, which treats of a solid body in motion as being somewhat similar to the core or kernel of a group of vortices.

d The hydrodynamic-electromagnetic analogy, which states that distributions of fluid motion are very similar to distributions of magnetic flux; so that one may calculate the fluid motion around a supposed vortex or group of vortices mechanically, by arranging electric currents or groups of currents in a manner analogous to the supposed vortices, and measuring the magnetic forces which result.

e The laws of vortex motion with reference to the action of vortices on each other, by which it seems possible to estimate the circulation or strength of the trailing vortex loop which is generated by a wing in flight.

f The laws of the actions of vortices with reference to their images in solid surfaces combined with the laws, so far as known, concerning the generation of eddies and vortices by friction, especially near sharp edges.

g The concept of a local direction of the wind as due to the effects of all vortices existing in the neighborhood of a wing — such as its own trailing vortices and the influence of neighboring circulations.

h Laws concerning the energy contained in various distributions of vortex motion by which one may estimate favorable arrangements of the trailing vortex systems in terms of the load carried by various parts of the wing span, and from which the drag might be estimated.

i Coefficients of friction and head resistance representing losses of energy which can be added to the losses attributed to the energy of the trailing vortices.

j Experience concerning the explosion of eddies and vortices and the causes and effects of such disturbances.

2 It is the writer's belief that there are engineers, mathematicians and experimenters in the Society who can give illuminating and interesting statements concerning the subjects mentioned; and that a group of such statements assembled in the form of a discussion would constitute almost a complete and classical theory of the action of wings in steady flight. This paper, therefore, outlines in

a preliminary way the bearing of these various theories and indicates their approximate exactitude.

a THE VORTEX THEORY OF LIFT

3 It is not difficult to believe that a component of circulation exists around a wing in flight. If it is granted that the wing carries any load at all, as wings evidently do, there is certainly a difference of air pressure between the lower and the upper surface. Consequently there are accelerations in the neighboring fluid from the under surface around in various circuits to the top surface; corresponding to the fall in pressure from one surface to the other. The quiet or still air into which a wing advances, experiencing these accelerations, must accumulate an upward velocity in front of the wing, and disturbances of a similar nature evidently must occur not only in front of the wing but also to the right hand and to the left hand. Since an upward motion in one region involves a downward motion in another, there must be a downward motion somewhere. Actually after the beginning of the flight there is a sort of circulation up in front and down behind; and consequently to the rear above and in a forward direction below.

4 In practice such motions can sometimes be seen in the form of little jerks or jumps of a fluid in the neighborhood of a model wing passing through it. It is not difficult to believe that this disturbance around a wing is rather similar in arrangement to the distribution of velocity around a vortex or group of vortices — their axes parallel to the span of the wing, and perpendicular to the direction of advance. The intensity of motion is very likely greatest near the seat of the disturbance.

5 According to mathematical theory the lifting force would be in proportion to the strength of the vortex and to its rate of advance, just as the lifting force on a wire in the armature of an electric motor is in proportion to the strength of the current and to the intensity of the magnetic flux from the pole pieces. A formula is given in the Encyclopedia Britannica for the theoretical action of a vortex which surrounds a circular rod which is projected sideways. A force develops perpendicular to the axis of the rod and vortex and at right angles to the motion of advance. This is very similar to the lift of a wing in flight.

6 Practical examples, however, suitable for mathematical analysis seem to be very rare, nevertheless the writer found one case

which seemed to be reasonably free from objectionable complications. This was a wing tested by Eiffel (Eiffel No. 8, at 9 deg. center section). From the measured pressures in this case the probable approximate velocities of the air near the wing surface were estimated, using Bernoulli's theorem. From these velocities a calculation was made of the circulation around the wing, which is the line integral of the tangential component of the velocity vector in a circuit around the wing. The result agrees with the theoretical formula for lift, or $L = \rho Vml$, in which L = force perpendicular to advance; ρ = density of air in absolute units; V = velocity of advance; m = circulation of the vortex; and l = mean span of sustaining vortex. It would be interesting to have more measurements of the circulation around wings.

FIG. 1 DIAGRAM TO ILLUSTRATE THEORY OF INITIAL MOTION AROUND WINGS

b THEORY OF INITIAL MOTION AROUND WINGS

7 It is very evident that no circulation exists around a wing when it is standing still in quiet air on the ground. Mathematical theory further declares that circulation cannot be expected to develop immediately at the beginning of the advance. In the first instant of motion conditions are supposed to be very much as they

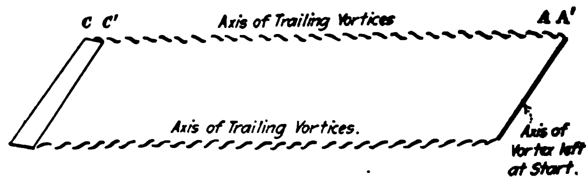


FIG. 2 DIAGRAM FURTHER ILLUSTRATING THE THEORY OF INITIAL MOTION AROUND WINGS

would be in a perfect fluid. The amount of circulation if zero at the start would be zero immediately afterward. For example, imagine, for simplicity's sake, an inclined plane moving from the position AA' to BB' in Fig. 1, starting suddenly from rest. The volume equivalent to the space AA'–BB' can be expected to be displaced and to go around the edges in such a manner that there is no net circulation around the section. Very soon after this beginning of motion, however, in the case of a real wing suddenly started in a real fluid, a very violent eddy or vortex is left behind at $A'B'$. When the wing has advanced to a further position CC' the conditions are as sketched in perspective in Fig. 2. There is a vortex loop stretching rearward from near the wing tips and joined together by the eddy generated at the starting point. This vortex circuit is completed by the sustaining vortex which circulates around the instantaneous position of the wing.

c VORTEX THEORY OF SHAPE

8 Suppose that the axes of a number of equal-sized vortices are arranged as the circles in Fig. 3. Then the direction of motion in the fluid due to their combined action is almost exactly[1] in the directions indicated by the full-line arrows. Suppose that a certain motion of translations is added to this particular arrangement of vortex motion. Then the resultant velocity of total result may

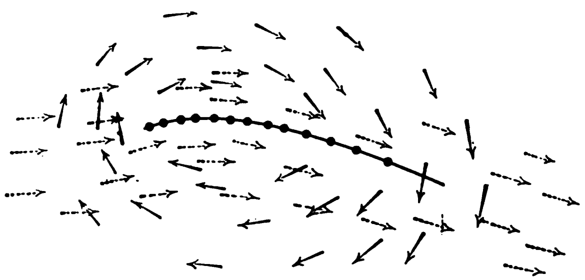


FIG. 3 DIAGRAM TO ILLUSTRATE HYDRODYNAMIC–ELECTROMAGNETIC THEORY

be in the directions indicated by the dotted arrows. This particular kind of fluid motion corresponds to a certain velocity of motion added to a certain arrangement of vortices.

9 Now it is a fact that the shape of the curved line of circles shown in the diagram is, as nearly as may be judged, the effective shape of a cross-section of a certain real wing (Eiffel No. 8 at 9 deg.), deducting an allowance for the thickness of the section. Also it is a fact that the spacing of the circles indicates the actual distribution of lifting force experienced by that wing between the front and the rear of the wing section. Also the component of horizontal velocity added to produce the dotted arrows is the velocity used in

[1] "Almost exactly," because the diagram contains an almost imperceptible allowance for the termination of the sustaining vortices within the length of the wing span and for the influence of their rearward extensions as trailing vortices. (Compare heading *g*.)

the published tests of this real wing, and the vortex strength used in preparing the diagram agrees with the vortex theory of lift (Par. 6) and the estimated circulation around the wing mentioned. It is evident from these figures that there is a close connection between the factors of distribution of fluid motion, and shape.

d HYDRODYNAMIC-ELECTROMAGNETIC ANALOGY

10 Diagrams like Fig. 3 are easily obtained, not by mathematical calculation, but by arrangements of electric currents and magnetic fields representing vortices and velocities, choosing any desired amount of flux to represent a standard velocity.

e ACTION OF VORTICES WITH REFERENCE TO EACH OTHER

11 The Encyclopedia Britannica gives formulæ for the action of groups of parallel columnar vortices upon each other, in terms of the strength of these vortices and their distance apart. It is interesting to estimate the strength of the trailing vortices from a biplane by observing their actions on each other. The pairs from the right-hand wing tips are of one kind or direction and revolve around each other in approximate circles. Those from the left-hand tips also revolve around each other, but in the reverse direction from the first-mentioned pair. Very careful experiment would be required to detect any error in the vortex theory of flight in terms of the action of these pairs of trailing vortices. The vortices can be seen in a smoky atmosphere when moving models are used.

f ACTION ON VORTICES WITH REFERENCE TO THEIR IMAGES

12 Many peculiarities of fluid motion are roughly explainable in terms of the action of eddies as if under the influence of their images in solid surfaces. For example, there is a remarkable difference in the circumstances surrounding the eddies formed at *B* and *B′* in Fig. 1 on the upper surface of the plane. The one at the rear *B′* should tend to pass off if considered as under the influence of its image. Conversely, the one at *A* should tend to remain with the plane.

g INFLUENCE OF THE LOCAL WIND

13 Vortices, although regarded as having their axes in some particular location, are usually considered as having an influence through the fluid in which they exist, just as the magnetic effect of

an electric current is considered as having an effect at remote distances. It is interesting to calculate the effect of the trailing vortices on a wing of short span. A wing in horizontal flight does not act as if encountering a horizontal onrush of the atmosphere. It acts as if there were a downward component of motion in the air around the wing, of very much the same amount that might be calculated from the strength of its own trailing vortices. This is manifest especially in the case of short wings by a correspondingly poor lift, as if something were reducing the angle of attack, and in a greater resistance as if climbing up through a descending wind. The hydrodynamic-electromagnetic analogy is capable of yielding interesting information in this connection.

h LAWS OF ENERGY CONTENT IN TRAILING VORTEX

14 Mathematically it would appear to be as easy to calculate the energy of vortex-motion lift in the wake of an airplane as it is to calculate electrical self-inductance. The arrangement of trailing vortices behind an airplane evidently depends considerably on the distribution of the loading along the wing span, because a wing can terminate in effect considerably short of the actual tip by an easing up of the lift. Calculations concerning the best arrangements would be interesting. The writer has made some approximate computation by assuming the trailing vortices to be a group of parallel columnar vortices: a sort of sheet of vortices constituting the wake of the wing. This method of calculation gives the usual values of the lift-drag ratio, when friction is taken into account.

i FRICTION AND HEAD RESISTANCE

15 In a practical way friction is a large item and it would be interesting to have separate tests for the friction losses of wings. Tests might be made of the resistance of a hoop or endless ring having the cross-section of a good wing.

j EXPLOSION OF EDDIES

16 Frequently the low pressure at the center of a vortex or eddy in real air appears to be penetrated by a rush of air along the axis. Knowledge about this, especially with reference to the effect, cause and control of such disturbances in the wake of wings, would be interesting.

DISCUSSION

John R. Freeman (written). The writer is hardly competent to discuss Mr. Durfee's paper in the language and symbols of mathematics, but a possible line of investigation of these phenomena occurs to him which may perhaps be useful.

About ten years ago when members of our Society were guests at a meeting in England of the Institute of Mechanical Engineers, the eminent mathematician, Dr. Hele-Shaw, presented an illustrated discussion on stream-line problems in air currents relating to airplanes, in which he showed the disturbing effect upon the streamline of models designed to represent various forms of wings. The fluid in that case was a liquid and the stream lines were represented by ingeniously colored liquid filaments. The investigation was along lines of previous investigations of the distribution of velocities in flowing liquids containing colored liquid threads which showed the eddy currents caused by such obstructions as bridge piers.

At that time it seemed to the writer that the difference in compressibility of air and water, and also the difference in inertia effect, impaired the analogy, and on further thought he laid out a line of experiments for tracing the motion of liquids around obstructions in channels by a combination of methods borrowed from the ultramicroscope and the moving-picture machine, although he has never found time to carry out the experiments. This method seems admirably adapted to experiments on air currents in connection with airplanes.

The method in brief is to make an optical cross-section in any desired plane by means of a thin, broad beam of intense light put into appropriate shape and parallel rays by proper condensing lenses, and an optical slit, analogous to that used with the ultramicroscope. By means of dust particles of proper density introduced in the air current, one can render visible the direction and velocity of the currents set up somewhat as he sees the air currents in his living rooms made evident by a sunbeam acting upon the suspended dust particles.

The narrow slit of light reveals only the motions in one plane and simplifies the observation by rendering the particles visible only while traveling in this optical plane.

In the case of the airplane, these motions would mostly be too rapid for the eye to follow, but within limits it is possible to observe them and to record them by a motion-picture apparatus which can be so constructed as to make 30 or 50 exposures per second instead of the customary 16. It is indeed possible to obtain exposures of much shorter frequency by means analogous to the shutter-testing device developed in Dr. Mees' Research Laboratory at Kodak Park, in which a rapidly revolving polygon of mirrors serves to catch and record the fleeting image. Also there have been devised and patented means of revolving polygonal prisms of glass, so adjusted that their reaction holds the image approximately stationary on the screen or film for the fraction of a second.

Such a series of photographs recorded in a short reel of film can be rotated for purposes of study at a much slower speed than that at which they were taken.

By these means, the writer believes the actual pathway of the particles of air as they pass either wing plane or propeller can be made evident and precisely recorded at velocities far higher than can be observed in any other known way, and a series of optical sections, analogous to those of the ultra-microscope or the sunbeam will simplify the hopeless complexity of a dense mass of particles traveling in various directions.

Edward P. Warren (written). It is manifest that any considerable development of the theory of wing action beyond the point already reached must be conditional on the use of new and more powerful and logical methods of attack. In most of the work so far done, whether by the simple assumption of plane impact and reflection or by such more elaborate methods as that of Kirchhoff and Helmholtz, the continuity of the air has been ignored, and the results have consequently been far from the truth.

The work of Lanchester, Kutta, and others on a vortex theory of sustentation seems to offer the most promising path to an analysis of wing action which shall be of real practical use. It leads to the only method which takes due account of the fact that there can be no actual acquisition of downward momentum by the air as a whole, since the center of gravity of the atmosphere cannot shift, and any downward motion imparted to the air in the neighborhood of the wing must be counterbalanced by an equal upward motion imparted to an equal mass at some other point.

Promising as the vortex theory is, however, it should not be overrated. There are many factors in the action of wings for which it does not appear to account, and the mathematical weapons are not at hand for applying it, except in the simplest cases. The electromagnetic analogy proposed by Mr. Durfee is very interesting, but it must be handled with care, particularly in connection with thick wings, where the air-flow changes from stream-line to turbulent types and back again with the greatest suddenness and in response to the minutest alterations of wing form or conditions of operation. It is doubtful if this analogy could be extended to any cases beyond those of the flat plate and the simplest forms of thin, cambered sections.

George de Bothezat[1] (written). The statement "*a*" of Mr. Durfee's paper constitutes in reality the well-known "Kutta theorem," discussed by Kutta himself (*Illustrirte Aeronautische Mitteilungen*, 1902; Sitzungsberichte der Königlichen Bayerischen Akademie der Wissenschaften, 1910 and 1911); by Joukowski (*Aérodynamique*, Paris, 1916) and Dr. de Bothezat (Report No. 28, Note I, from Fourth Annual Report, National Advisory Committee for Aeronautics).

The statement "*c*" was first made by Lord Kelvin with reference to an example actually classical (the so-called atmosphere around a system of two rectilinear and parallel vortices rotating in inverse sense).

The statement "*d*" of the hydrodynamical-electromagnetic analogy is well known. But the suggestion to study the flow around an airfoil by this method is of interest, and such experiments conducted in a suitable manner could bring valuable results.

A solution of the question proposed in statement "*e*" is directly obtained by the successive application of the Kutta theorem, Lord Kelvin theorem on the constancy of circulation and the Stokes theorem connecting circulation with vortex intensity. (See Report No. 28 of the Fourth Annual.)

The statement "*g*" is not quite clear; if local wind means only the instantaneous value of the fluid velocity at a given point around the airfoil, it is only a regular conception.

The statements "*h*" and "*i*" demand very careful consideration, because it seems that in the case of hydrodynamical phenomena some special conditions may occur which we do not meet in electromagnetic phenomena.

[1] Aerodynamical Expert, National Advisory Committee for Aeronautics, Washington, D.C.

F. W. CALDWELL[1] (written). This paper is very timely, particularly in view of the growing tendency among aeronautical engineers to regard the classical coefficients K_x and K_y as inadequate.

It has been almost universally the practice to write $L = \frac{\rho}{g}K_ySV^2$ and $D = \frac{\rho}{g}K_xSV^2$ where L is the lift, D the drag, S the area of the supporting surface, V the velocity of advance, ρ the density of the air in weight units, g the acceleration due to gravity, hence $\frac{\rho}{g}$ the density of the air in slugs.

It is well known as the result of experience that the values of K_y and K_x vary somewhat with velocity and also with the size of the surface under consideration. If l represents one of the linear dimensions of the surface it is assumed that the values of K_y and K_x are functions of the product Vl. This is known as the scaling effect.

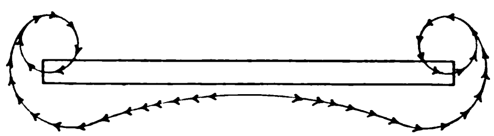


FIG. 4 DIAGRAMMATIC REPRESENTATION OF FLOW

Information on the scaling effect is very meagre. In the case of propeller sections we have been forced to make use of characteristics obtained at a speed of 30 miles per hour and apply them to conditions where the speed obtained is as great as 600 miles per hour. While the use of scaling rules and empirical factors worked out in practice have enabled us to produce very fair results, the need for accurate data has been pressing.

The high-speed wind tunnel operated by the Technical Section of the Department of Military Aeronautics is of the venturi type and shows an exceptionally uniform flow at all speeds up to about 500 miles per hour. The flow is produced by means of an especially designed 24-blade propeller which produces a suction of about 16 inches of water at the large end of the venturi.

An extensive series of experiments has been started on propeller airfoils in order to determine the effect of speed on the lift and drag coefficients.

It is desired to call attention to the fact that it is not sufficient to understand and be able to calculate the circulation about a hori-

[1] Aeronautical Engineer, 330 Edgewood Ave., Dayton, O.

zontal axis perpendicular to the direction of motion; there are also very important vortices about axes parallel to the direction of motion. These are particularly apparent in the form of tip vortices. Their intensity might be estimated similarly to the calculation for fore and aft circulation.

The flight vortices have been visualized for the first time in the army wind tunnel and a moving picture of the phenomenon has been made. While the moving pictures show up very well on the screen it has, unfortunately, been impossible to make them show up well in print.

Fig. 4 shows the type of flow corresponding to normal flight conditions. It is supposed to represent a view of the plane taken from upstream.

F. E. Cardullo. As I understand the theory which Mr. Durfee is attempting to develop, instead of considering the reactions due to the acceleration produced under the action of the motion through the air of a plate, which may be straight or curved, he considers these reactions from the standpoint of the Bernoulli theory, on the basis that differences in velocity of air relative to the plate will exist on its two sides, and that in consequence to these differences there will be a difference of pressure and a lifting force. He points out that there are certain vortex motions at the ends and at the trailing edges of the plate. The best method of attack would seem to me to be that of testing wing sections either by the emission of smoke from a fine orifice or with threads attached to needles. This method gives an opportunity for studying the vortices which represent irregular motion and lost energy. Mr. Durfee prefers to attack the problem from the standpoint of the investigation of these stream lines and velocities rather than the development of the theory of the reaction on the surface produced by acceleration. I do not know that there is much choice in the mathematics of the two methods, but in either case the mathematics is too difficult to offer a practical solution.

The Author. Mr. Warner's remarks with reference to the sensitive fluctuations between one type of motion and another indicate the probable cause of much uncertainty and doubt connected with aerodynamic science. Although the action of a first-class wing is comparatively simple and easy of analysis the action of a poor wing is likely to be intricate and difficult of analysis, even though the shape may be simple in the last case and refined in the other.

At least three or four types of motion might be named and described. There is first the type of motion which occurs theoretically in the case of a perfect fluid and which may be observed, according to Lord Kelvin, in any fluid at the instant when the wing or other object is first put in motion. Radical modification of this ideal system of motion may take place owing to the formation of eddies. This may result almost immediately in an alteration of the effective shape of the body by means of eddies which are carried along by the object in question, perhaps on the under side of a wing, sharply curved, and on the upper side of a wing, slightly curved, or in the rear of objects bluntly shaped. Either with or without the above mentioned modification (type two), a third type of motion is produced in the case of successful wings. This is a circulation around the instantaneous position of the wing in such a manner that the atmosphere below the surface moves forward (with reference to the still air through which the wing is flying) and the atmosphere above it moves backward. The creation, existence, and preservation of this circulation is intimately connected with the vortices which Mr. Caldwell mentions as parallel to the direction of motion. As it is evident that such trailing vortices must decay through friction and other causes, and that the eddies mentioned under the second type of motion must also decay or suffer violent destruction or be swept away, it follows that violent fluctuations in the type of air flow are likely to occur in an irregular manner. The causes which determine which types or type of flow will occur or to what extent each of several will occur in combination probably have much to do with the scaling effect mentioned by Mr. Caldwell. The scaling effect is probably influenced also by the fact that the drag of a wing is composed partly of friction and partly of a rearward component in the reaction of the sustaining vortex. The following table shows this.

C_L, coefficient of lift	absolute unit	= 0.5 (units of force per unit density and area at unit velocity)
	French "	= 0.0625 (kilograms per square meter at one meter per second)
	English "	= 0.00255 (pounds per square foot at one mile per hour)

Corresponding strength of vortex	= $\frac{1}{4}$ (breadth of wing) × (velocity of wind)
Corresponding mean velocity over top of wing with reference to wing	= $\frac{9}{8}$ (velocity of wind) plus (thickness correction)
Corresponding mean velocity over bottom of wing with reference to wing	= $\frac{7}{8}$ (velocity of wind) plus (thickness correction)
C_F, approximate coefficient of friction in absolute units	= 0.005 (or 0.0025 for each side)
C_{Da}, aerodynamic drag coefficient or coefficient of drag due to vortex of reasonably good distribution and for aspect ratio = 6, and $C_L = 0.5$	= 0.030
C , coefficient of drag in absolute units = $C_F + C_{Da}$	= 0.035
Calculated reasonably good lift drag ratio for aspect ratio = 6 at value of $C_L = 0.5 = C_L \div (C_F + C_{Da})$	= 14.3

Friction is an important part of the drag especially at low angles when the drag due to lift is small. But the coefficient of friction is not a constant at different speeds when it is defined like the other coefficients as a force divided by area, density and squared velocity. For this reason one may expect that high velocities and large dimensions will result in better lift-drag ratios.

The analysis with reference to the direction of the reaction of the sustaining vortex is improved by the concept which I have spoken of under the heading of the local wind. This concept is not exactly with reference to the instantaneous value of the fluid velocities at a given point but is with reference to conditions which surround the wing or blade. For example, in the case of the upper wing of a bi-plane there is a circulation of the atmosphere around the lower wing in such a direction as to increase slightly the intensity of the wind which acts against the upper wing. It is the apparent velocity and direction corrected by such influence which I would speak of as the local wind with reference to this upper plane. In this particular case this local wind with reference to the upper plane contains also certain components of nearly vertical velocity due to the existence of the four vortices which trail behind the four tips of the wings. In other words a wing appears to suffer from the downward motions due to the existence of the trailing vortices in the same sense that the tail-plane suffers; but in lesser degree. In this manner it is possible to understand the peculiarities of aspect ratio, since by lengthening the wing a greater proportion of the span is placed in a region remote from the tip vortices so as to be influenced less severely. Presumably the blades of a propeller or fan may be similarly considered as moving in a local wind of an intensity and direction determined by the complicated vortex system which constitutes the blast. The local wind with reference to the blade then is not precisely the velocity at that point because that velocity is in part made up of the circulation around the blade itself. By local wind I mean the conditions which in effect act on that blade. A precise definition of this concept might be to say that the local wind is indicated by the velocity and direction in which the sustaining vortex would itself move if the wing or blade should vanish.

Mr. Warner's remarks as to the care with which the electromagnetic analogy must be used are very much to the point. The experiments must be conducted, as Doctor de Bothezat remarks, "in a suitable manner." The air will not follow the desired and ideal type of motion if conditions are permitted to exist more favorable to another kind of motion. Confining one's self to arrangements of vortices, all of the desired kinds as shown in Fig. 3, it is not difficult to postulate our arrangement (by crowding the vortices nearer the leading edge) so that the fluid motion will conform in theory to the shape of a flat plane, but the resulting diagram is of such a character as to suggest the probable formation through friction of an intense eddy at the leading edge. This appears to be the case in practice; and the effective shape of the wing appears to be changed by the presence of this eddy and the electromagnetic analogy impaired by the difficulty or impossibility of producing such circumstances. On the other hand, by attempting to rearrange the electromagnetic model so as to avoid unfavorable conditions one soon discovers a series of shapes and typical designs which are well known to be good. Futhermore, the analogy can furnish useful information with reference to the local wind at various points. It is not altogether inapplicable to moderately thick shapes because, according to Lord Kelvin, there is considerable similarity between the fluid motion around a solid body and around a group of vortices.

Mr. Caldwell's remarks as to the intensity of the tip vortices are of interest. These vortices may be observed in a smoky atmosphere very readily by using a model which is driven through the air guided by a cable trolley. The intensity, so far as I have been able to measure it, is the same as the theoretical intensity of the sustaining vortex. The measurement of intensity is made by using a small biplane and observing the rotation of the pairs of trailing vortices around each other; for example, by observing the rotation of these from the right-hand tips. It is interesting to inquire in what manner the low pressure is preserved along the axis of these trailing vortices. This inquiry leads one to become interested in the vortex which joins the trailing vortices together at the rear as shown in Fig. 2. Presumably this vortex is renewed from time to time so fast as the original system breaks down.

Mr. Freeman's remarks with reference to a method of observations derived from the ultra-microscope and the moving picture machine are very attractive. By making the observation in still air with a moving model it would seem possible to learn very much concerning the formation and decay and internal arrangements of the entire vortex loop.

In answer to a question by Professor Cardullo as to whether or not the mathematical method was the only practical one, the author stated that he believed that there were many useful experimental methods. The present paper was mainly an analysis of the various elements in the action of a good wing, intended to be useful as a guide in many lines of investigation.

No. 1705

AIR FANS FOR DRIVING ELECTRIC GENERATORS ON AIRPLANES

By Capt. G. Francis Gray[1], U. S. A.

Lt. John W. Reed[1], U. S. A.

and

P. N. Elderkin[1]

Non-Members

In this paper the authors briefly describe the method employed by the Radio Development Section of the War Department in testing air fans used for driving the electric generators usually installed on airplanes for radio communication. They also discuss at some length the various types of air fans and present numerous photographs and curves clearly illustrating the construction of the fans and their operating characteristics.

The difficulty of the problem lay in designing a fan which would turn at constant speed in the air streams of widely varying speed set up by the airplane in flight. The various types of fans tested were: Fixed-blade fans of special blade shape; fixed-blade fans with wind brakes centrifugally regulated; fixed-blade fans using a friction clutch or a friction brake centrifugally regulated, and pivoted-blade fans in which the pitch is centrifugally regulated.

DURING the war extensive use was made of radio telegraph and telephone apparatus on military airplanes, and the problem of power supply for such equipment received a great deal of attention. The possible sources of energy may be listed as follows:

a Storage batteries or dry batteries

b Generators driven from the airplane engine, with or without floating storage batteries, and supplying the radio sets directly or through dynamotors

c Generators driven by separate gasoline engines

d Generators driven by air fans or "windmills," placed in the air stream outside the airplane fuselage.

2 From an economical point of view, method *b* is preferable.

[1] Engineering and Research Division, Radio Development Section, War Department, Washington, D. C.

Presented at the Spring Meeting, Detroit, Mich., June, 1919, of The American Society of Mechanical Engineers.

It was seriously considered but involved coöperation between organizations normally operating independently and its adoption was delayed. Meanwhile the practice in our Army followed that of our Allies, principally the French, in the use of method *d*, mounting the generator outside the airplane fuselage and driving it with an air fan.

3 This paper reviews the work done on the development of air fans for this service as carried out by the Radio Development Section of the Signal Corps. As the work was done under the press of military necessity it was directed entirely by utilitarian considerations, was often fragmentary, and neglected investigations, the need of which was realized but for which time and personnel were not available. This record is presented with the work still in unfinished form in the hope that results obtained may be useful to those who may have occasion to carry out further investigations on the problem.

Fig. 1 Air Fan and Test Generator Mounted in Wind Tunnel

CONDITIONS FOR WHICH THE AIR FANS WERE DESIGNED

4 Two sizes of generators were to be driven by the air fans it was desired to develop. The essential data on these are as follows:

	Generator for Radio Telegraph Sets	Generator for Radio Telephone Sets
Generator diameter, in.	$6\frac{1}{8}$	$4\frac{15}{16}$
Generator output, watts	200	80
Required power from fan, watts	330	250
Normal speed, r.p.m.	4500	4000

METHODS OF TEST

5 Practically all of the tests recorded in this paper were made with the wind tunnel of the Bureau of Standards, and the management and personnel of the Bureau aided materially in expediting

FIG. 2 TEST GENERATOR WITH STREAMLINE CASING

them. A special testing generator was mounted in the wind tunnel as shown in Fig. 1, and the fan to be tested was attached to it. This generator was provided with a magnetic tachometer, a separately excited field, and convenient means for applying load to the armature circuit. The external shape and size of the machine were made identical with those of the radio generators, with which the air fans were to operate in service, by the addition of the molded micarta streamline casing shown in Fig. 2. With this generator and the regular wind-tunnel equipment for measuring wind velocity, tests could be made rapidly and accurately.

6 All of the air fans considered in this development have normal speeds of 4000 r.p.m., or above, and the centrifugal forces are very

considerable. It was therefore found necessary to provide an overspeed test to precede the wind tunnel test. For this purpose a 30-hp. motor operating through a 43 to 3 De Laval speed-increasing gear to a suitable shaft extension for the fan was set up. This device was capable of driving the air fans at speeds up to 14,000 r.p.m. and to prevent damage to gears or bearings in case of failure of the fan, a "weakest point" was provided by a replaceable notched-shaft extension. Failure at this point merely resulted in the breaking of the replaceable shaft extension, and of course the destruction of the fan, without injury to bearings or gears. Heavy guards prevented the flying fragments from causing any

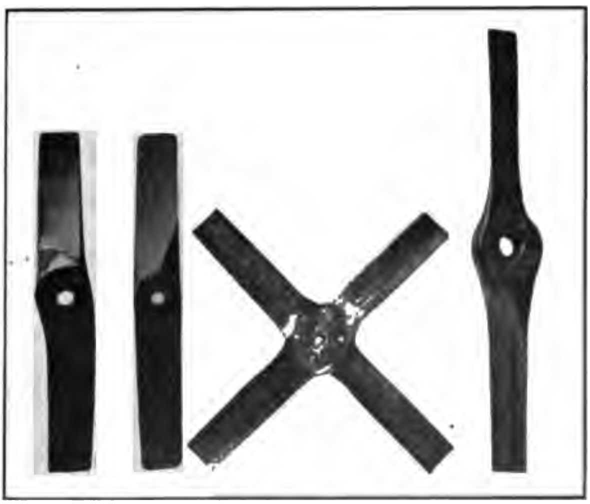


Fig. 3 Pre-War Air Fans

damage. This equipment was designed by Capt. F. E. Pernot, Officer in Charge of the Signal Corps Laboratories at the Bureau of Standards.

7 There is one serious objection to this method of overspeed testing in that it absorbs a large amount of power and puts abnormal thrust strains on the fans. A baffle, to prevent air flow materially reduced this effect, and later experience indicated that it would have been preferable to make the overspeed test in a very simple auxiliary wind tunnel with an aperture just large enough to drive the unloaded fans at the speed desired. Nevertheless the apparatus just described was used in all tests discussed in this paper.

8. In cases where the strength of a fan was problematical, it was first given an overspeed test only slightly in excess of the running speed expected in the wind tunnel. After the wind-tunnel test it was again tested to the required overspeed or to destruction. The best fans were required to withstand up to practically double normal speed.

SIMPLE NON-REGULATING AIR FANS

9 As in all power-plant problems, the ideal in this development was a constant-speed drive for the radio generator. When the development began, the only air fans available were non-regulating

Fig. 4 Type FA-3, Non-Regulating Air Fan for Radio-Telephone Generators

wooden fans such as those illustrated in Fig. 3. These fans have the characteristic that rotational speed is practically proportional to air speed, and when combined with the normal variations in the speed of the airplanes this fact leads to very severe requirements for the generator. As an example, a generator which would give satisfactory performance at 3750 r.p.m. was required to withstand an overspeed test of 14,000 r.p.m. for mechanical performance and, with the addition of a very special regulating device, to hold its voltage within 15 per cent at speeds from 4000 to 12,000 r.p.m.

10 Although the need for better air fans was obvious, the non-regulating types had to be used until others could be developed, and a considerable amount of work was done in adapting them to the

particular requirements to be met. Fig. 4 shows the construction of one of the final forms used with generators for radio-telephone sets, and Fig. 5 its performance in the wind tunnel. Figs. 6 and 7 give similar information on a larger-size fixed-blade fan used for radio-telegraph sets. Fig. 7 also gives an idea of the variation in individual fans supposed to be identical. Considerable trouble was experienced with these fans due to the fact that the method for calculating air fans was assumed to be the same as that used for propellers. That this is not the case and that errors in using propeller formulæ are considerable is shown by the following data.

11 A fan of the type shown in Fig. 6 was calculated by simple

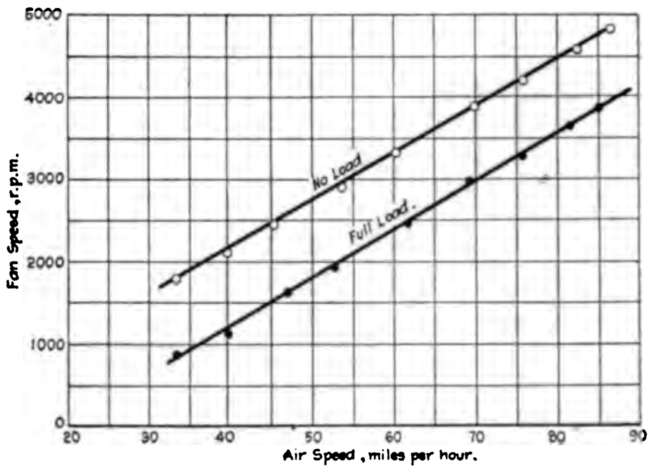


Fig. 5 Performance Curves of Type FA-3 Air Fan

(Diam., 15 in.; pitch, 2.1 ft.)

helix formulæ to have a pitch of 2 ft. Theoretically this fan when unloaded (i.e., zero slip) should run at 4500 r.p.m. in a wind stream of slightly over 100 miles per hour, or assuming 18 per cent slip under load, the wind stream required to bring the fan up to speed should be approximately 120 miles per hour. But this fan actually runs at full speed under load in an 82-mile-per-hour wind. The calculated value of the normal-speed air under load is thus 45 per cent high, and if the slip is actually the assumed 18 per cent, the non-slip air speed for normal rate of rotation is 70 miles per hour instead of 100 as calculated, which gives the effective pitch of this fan as 1.37 ft. instead of 2 ft., the value given by the propeller calculations.

12 The explanation of these results by propeller designers is

that the no-lift pitch differs from the blade pitch, a phenomenon well understood but not applied to air fan design at the time these tests were made. The magnitude of the differences and the lack of information regarding the correct numerical constants for design

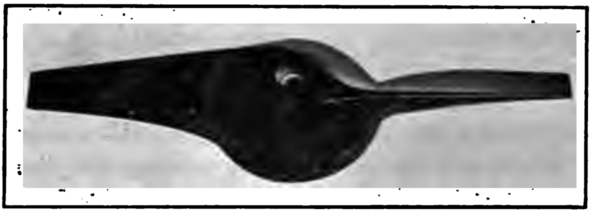


FIG. 6 TYPE FA-6, NON-REGULATING AIR FAN FOR RADIO-TELEGRAPH GENERATORS

formulæ lead to the use of entirely empirical methods of design for the fans that were put into service.

13 Obviously it is desirable to establish the correct method of calculating these fans, but the pressure of more important work

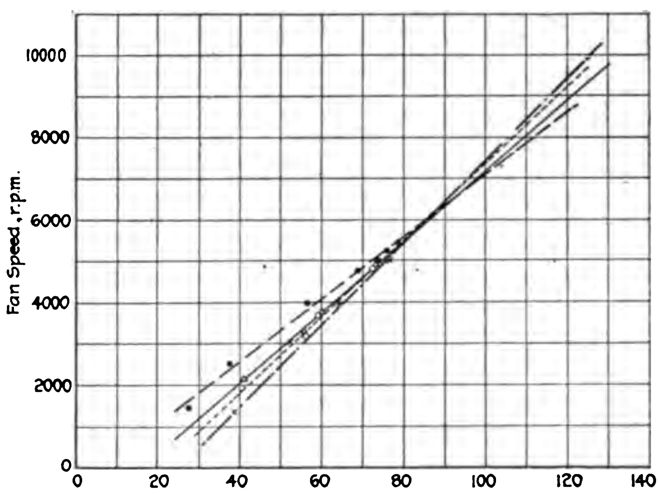


FIG. 7 PERFORMANCE CURVES OF TYPE FA-6 AIR FAN

(Diam., 20 in.; pitch, 1.75 ft.)

prevented its being undertaken by the Radio Development Section. The subject was given much attention by Mr. E. N. Fales, of the Airplane Engineering Department, Bureau of Aircraft Production of the Army, who assisted the Section very materially during the early stages of the development described in this paper, and it is hoped he will publish his conclusions.

REGULATING AIR FANS

14 No one can work with simple air fans without being impressed with the desirability and apparent simplicity of modifying them so that instead of the straight-line relation between air speed and speed of rotation there will be more or less tendency toward constant rotational speed with varying air speed. Among the disadvantages which may be corrected by making the air fans self-regulating are the following:

- *a* Undue head resistance
- *b* Excessive centrifugal stress in armature windings
- *c* Vibration
- *d* Commutator and bearing troubles
- *e* Troubles with voltage regulators
- *f* Troubles in radio sets due to varying voltage.

During the progress of this development a great many schemes for the construction of self-regulating fans were proposed. As all previous experiences led to the conclusion that the performance of a fan could not be predicted with any degree of accuracy, every reasonable suggestion was carefully considered and if possible tried out. Fundamentally all the fans tested may be classified under five principles of operation as follows:

- *a* Fixed-blade fans of special blade shape
- *b* Fixed-blade fans with wind brakes centrifugally regulated
- *c* Fixed-blade fans using a friction clutch centrifugally regulated
- *d* Fixed-blade fans using a friction brake centrifugally regulated
- *e* Pivoted-blade fans in which the pitch is centrifugally regulated.

15 *Fixed-Blade Fans of Special Blade Shape.* Certain air fans submitted to the Signal Corps were supposed to give speed regulation on the principle that the angle of attack and efficiency can be made to alter with the wind velocity and in such a way that the ratio of wind velocity to r.p.m. need not be constant. Tests were made first on a series of blade sections and later on a fan, shown at the right in Fig. 3, which was said to possess the regulating

features in a marked degree. The conclusion from these tests was that at no load the performance of properly designed fans tends to confirm the hopes of the designers though the magnitude of the regulating effect is small; but under load, particularly varying load, it was impossible to distinguish any improvement over the ordinary fixed-blade types of fans. Consequently this method was discarded as of no practical value.

16 *Fixed-Blade Fans with Wind Brakes.* Fans of this type are provided with wings, pins or other projections which are normally enclosed in a recess in the blade, but which are caused by centrifugal force to emerge and so retard the rotation of the fan. They

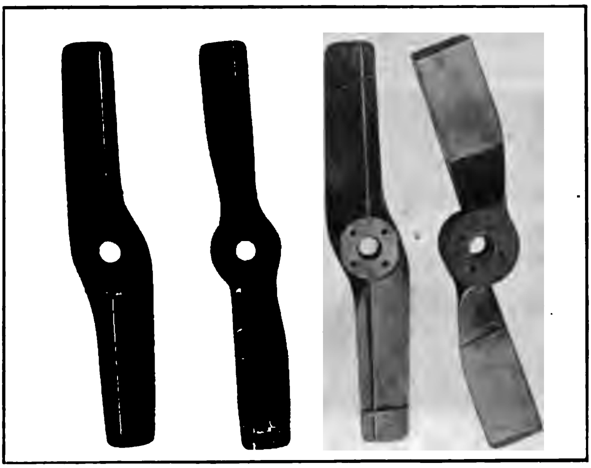


FIG. 8 FIXED-BLADE FANS WITH WIND BRAKES

have the disadvantage of low efficiency and consequent high head resistance, but were considered worth trying. Fig. 8 shows various development models. Round rods, rods with cloth and metal wings, flat strips, etc., were tried for the moving brake element. Many showed practically no regulation of course, but others were reasonably promising. Among such was the type in which the brake arms were pivoted and moved out into the air by the action of centrifugal weights. Another interesting fan had the brake arms actuated not by centrifugal force but by the air pressure on a plate mounted in advance of the nose of the fan, and gave a curve in which, for a certain range, the r.p.m. actually decreased with increase in air velocity.

17 The net conclusion from this research was that the windbrake principle would give fairly good characteristics over a limited range of air speed, and with further refinement it might have been worth putting into production. However, that stage was never reached due to the work on the pivoted-blade type of fan described below.

18 *Friction-Clutch and Friction-Brake Fans.* As in the types of regulating air fans discussed above, the friction-clutch and friction-brake principles of operation are inherently objectionable on account of their low efficiency and consequent high head resistance. They offered a possible solution, however, and were tried out experimentally. In the friction-clutch type of fan the fan hub is capable of free rotation about the shaft of the generator, which it drives

FIG. 9 TYPE FA-4, VARIABLE-PITCH AIR FAN

through a friction clutch. The clutch is released by the action of centrifugal weights when the speed passes the predetermined maximum, permitting the fan to run at higher speed than the driven shaft of the generator. The curve of this fan was satisfactory for the first model, but the heating due to slipping of the clutch at high velocities was too great to permit the use of the fan in practice and the development was abandoned.

19 In the friction-brake type of fan the fan is keyed to the generator shaft and is prevented from exceeding normal speed by a brakeshoe operated by a centrifugal weight and bearing on a plate attached to the generator frame. This also was tried, and gave results essentially similar to those obtained with the clutch type. With the very considerable variation in air speed met with on airplanes,

the friction devices generated so much heat that their use was impossible.

20 *Variable-Pitch Air Fans.* The ideal principal for the design of a regulating air fan is that of varying the pitch of the blades to correspond to the variation in air speed; this principle is by no means new, having been considered in a variety of forms for propellers for a number of years. The great difficulty in actual construction has been that the mechanical strength necessary to withstand the very high centrifugal forces and the delicacy of operation necessary for close regulation are very hard to combine.

21 This objection does not apply so forcibly to fans using very thin blades which are expected to warp under centrifugal action to change the pitch and a few samples which it was hoped would operate on this principle were tested. They were unsuccessful principally on account of mechanical construction and improved samples were never made up. The method is in any case of doubtful utility since it can probably take care of only limited variations in speed, whereas the pivoted blade fans are capable of regulating over the widest variations in air speed likely to be met with in airplane practice.

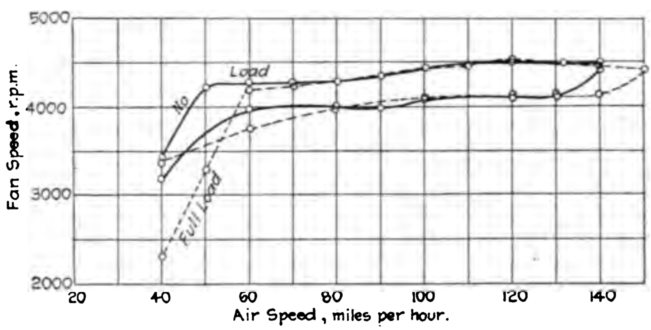


FIG. 10 PERFORMANCE CURVES OF TYPE FA-4 AIR FAN

22 *Pivoted-Blade Air Fans.* The earliest pivoted-blade fan to come to the attention of the Radio Development Section was made by the Sperry Gyroscope Company, but it proved unsuccessful, due to mechanical weakness. It was, however, the forerunner of very successful fans, and serves to illustrate the general principle on which all operate. The blades are mounted on bearings and are capable of rotating through a considerable angle. Centrifugal weights are mounted on arms attached to these blades and tend to

turn them in the proper direction to increase the pitch. A resisting spring and necessary hub complete the mechanism.

23 The first model to perform satisfactorily is shown in Fig. 9. It was designed by Mr. Thomas Slate, of the American Mechanical

Fig. 11 Type FA-4-A, Variable-Pitch Air Fan

Improvement Company, Washington, D. C., and its performance is indicated by Fig. 10. The very great improvement over the fixed-blade fans then in use made the production of this fan highly desir-

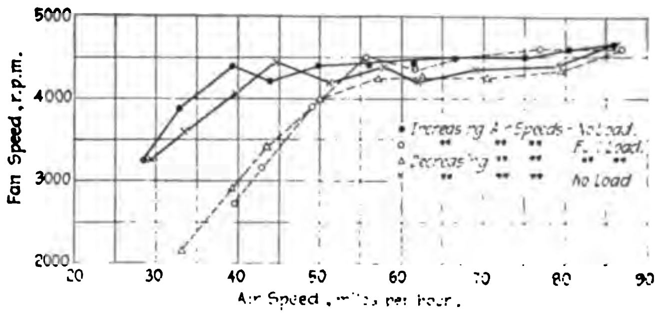


Fig. 12 Performance Curves of Type FA 4-A Air Fan

(Diam., 20 in.; pitch variable.)

able, and it was undertaken at once under purchase specifications as follows:

Operating air-speed limits .... 15 to 300 m.p.h.
Normal speed .... 4500 r.p.m.
Speed variation less than .... plus or minus 4 per cent
Overspeed test .... 8000 r.p.m.

It was of course realized that this fan was by no means in its final form, and work toward improving it and providing new sources of production was carried on as rapidly as possible. One of the objections to the original design was that the particular mechanism used

FIG. 13 TYPE FA-8, 200-WATT, SINGLE-BLADE, VARIABLE-PITCH AIR FAN

caused the centrifugal force to increase as the sine of twice the angle, while the spring restraining force increased according to a straight-line relation. In the second successful design this was corrected by the use of a special linkage between spring and blade

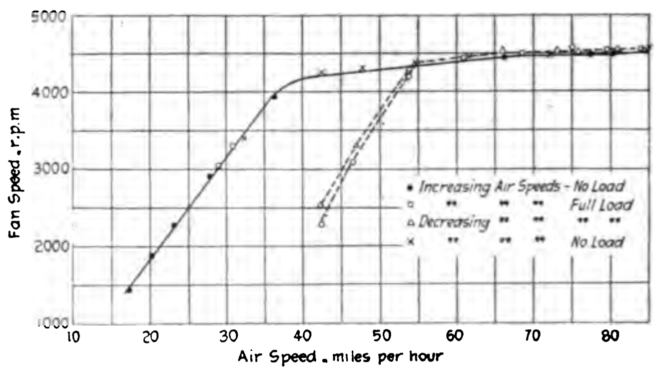


FIG. 14 PERFORMANCE CURVES OF TYPE FA-8 AIR FAN

(Swept diam., 20 in.; pitch variable.)

which modified the curve of spring resistance to fit that of the centrifugal weight. Fig. 11 shows one of these fans and Fig. 12 its performance. In this as in many other cases of radio development work, the design was conceived and outlined by the Signal Corps engineers and the details worked out at once by the manufacturer (in this case the American Propeller and Manufacturing Company) in a form ready for immediate production.

24 The third successful design of variable-pitch air fan was the work of Mr. Pinaud, of the Des Lauriers Aircraft Corporation

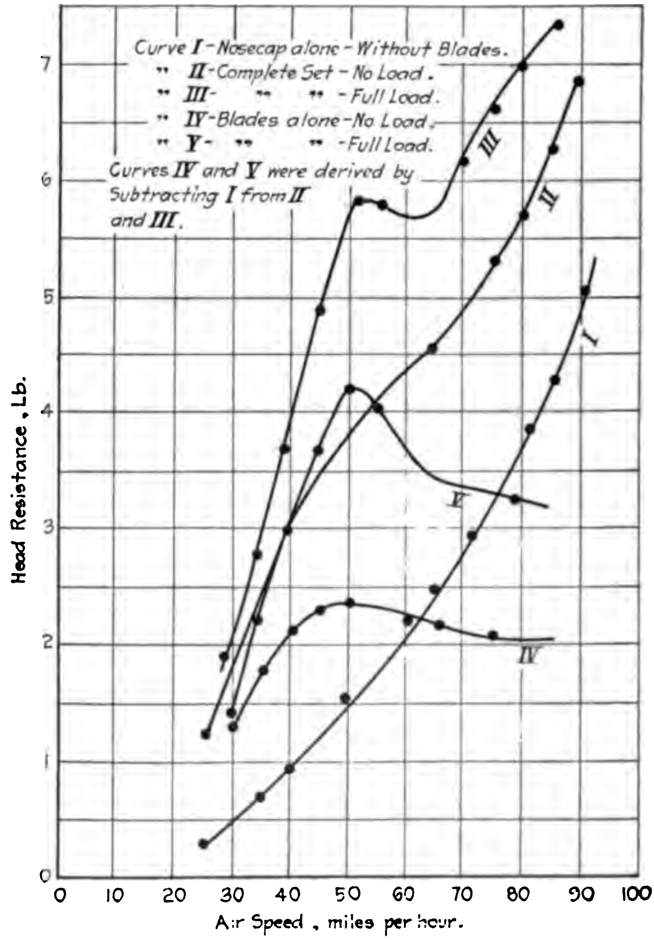


Fig. 15 Head-Resistance Curves, Type FA-4 Air Fan

and differs very radically from those previously described. It uses only one blade, and its principal advantage is that by ingenious counterbalancing practically all strain is taken off the bearings whereas in other forms the thrust bearings must carry a very considerable load — both radial and thrust — due to the high speed of rotation, and the unbalanced designs of the blades. Friction, of course, results in considerable "lag" in the performance of the fan. The mechanical construction of the Pinaud fan is also a considerable improvement over its predecessors in simplicity and ease of manufacture. Figs. 13 and 14 show its construction and performance.

25 In addition to the three types of regulating air fans which were sufficiently perfected to justify production, a number of others were considered and may be of interest, although the development was not carried to the point where the fans could be put into production. A very beautiful but delicate all steel design was made by the General Electric Company; the Frederick Piere Company made a fan essentially similar to the FA–4A type discussed above

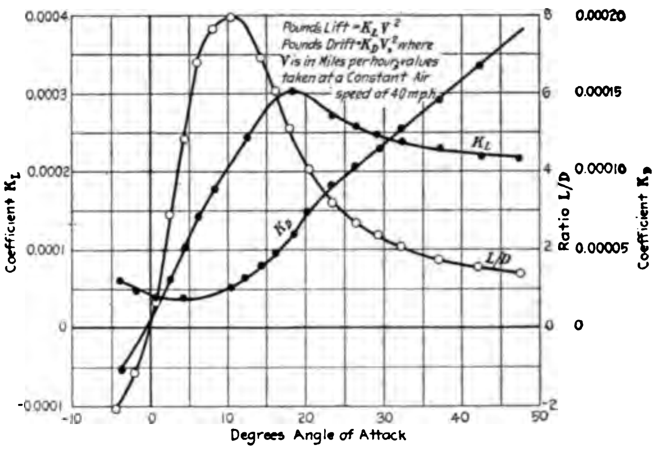


Fig. 16 Airfoil Curves of Type FA–4–A Air Fan Blade

except that the blades were of sheet steel welded to steel shafts and weighed little if any more than the cast aluminum blades; the National Electric Signal Company produced a fan interesting theoretically but very complicated and comparatively slow in action; the Air Motors Corporation made a design using either two or three blades and especially notable in that the hub was only about 1 in. thick instead of 3 in. or more as in other designs. The Sperry Gyroscope Company overcame bearing difficulties in a very ingenious manner. Their fan is of the two-blade design with blades offset and geared together as in the types previously discussed, but they have a distinct improvement in the way they take up the thrust due to centrifugal action. Instead of thrust bearings, a steel-rod tension member is used. It is located in the axis of the blade and is rigidly fixed to the hub at one end and to the outer end of the blade at the other. In addition to carrying the centrifugal load, this rod is twisted as the blade turns and its torsional force acts instead of the spiral or leaf spring of the other designs to oppose the twisting effort of the centrifugal regulating weights. The elimination of the friction in the thrust bearings undoubtedly increased the sensitivity of the fan, but, in the samples so far tested, it was found difficult to obtain sufficient strength combined with sufficiently low torsional force and high permissible angle of twist within the dimensions considered suitable. A further difficulty was met in the built-up blades used, but this could undoubtedly be easily overcome.

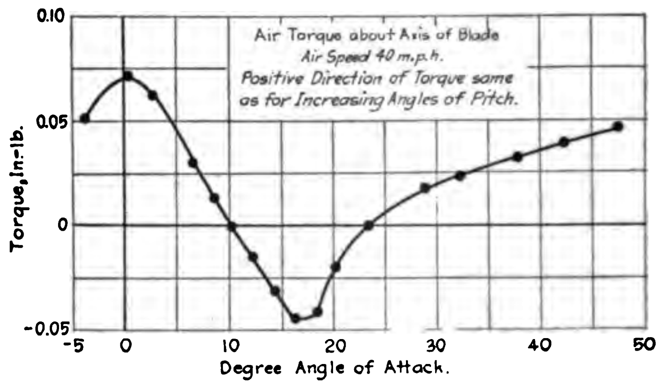


Fig. 17 Air Torque Curves of Type FA-1-A Air Fan Blade

HEAD-RESISTANCE TESTS

26 A demonstration of the value of the regulating air fans in reducing head resistance was made by the direct test of a complete radio-telegraph transmitting set in the wind tunnel. A run was first made with the set equipped with a fan hub and nose cap but no blades; then a second run was made with a fan operating normally with the generator both loaded and running light, and finally a third run was made with the fan blades locked at several values of pitch. The results of these tests are shown in Fig. 15, from which the following conclusions may be drawn:

27 The head resistance of the set fully loaded in an air stream of 75 miles per hour is 6.5 lb. At this air speed the resistance of the blades alone equals the resistance of the body, after which the resistance of the blades remain nearly constant while that of the

body increases as the square of the air speed. The body alone has 5 lb. head resistance at an air speed of 90 miles per hour, and at the usual ratio of $\frac{1}{8}$ hp. from the engine per lb. of head resistance, this requires 0.36 hp. from the engine. On a basis of 20 lb. in the fuselage per hp. from the engine, the set could weigh 13 lb. more than it now does and still be no more load on the engine provided it was mounted in the fuselage and obtained its power directly from the engine. This emphasizes the desirability already mentioned of obtaining all electric power by direct drive from the engine rather than from air fans where it is possible to do so.

AIRFOIL TESTS ON BLADES FOR REGULATING AIR FANS

28 In designing blades for regulating air fans the effect of wind pressure in producing torque around the blade axis has been questioned, and obviously such an effect might seriously interfere with the performance of the fan. To obtain data on this point tests were made on blades from a fan of the type shown in Fig. 11, using an airfoil balance, with the results shown in Figs. 16 and 17. The conclusion is that this effect is negligible so far as practical designing is concerned.

No. 1706

MECHANICAL LIFTS, PAST AND PRESENT, AND A NEW METHOD FOR THEIR BALANCING

By Lieut. J. F. Robbins, U.S.N.R., Washington, D. C.

Associate-Member of the Society

In this paper the author first describes at some length the various types of mechanical and hydraulic lifts that have been developed for use in canal locks. He then gives particulars regarding a new type of lift in which the points of support of two counterbalancing loads are so interconnected that the movements of the supporting elements of the structure are synchronized and it is made impossible for the supports to get out of level. The advantage of this scheme and its adaptation to freight-car lifts, lifts for launching and dry-docking and lift bridges, as well as to canal-lock lifts, are also brought out.

MANY types of mechanical lifts or elevators for lifting vessels over elevations have been proposed and built in the course of the development of waterways. However, the original principle involved in the masonry lock invented by Leonardo da Vinci still holds its superiority, mainly because efforts to develop mechanical lifts have failed to keep pace with the increasing size of vessels to be transported.

2 The inclined railway for hauling the load up and over elevations by cable served its purpose, to a certain extent, in the early days of canal development, but had its apparent limitations.

3 A gated tank on wheels was built in 1874 on the Chesapeake and Ohio Canal, to be hauled up an incline by means of cables with counterweights, but failed in its practical application.

4 In the same year, 1874, Edwin Clark, an English engineer, invented and built the first balanced hydraulic-lift lock. This structure, built at Anderton, England, provided means for transferring vessels between levels of the canal on single-plunger lifts.

5 Clark and his associates later built balanced lifts at La Louvière, Belgium, and Les Fontinettes, France, having a lift of 50 ft.,

---

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

the lock chambers being 140 ft. long and 19 ft. wide, with a navigable depth of 7 ft. 10 in. He also proposed to use this principle for transferring trains of freight cars between different levels, but did not put the idea into practical use.

CANAL-LIFT DEVELOPMENT IN GERMANY

6 In Germany much work has been done in developing mechanical-lift locks. Hoffman, a German engineer, invented and built a floating lock supported on tanks; and later, in the Dortmund and Ems Canal, locks were built on this principle having a lift 68.5 ft. The lock chamber was built in the shape of a box with end gates, 229.6 ft. long, 28.2 ft. wide and 8.2 ft. deep, supported on five steel cylinders or tanks. The weight of the structure and water load was supported by the buoyancy of these tanks which floated in wells 30.17 ft. in diameter. The lift is made in approximately 15 min. by four large screws operated by a 150-hp. motor. This installation, while larger than any mechanical lift previously built, fails to take advantage of the counterbalancing effect that obtains in the balanced lifts built by Clark. This necessitates a considerable expenditure of energy in overcoming the attraction of gravity, which is, of course, unnecessary in any form of balanced lifts.

7 In Austria an international competition for a canal lift was authorized for the Danube-Oder Canal. The difference in elevation to be overcome was 62.5 ft. and the lock was to be 229.6 ft. long, 28.2 ft. wide and 8.2 ft. deep. The competition, which closed in 1904, awarded the first prize of 100,000 kronen to a design involving the old principle of the inclined plane. The design awarded the second prize was unique, at least, in proposing a revolving elevator. The scheme called for a floating structure about 175 ft. in diameter and 230 ft. long, with a pair of swinging boxes somewhat similar to a Ferris wheel. With the elevator built lengthwise in the canal, it was proposed to float a boat into the upper or lower box through end gates, and by rotating the wheel transfer the vessel from one level to another.

8 The firm of Hoppe in Berlin has made a study of the application of a multiple of hydraulic plungers for operating large lifts, but found an insurmountable difficulty in maintaining the perfectly uniform and synchronous movement of a number of plungers which is absolutely necessary when handling loads greater than can be carried on one plunger.

TRENT CANAL LIFT LOCKS

9 The largest and most successful example of the application of the balanced hydraulic lift was built in 1905 in Canada in the Trent Canal at Peterborough.[1] This lock, shown in Fig. 1, together with the similar one at Kirkfield on the Trent Canal, has been in successful operation since put into commission and has various economic advantages over the masonry type of lock. The total lift of 65 ft. — over twice the height of the lift of the Panama Canal locks — has been made in the record time of 6½ min., the average time necessary to pass a vessel being from 10 to 12 min. Since only

FIG. 1 PETERBOROUGH HYDRAULIC-LIFT LOCK, TRENT CANAL

a comparatively small amount of water is used in making a lockage, most of the normal flow through the canal is available for water power, some of which is used for operating auxiliary pumps and lighting.

10 The lock at Peterborough consists of two steel boxes 140 ft. long, 33 ft. wide and carrying 10 ft. of water, each supported on its centrally located plunger 7½ ft. in diameter with a 65-ft. stroke. See Fig. 2. The cylinders or presses into which the plungers extend are connected by a 12-in. pipe with a gate valve, so that with one

[1] This lift has been fully described and illustrated in the *Scientific American*, July 7, 1906, the *Engineering Record*, March 30, 1907, and various other engineering publications.

box with its water load at the upper level and the other at the lower level of the canal, by opening the valve in the cross-connection the position of the lock chambers is changed and the lockage made.

11 The stroke of the plungers has been set so that the box at the upper level stops with the water level in the box about four inches

FIG. 2 ONE OF THE PLUNGERS OF THE PETERBOROUGH LOCK

below the water level in the canal, and takes on this additional water load. This excess load, amounting to about 50 tons, when taken into the upper lock acts as a surcharge to overcome friction and gravity and bring the upper lock down and the lower one up.

12 When it is desired to transfer a boat from one level to the

other, the boxes being in their respective positions at the end of the stroke of the plunger, the clearance between the end of the box and canal is first closed by inflating an air hose laid down the side walls and across the sill. Then this clearance space is filled through wicket gates in the end gates of the lock and canal. The clearance space having been filled and water levels in canal and lock having been equalized, the end gates, which are hinged across the sill of lock and canal, are folded down, making a continuous stretch of water from the canal to lock. Then the vessel is moved into the lock, displacing its own weight of water, of course, so that the load is constant, regardless of the size of the vessel. After the entrance of a vessel into one or both of the locks the end gates are closed, the clearance space emptied, and the cross-connection valve gradually opened, allowing the upper box to come down and forcing the lower one up.

13 While this installation of balanced lifts has proved to be economical in first cost, operation, time and quantity of water required to make a lockage, the locks are not large enough to accommodate vessels or cargo barges of very great capacity.

14 It was found, however, that the size and load to be handled were about as extensive as could be supported on the cantilever structure of the box over one plunger. Also, the total weight of one lock chamber, plunger and water load, which is in the neighborhood of 1900 tons, was about the maximum that could be safely supported on the masonry foundation for the cylinder castings.

15 For these reasons, and because no scheme has been found for safely supporting the load on more than one plunger, no balanced lift has ever been built of greater size than the one at Peterborough.

PROPOSED LIFT LOCKS FOR N. Y. STATE BARGE CANAL

16 A vertical-lift lock was proposed for the Erie Canal at Lockport, N. Y., where a single mechanical lift was to take the place of a double flight of masonry locks there used to overcome an abrupt change in elevation of 56 ft. The design called for a steel box 225 ft. long, 29 ft. wide and 9 ft. deep, which was to be supported by 88 link-and-pin chain cables attached to floor beams and running up and over sheave pulleys and down to 1000 tons of cast-iron counterweights. The sheave pulleys were to be carried by steel shafts supported, one on each side, in the permanent structure built at

each side of the canal. The movement of the load was to be controlled by a number of brakes on the shafts.

17 At Cohoes, on the New York State Barge Canal, the installation of a mechanical-lift lock was seriously considered, and a board of engineers investigated and reported on three different designs. Here, in the place of 16 masonry locks, it was proposed to install a pair of balanced lifts to overcome an elevation of about 120 ft. The specifications called for two counterbalancing tanks 310 ft. long, 28 ft. wide and 12 ft. deep, each capable of floating two vessels

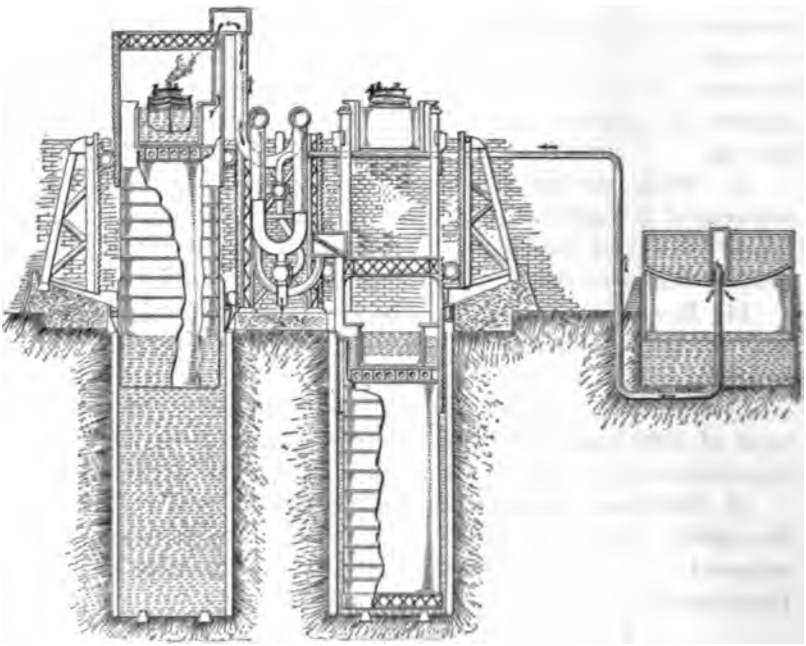


FIG. 3 AIR-SUPPORTED LIFT LOCK PROPOSED FOR N.Y. STATE BARGE CANAL

of 1000 tons capacity. The three designs considered proposed to make use of three different supporting mediums: air, sets of cables, and hydraulically operated plungers.

18 *Air-Supported Lifts.* In the pneumatic design the weight of the steel box, or lock and water load, was to be directly supported on the elastic cushion of air maintained under the load by having the steel sides of the box extend downward below the surface of water inside a large rectangular caisson built in the canal. See Fig. 3.

19 The two similar structures, either in tandem or parallel,

were to have the air space under each box connected by huge air mains 21 ft. in diameter, with the necessary return bends and valves for shifting the supporting air from the space below one lock to the other when the locks were to be shifted between levels.

20 This scheme would necessitate an excavation or pit somewhat larger than the area of the lock and somewhat deeper below the lower level of the canal than the height of the lift, or 120 ft. Built into the retaining walls of this pit were to be steel side walls to act as guides for the moving structure and to carry apparatus for maintaining the lock on an even keel.

21 A feature necessary to the successful operation of such a structure is some method of counteracting the action of excess loads at one end brought about by the banking up of water due to the wind or the entrance of a vessel.

22 The method proposed for preventing any such tipping action was to install a shaft running the full length of the lock on each side between the box and the side walls, the shafts to be equipped with gears which were to mesh with stationary racks attached to the lock structure and the side walls. Since the ability of such a device to overcome the tipping forces would depend on the torsional strength of a shaft some 310 ft. long, it is improbable that it could perform its functions successfully without an excessively large shaft being used.

23 *Cable-Supported Lifts.* The second design considered by the board of engineers covered a pair of steel boxes of the specified dimensions supported by numerous cables running over sheave pulleys on the permanent structure from one box to the other, each load thus counterweighting the other and being shifted from one elevation to another by a surcharge of water in the upper box.

24 *Plunger-Supported Lifts.* The third proposal covered a pair of steel boxes working up and down in balance with each other and supported on three steel plungers under each load. The cylinders were to be connected by piping with suitable valves so that when the upper lock came down the lower lock would be forced up, registering with the upper level of the canal. As in the pneumatic scheme, it was recognized that some device for maintaining the level of the box would be necessary. In this case the movement of the three plungers was to be coördinated by a central counterweight directly connected with both ends of the tank "in such a manner that the weight would always act to overcome the effect of any unbalanced load."

[illegible]

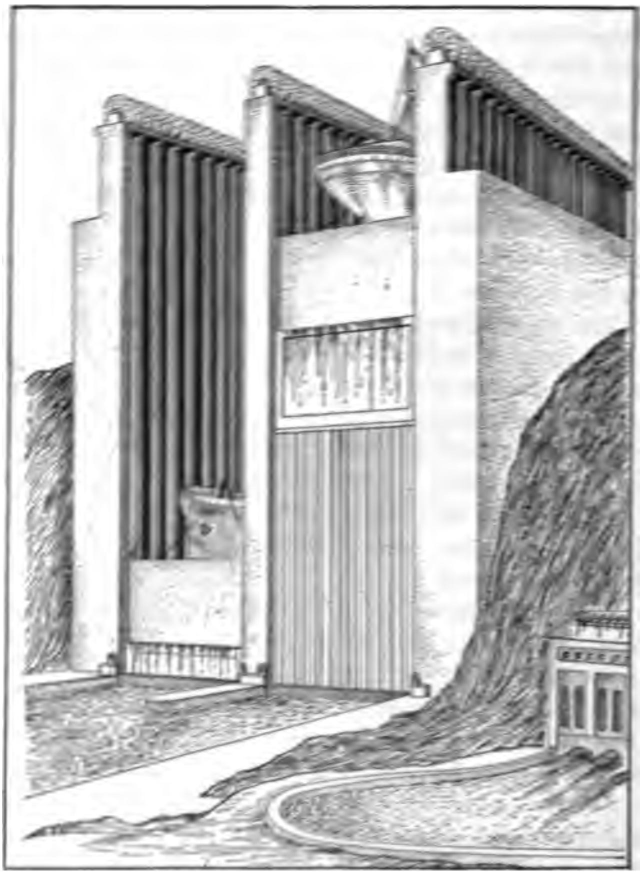
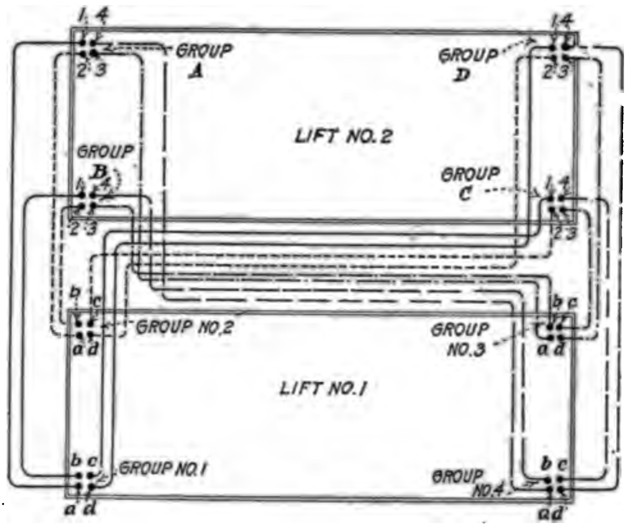


FIG. 1. CABLE-SUPPORTED LIFT LOCK PROPOSED FOR LAKE ERIE AND ONTARIO SANITARY CANAL

able for maintaining the level of an unstable water load. The report of the board of engineers shows considerable interest in the future possibilities of the hydraulic-plunger lift, but recommended four masonry locks with a lift of 28 ft. each.

PROPOSED CABLE-SUPPORTED-LIFT LOCKS

26 An interesting design for a balanced-lift lock has been described by Dr. J. A. L. Waddell,[1] in connection with the proposed Lake Erie and Ontario Sanitary Canal and power project. See Fig. 4. Here, in order to make available for power purposes the water which would be used for lockage in the masonry type of lock, it is proposed to install two pairs of balanced lifts, one of 208 ft. and



Lift No. 1: Plungers of Groups Nos. (1, 2, 3, 4) connected to plungers Nos. (1, 2, 3, 4) of Groups (A, B, C, D) of Lift No. 2., and conversely from Lift No. 2. Each plunger of Groups A, B, C, D is connected to a similar plunger in Groups (1, 2, 3, 4) of Lift No. 1.

FIG. 5 DIAGRAM SHOWING CONNECTIONS FOR HARRIS BALANCED LIFTS

(Plungers of each group of one lift are hydraulically connected to the four corners of the other lift, so that the pressure due to one load is uniformly distributed to the other, making it impossible for either lift to get out of level.)

one of 104 ft. lift. The lifts, as described, are to be supported by cables connecting the inboard sides of the two parallel boxes and running over fifty-six 20-ft.-diameter sheave pulleys mounted on the retaining wall between the boxes. The outboard edge of each box is connected by a similar set of cables running over pulleys and to counterweights suspended outside the two outer retaining walls.

27 This design, adopted as the most feasible *known* method of overcoming the high lift with locks of large size, is somewhat similar

[1] *Scientific American,* March 23, 1918.

since the red, brown and twisted cords are carried in the same way from the other three corners of the first shelf to all corners of shelf No. 2, no part of either shelf can move up or down without all parts of the other shelf moving an equal distance in the opposite direction. Any part of either shelf will sustain loads equal to the full combined strength of the cords supporting that part.

35 The device is not limited to any one size or shape in securing these results. If the shelf is too long to carry the necessary load by the cords supporting it at the corners, a second group of cords can be attached to each shelf at any equal distance from the center of each shelf, carrying from each part of shelf No. 1 to each corresponding part of shelf No. 2 and duplicating the arrangement of the first set; these shelves can then be loaded to the combined strength of the 32 cords and at the same time be practically divided into three sections as to their strength for carrying these loads. Then, if force enough to overcome the combined friction of all the cords be applied to either shelf, all parts of each one will travel up or down the same distance at the same time.

36 It is apparent that with the two loads supported in this manner, either by plungers below the lifts, or overhead cables, no load, up to the limit of the design of the structure, could force either lift out of its normal plane or level, however eccentrically the load might be placed. If it were desired to build a pair of lifts several hundred feet long, the total length would be divided into a certain number of spans required by economy of design, and the required number of similar systems of balancing plungers would be used and the various systems connected as shown in the sketch of a single system. Thus, by increasing the number of similar systems it would be possible to build lifts of practically unlimited size.

FREIGHT LIFTS

37 The utilization of the potential energy of one of two similar loads to shift them between different levels has only been applied in a large way to the transfer of vessels in canals. But the foregoing illustrated scheme of supporting two such loads makes it feasible to apply the principle to other uses. Clark's idea of transferring trains between different levels on single-plunger lifts may now be applied to the handling of freight between subway and surface levels, on lifts of sufficient size to carry any number of freight cars.

38 In view of the difficulty and expense of present methods of delivering freight, for example, into New York City by car ferry and the fact that no feasible method has hitherto been proposed for getting freight from subways to surface levels other than the hauling of trains up inclines or breaking freight below and lifting it on elevators of small capacity, this scheme is put forward as a solution to the problem of overcoming freight congestion.

39 With the increasing value of real estate, the use of large surface areas becomes prohibitive, making many floors above and below surface levels necessary. This scheme, applying balanced lifts to freight-terminal warehouses and points of distribution, would obviate present long hauls both by truck and train, and make it possible to utilize space to the greatest advantage. For the handling of freight it would be advisable to design the lifts so that a certain load could be lifted up in excess of that coming down. This could be done by having a certain number of lifting plungers under each lift, with a pumping plant of the necessary capacity to lift the excess load.

40 Fig. 7 shows a typical design of a freight lift for handling fifteen cars on each of two three-track bridge structures between subway and surface levels. After assuring the level and uniform travel of the lift by a certain number of balancing systems of plungers, certain plungers of the remaining groups have their pipe connections led through a pumping plant for controlling and lifting excess loads.

41 While it is true that in less-than-carload-lot freight terminals the outgoing tonnage is usually two or three times the incoming tonnage, this capacity for lifting a certain per cent excess load would be advisable and methods of handling such incoming and outgoing freight would have to be so coördinated as to make use of the loads in so far as possible. The available portion of the potential energy of the excess outgoing tonnage might be used to advantage to lift occasional excess incoming loads by storing the excess energy in accumulators and using it to assist in elevating incoming tonnage.

LIFTS FOR LAUNCHING AND DRY-DOCKING

42 This principle may have a possible application in the field of shipbuilding in connection with the dry-docking and launching of vessels. For example, it might have been applied to advantage in the launching elevator recently built at the Ford plant for launch-

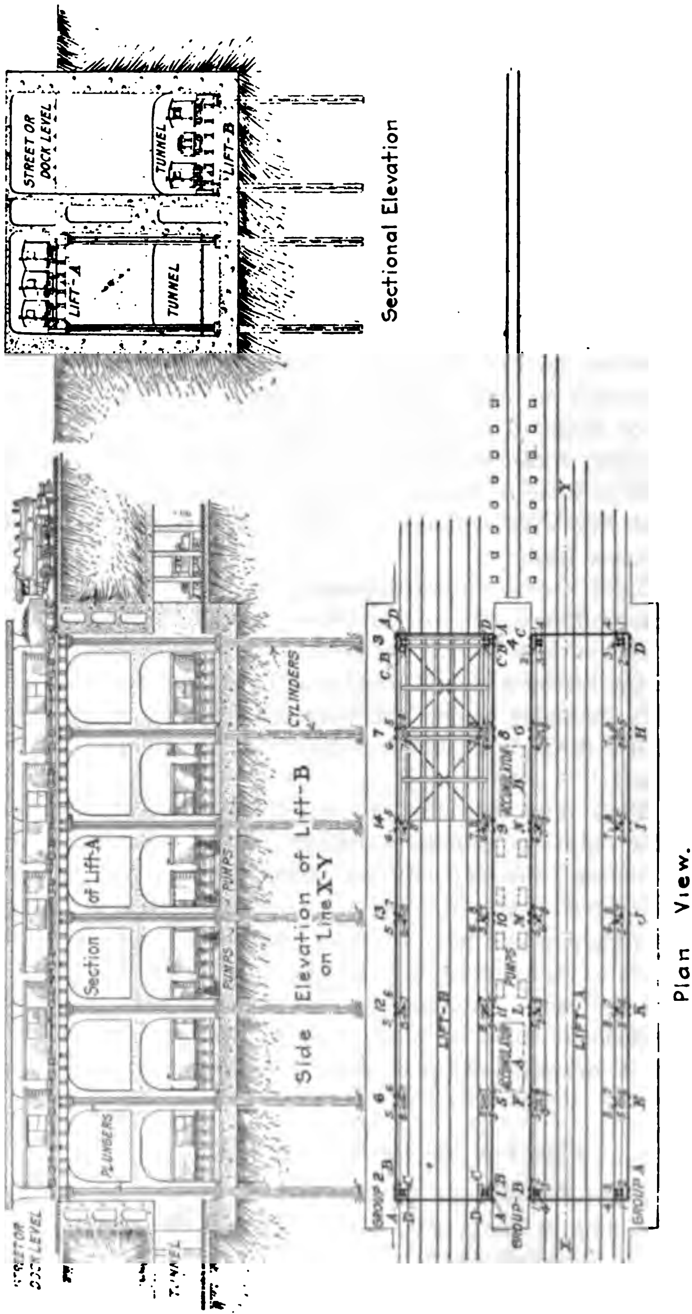


Fig. 7 Typical Design of a Hydraulic Balanced Freight-Car Lift

ing Eagle boats. In this installation two pairs of simple hydraulic jacks on each side of the platform are used to support the movable structure and boat and to lower the boat into the water. It is then necessary to pump the jack plungers and platform back up into position.

43 An elevator of this kind might be supported by groups of four plungers under each corner with a nearby counterweight to balance the weight of the platform, supported on an equal number of plungers. With the plungers connected as described there would be no possibility of their tipping or binding, even though the platform were eccentrically loaded. The counterweight would bring the platform back up into position, without the necessity of a pumping plant or any power for control or operation.

LIFT BRIDGES

44 This scheme may be applied to lift bridges where the span is too great for the use of the so-called "jack-knife" bridge. Here again the bridge structure could be supported by four plungers at each corner connected to four plungers under each corner of the counterweight situated under the roadway at one end of the bridge. The plungers under the counterweight might be two or three times the diameter of the plungers supporting the bridge, and so reduce the stroke of the counterweight to one-half or one-third that of the bridge.

45 The same principle of interconnecting the points of support of bridge and counterweight may be applied with supporting cables instead of plungers. It is probable, however, that the plunger lift would be the most economical design as no heavy overhead truss would be necessary and the towers now used in counterweighted vertical-lift bridges to support the overhead truss, counterweights and motors could be made very much lighter, since these towers would only be needed as guides at each end of the bridge. The movement of the bridge would be properly regulated by a train of throttling valves in the system of interconnecting pipe lines, and all operations would be governed by an interlocking system of automatic control.

BALANCED LIFTS AS APPLIED TO ERIE AND ONTARIO CANAL PROJECT

46 While the lift of 208 ft. in the proposed lock of the Erie and Ontario Canal project is considerably higher than any existing

plunger lift, it is within the range of the feasible application of a multiple of supporting plungers. Such lifts as are proposed, supported on plungers, would be absolutely positive in their relative movement and would have their equilibrium and level assured without auxiliary apparatus. Plunger lifts would render unnecessary the immense counterweights, the total weight of which has to be equivalent to the weight of one lock with its water load. By supporting the loads on plungers below the structure, the retaining walls, which support the sheaves and total weight of the locks, could be made very much lighter as they would be needed simply to act as guides to the movable structure. With these locks supported on the proper number of systems of balancing plungers, as

Fig. 8 View Underneath Model No. 3 of Harris Hydraulic Balanced Lift, Showing Plungers, Piping Connections, etc.

determined by economical design, the groups of four plungers at several points in their travel upward would pick up guides to act as stiffeners similar to those used with the long single plungers of passenger elevators. These guides would hang by chain or cable from the lock structure and would be provided with a loose collar for each plunger and guide in tracks in the side walls, thus breaking up the unsupported length of the plunger columns sufficiently to carry the load without bending.

47 In the cable-supported lifts described by Dr. Waddell it is proposed to use electric motors connected to the sheave-pulley shafts for shifting the locks. Since the friction of the extensive cable system and the inertia of two such loads, estimated to be

approximately 50,000 tons each, would be very great, this would entail a considerable expenditure of power. In view of this fact it would seem more economical to make use of a surcharge of water in the upper lock for shifting the locks, similar to the manner in which the Peterborough locks are operated. With the two loads supported on systems of balancing plungers, assuring perfect synchronism of movement, control in starting and stopping would be

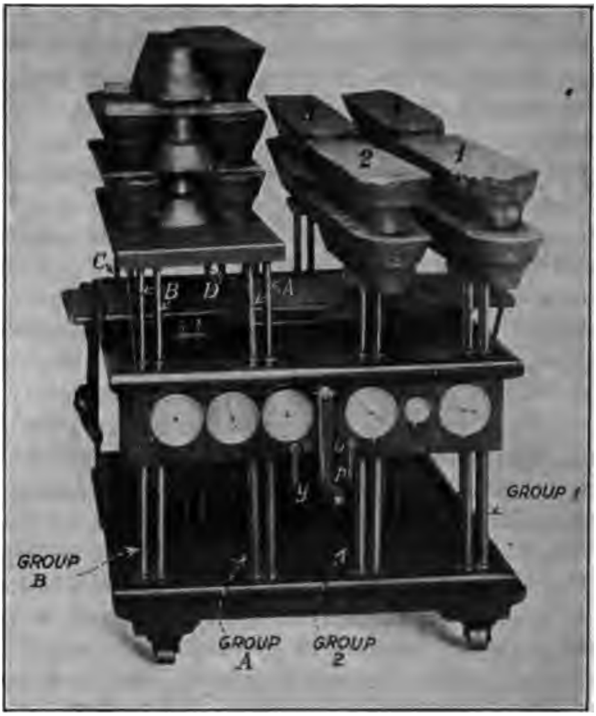


FIG. 9 MODEL NO. 3 WITH EACH GROUP OF PLUNGERS OF ONE LIFT HAVING INDIVIDUAL LOADS AND AN EQUAL LOAD ON THE PLATFORM OF THE OTHER LIFT

maintained by a train of simultaneously operated throttling valves in the various interconnecting pipe lines.

48 The proper sequence of events in the operation of such an installation, as a whole, would be governed by semi-automatic interlocking control systems under the supervision of one man, so there would be little possibility of anything going wrong.

49 Figs. 8 and 9 illustrate the hydraulic-plunger application of this principle. Fig. 8, a view of the underneath side of Model No. 3,

shows the four pipes running from the four balancing plungers at each corner of one platform to the four corners of the other platform. Under the middle of each platform are two groups of four pump plungers connected to a common pipe through a small geared pump to the other platform. At the left end of the model is a small accumulator, with a hand pump, which can feed through a common pipe and check valve in each of the 16 small pipe connections between the balancing plungers. The accumulator is kept under sufficient pressure to feed through any check valve to replace leakage in any part of the system. Another train of check and throttling valves — one in each of the 16 interconnecting pipe lines — connects to a common pipe and back to the accumulator for throttling down the platform that happens to be in the upper position when it is desired to put the lift out of commission for repairs.

50 In another view of Model No. 3 (Fig. 9) the pump plungers were secured down out of the way and a load of two pigs of lead was balanced over the cap of each set of balancing plungers. An equal load was placed on the other platform. With the model loaded in this way it was possible to force down any one of the four groups of plungers by the pressure of the hand, when the other platform would rise and the remaining three groups of plungers would descend at the same rate; showing that no dependence was necessary on the platform structure for maintaining the level of the groups and that no strains would be produced in such a structure for maintaining the loads level. Moreover, when loads of 1200 lb. were placed eccentrically on each platform the lifts remained level when moved up or down by means of the pump and pump plungers.

51 The gages on the front of the model were installed for the purpose of studying the various pressures. The first one on the right shows the accumulator pressure; the small gage indicates the pump pressures, and the remaining four gages are installed in the pipe lines from the cylinders of one group of balancing plungers.

52 In its application to canal-lock lifts this principle has no limit to which it may be extended. It may be applied in the great canal systems being developed in Canada, and in the possible future canalization of the United States for ocean-going shipping. It has the very great advantage of economy of water necessary to supply the lockage of vessels. This feature makes it possible to build canals over territory where the rainfall over areas at summit levels is insufficient to supply the water necessary for lockage in the old masonry type of lock. This matter of taking the water supply

now used for water power for use in proposed canal systems has alone been a considerable item of cost of proposed waterways.

53 While this scheme of interconnecting points of support of two loads, as described, has been granted basic patent rights and some little work has been done in the design and development of models and in anticipating the many engineering problems involved, no application of it has yet been undertaken.

54 It is believed, however, that by the development of this method of so connecting the points of support of two counterbalancing loads as to synchronize the movement of the supporting elements of the structure and make it impossible for the loads to get out of level, its inventor, William Thomas Harris, of Chicago, has solved an important mechanical problem, and has opened a way to the future development of waterways and to more efficient methods of handling freight.

DISCUSSION

Thos. H. Rees[1] (written). I happened to be located in Chicago at the time when Mr. Harris was developing the principle and design of his balanced lift and had the pleasure of several discussions with him on various features of his invention. I was greatly impressed with the ingenuity of his device and with the possibilities of its practical application.

The older types of lift are so well known that they require but little comment. In all of them that involve large dimensions difficulties are encountered in preserving a level position under unsymmetrical loads. Mr. Harris's simple application of well-known principles solves these difficulties and preserves the horizontality of the loaded platforms under any distribution of the loads.

In the simple system of four groups of plungers to each lift as illustrated in Fig. 5 of the paper the cross-connections between cylinders are readily traced $A1$ to $1A$, $C4$ to $4C$, etc., but in the multiple system shown in Fig. 7 they are not so readily apparent. The system of notation given below will show plainly the interconnections between groups and cylinders.

[1] Colonel, Corps of Engineers, U.S.A. Office of Division Engineer, Southeast Division, Savannah, Ga.

1		2	5		6	9		10	9		10	5		6	1		2
	A			*E*			*J*			*K*			*F*			*B*	
4		3	8		7	12		11	12		11	8		7	4		3

LIFT *B*

1		2	5		6	9		10	9		10	5		6	1		2
	D			*H*			*M*			*L*			*G*			*C*	
4		3	8		7	12		11	12		11	8		7	4		3

A		*B*	*E*		*F*	*J*		*K*	*J*		*K*	*E*		*F*	*A*		*B*
	1			5			9			10			6			2	
D		*C*	*H*		*G*	*M*		*L*	*M*		*L*	*H*		*G*	*D*		*C*

LIFT *A*

A		*B*	*E*		*F*	*J*		*K*	*J*		*K*	*E*		*F*	*A*		*B*
	4			8			12			11			7			3	
D		*C*	*H*		*G*	*M*		*L*	*M*		*L*	*H*		*G*	*D*		*C*

In Lift *B*, the groups are lettered and the plungers are numbered. In Lift *A* the groups are numbered and the plungers are lettered. The connecting pipes or cords are from *A*1 to 1*A*, *J*10 to 10*J*, *M*11 to 11*M*, etc.

In Fig. 7 there is an odd pair of groups at the middle of each lift which are assumed to be the operating plungers with pump connections and which might be connected as follows:

13		13
	N	
14		14

13		13
	O	
14		14

N		*N*
	13	
O		*O*

N		*N*
	14	
O		*O*

In the balance lift for freight cars, instead of using operating plungers with power pumps, it would be feasible to build tanks in the platforms and by means of service tanks, one high and one low, to admit water augmenting the lighter load and draw off water reducing the heavier load and thus secure the necessary overbalance to produce motion in the desired direction. A centrifugal pump would restore the used water to the higher service tank, the total work being the same as with pump-driven plungers.

Referring to Par. 43, under the heading of "Lifts for launching and dry-docking," it would be advisable to use a tank of water as a counterweight and to draw off water when the platform and vessel become submerged in order to compensate for their buoyancy and permit the platform to sink clear of the floating vessel. In other cases where varying loads are to be handled adjustable water ballast will be found useful.

It is true (Par. 50) that with a simple system of 16 plungers no strains would be produced in the superstructure for maintaining the level of the loads, but with a multiple system each set of 2 times 16 plungers acts as a separate unit and any force applied through one set of plungers to produce motion must be transmitted through the superstructure to overcome the friction in the plungers of other systems. Such transmitted forces will, however, be balanced and will have no tendency to disturb the levels.

It will be most interesting to observe the first practical application of this balanced multiple plunger system on a large scale, and it is to be hoped that such application will not be long delayed.

A. W. Moseley (written). The practicability of balanced locks seems to have been demonstrated. No balanced locks of great size have ever been built, however, and difficulties of construction and operation which have never been met will surely require attention. Among the most prominent difficulties are:

1. The keeping of each lock tank in perfect level.
2. With chain supports,
   a large number required,
   b the mechanism necessary to insure reasonably equal load on each cable,
   c the mechanism to control the motion of the lock tanks, and
   d the unbalancing effect of the weight of the chains.
3. With hydraulic plungers,
   a with reasonable pressures, the enormous amount of plunger area

b the complicated piping necessary to secure the trim of the lock tanks,
c the complexity of the valve control,
d the side support of the plungers in case of high lifts, and
e the unbalancing effect of the weight of water displaced by the plungers.

Lieutenant Robbins seems to be strongly in favor of the use of plungers cross-connected on the Harris plan. In view of the difficulties itemized above, it would seem that some method might be devised to simplify and cheapen the entire construction. His objection to the use of cables appears to be chiefly due to the difficulty of control of operation, and this objection seems to be well taken.

A careful reading of the paper gives no hint that serious consideration has ever been given to a combination of the hydraulic and cable methods of operation. This combination is successfully used in very many elevator installations; not, however, so far as I know, for elevators balanced in pairs. It would seem that such a combination of methods would lend itself admirably to the problem under discussion in a manner somewhat as follows:

The full weight of the lock tanks would be balanced against each other by means of cables, such cables being arranged according to the Harris plan. It may be noted in passing that the unbalancing effect noted in 2-*d* and that noted in 3-*e* would tend to offset each other.

Plunger-lifting capacity amounting to perhaps a fourth or a fifth of the weight of the lock tanks and their contents would also be provided, and this plunger installation would serve to operate and control.

The subject of the paper is fascinating and one destined to take a very important place in the field of mechanical engineering, in view of the growing prominence of traffic by canal in all parts of the world. A general survey such as is given in the paper under discussion is most timely.

F. H. Frankland [1] (written). The author makes the following statement in referring to the proposed mammoth cable-supported lift locks for the Erie-Ontario Canal: "It is believed that there would be considerable difficulty in maintaining the equilibrium of these lock chambers." He does not, however, state upon what

[1] Consulting Engineer, New York City.

grounds his belief is based. In any lift lock safety demands that some mechanical device be applied which will maintain the level position of the lock chambers. In the cable-supported lift lock this can be provided for in either one of two ways; first, by placing all the supporting sheaves on continuous shaft, or, second, by a vertical screw shaft at each corner operated by an electric motor.

Maintenance of equilibrium of cable-supported lock chambers can be had to the greatest nicety, but the maintenance of equilibrium when hydraulic plungers are used is of the greatest difficulty, because it involves the maintaining of constant volumes of liquid under each plunger, especially with the author's design. This great difficulty is due to the slight compressibility of water, to the unavoidable leakage under the high pressures required, to the expansibility of the pipes, and especially to the air content of water, which it is impracticable to remove entirely. The cable-supported lift lock does not present a difficult problem in maintaining the horizontal position of the chambers, be they fifty feet long or one mile long. Can the author explain how it is possible that "any excess load on the one end of one lock, tending to force it down, would lift the end of the other up and would be cumulative in effect with the water load"? It is granted, of course, that the above contention might be correct if it were possible for any excess live load to exist at the end of a lock tank. The amount of water admitted to each lock chamber can be controlled to any degree of exactitude, and, naturally, the tank is in equilibrium at all times, owing to the fact that the live load is constant and always uniformly distributed. It should be needless to point out that the load and its reaction are always constant, no matter whether there is a vessel in any position in one, both, or neither of the locks.

The author says, "Plunger lifts would render unnecessary the immense counterweights" (as applying to the cable-supported locks). Evidently he has not considered the necessity for equally great or greater counterweights which would be required on the hydraulic accumulators of plunger lift locks. In any case it is almost a self-evident fact that the plunger lift lock would be much more expensive than the comparatively simple cable-supported locks.

It would appear to the writer that Mr. Harris's proposed scheme, involving, as it does, an exceedingly intricate plunger, pipe and valve system, would insure constant trouble through difficulties of functioning and maintenance.

Under the caption of "Lift Bridges" the author says, "It is probable, however, that the plunger lift would be the most economical design, as no heavy overhead truss would be necessary." The writer, being a specialist in the design of vertical-lift bridges, can confidently assure Lieutenant Robbins that the hydraulic plunger lift bridge is *not* nearly as economical as regards first cost, operation, or maintenance, as cable-supported lifts. The writer wishes to dispel the illusion created in the paper as regards the "heavy overhead truss." There is no heavy, nor indeed any, overhead truss required for cable-supported lift bridges. Consequently the author's contentions regarding lift bridges fall to the ground. The writer wishes to point out here the very serious difficulties presented in hydraulically-operated lift bridges by the necessity of carrying the pressure pipes under the channel.

THOMAS E. BROWN[1] (written). Lieutenant Robbins' paper is interesting and valuable, especially in its complete, though brief, synopsis of the art as applied to large structures up to the present time. The writer is especially interested in view of the reference in Par. 24 and 25 to the three-plunger lift design offered by him for the proposed lift lock at Cohoes. This plan was approved to the extent that, while reporting in favor of another route, the board of engineers recommended the plan in the event that the Cohoes route be adopted.

In Par. 25 the author implies that "mechanical devices" are not suitable for the stabilization of variably loaded structures, and apparently criticises the writer's design in which the ends of the tanks were connected to a central stabilizing counterweight. The writer fails to appreciate why the term "mechanical" applies to this construction in any different sense than to the cordage system described in Par. 31, or to the accumulator and plunger systems elsewhere described. If a distinction is to be made between metallic and hydraulic devices, then it may be pointed out that hydraulic pressure must be controlled and confined by metallic devices which certainly are mechanical.

Broadly speaking, any stabilizing device must be mechanical.

The principle described in the paper is very old, but as described involves an excessive number of cords or plungers. The broad principle may be stated as follows: *If a body is connected by flexible members of constant length, from at least three supporting corners to a single point, then when said point moves vertically the body will move vertically and remain parallel to its primary position.*

The principle is the same whether the connections are pipes

[1] 35 Nassau St., New York, N.Y.

full of water or cords overhead, but constant length is essential. The single point may be on a counterweight or may be on a similar body, as one of the platforms or tanks under discussion. When two bodies are to be kept parallel, at least three corners on each may be connected to a single point on the other. The principle seems too obvious to be patentable, nevertheless the writer was granted a patent involving this principle eighteen years ago.

In view of the simplicity of the principle the great multiplication of plungers or cords proposed in the paper seems entirely unnecessary. Experience shows that with long narrow structures, supported at their longitudinal center line, transverse moments are easily sustained by the fixed guiding structure, and, therefore, devices are needed only for longitudinal stabilization; in such cases only two points, the ends, need be connected. It would seem, therefore, that the very large number of elements proposed in the paper is entirely unnecessary.

A great trouble in hydraulic apparatus is leakage, which may occur from a variety of causes, at any moment, and it is believed that maintenance of the constant quantity of liquid required in each separate element of the proposed system will be found impracticable.

Many years ago the writer prepared a design of a plunger train lift for the Reading Railroad, on the multiple ram accumulator principle suggested in this paper. After very careful consideration, the engineers decided against it, for the reason mentioned, and the writer withdrew the design and substituted another which met the objections. The latter design involved several plungers taking pressure from a single source, one plunger acting as a governor or pilot for all the others.

The design proposed by the writer for the Cohoes lock consisted of two tanks separated by a central guiding structure sufficient to sustain all overturning and transverse moments; each tank was supported on three plungers, the natural strength of the tank, *i.e.*, that due to the minimum metal practicable, requiring but three points of support.

All three plungers were connected and under equal pressure. Stabilization was accomplished by cords (a group of very strong cables) from each end of the tank to a counterweight, each tank having a separate counterweight. Each counterweight was sufficiently heavy to provide for the maximum possible variation of load, which was assumed to be that due to such an angle of the water surface as would cause a spill over one end of the tank.

The stabilizing counterweights were considered necessary in view of the recognized impracticability of maintaining a constant volume of liquid under the plungers, and especially so in the event of leakage. Had the cords connected one tank with the other, then a decrease of volume of liquid would throw the entire weight of both tanks on the cords and guiding structure, which for safety would have to have been constructed to carry this load. The use of stabilizing weights removes this difficulty.

The Cohoes design with only six plungers and a simple system of piping and valves appears to the writer far superior to the arrangement suggested in the paper requiring some 48 plungers and an equivalent number of pipes and valves.

There have been many installations on a smaller scale of multiple plungers with levelling devices. About thirty-five years ago Mr. R. C. Smith, then engineer of the Otis Elevator Company, designed a gate for the Mills Building, New York, with a span of 40 feet or more, supported by a plunger at each end. These plungers were controlled by a valve operated by the gate itself, to maintain the level.

The writer, some years later, designed and installed a double plunger passenger elevator in the Saint Paul Building, New York City, and later a double plunger elevator in the Prudential Building, Newark, N. J.

A very elaborate system of multiple plungers was designed and installed by Mr. D. L. Holbrook, in the Hippodrome, New York, to lift the stage and the aquatic tank. These installations operated successfully for many years and probably are still working.

At the end of Par. 46 the author describes a device for supporting long plungers at intermediate points. This device is an invention of the writer's, and was first installed by him in the tank shop of the Pennsylvania Railroad at Altoona, Pa.

In Par. 6, describing the Hoffman lock, the author appears to be in error in assuming that the lock is unbalanced. Such a lock should be perfectly balanced under normal conditions and the power required, as in any other balanced lock, only that necessary to overcome friction. The large motive power is provided to take care of variations of level in the canal, and provision must be made for this condition in any system of lift locks.

The writer believes that Mr. Harris has merely solved, in a complicated manner, a problem already solved by many others in much simpler ways.

The author's paper is of great value in calling attention to this important subject and its possibilities, which appear to have been generally overlooked. Whatever difference of opinion of methods and details may exist, the general conclusions reached as to the advantages of such constructions in the solution of canal and railroad problems are in accord with those of the writer.

THE AUTHOR desires to thank Colonel Rees for his constructive criticism of this paper and for suggesting some details of connection and design.

Limited space prevented the author from giving a detailed description of the design for a car lift (Fig. 7). In this design, it was proposed to use the end groups of each lift, group *A, B, C, D,* and Nos. 1, 2, 3, 4, for balancing plungers interconnected as shown in Fig. 5. After providing for one complete set of balancing plungers and two balancing plungers from each of the groups *I, K, L, N,* and 9, 11, 12, 14, making 24 balancing plungers under one lift, the remaining 32 plungers have their piping connections led through the pumps to the other lift. This design provided for lifting up a load fifty per cent in excess of that coming down.

An accumulator system is incorporated in the design to store excess energy from heavier down-coming loads and to assist the pumping plant in lifting excess loads up.

The author believes with Mr. Moseley that the practicability of the balancing principle has been demonstrated and agrees that there are problems of economical design to be solved for each different application of the principle. It is believed, however, that there are no details that may not be worked out by following present accepted engineering methods.

Pressures between 1000 and 2000 lb. per sq. in. can be carried as safely as lower pressures and result in economical plunger areas. Multiples of plungers permit of economical lengths of span and reduce the weight of steel necessary in the lift structure. Plungers, cylinders, and castings are all similar and can be machined, cast and assembled in multiples in any large shop.

Using the higher pressures and small plungers, piping and valves for economy, these parts need not be complicated nor of special manufacture. The piping between balancing plungers depends for positive action on its simplicity. There must be only one pipe connecting any one plunger under one lift with a similar plunger under the other lift.

Connected with each of these separate pipe lines there would be one check valve connecting with an accumulator system for make-up feed to compensate for leakage around the plunger packings.

Mechanism for side support of long plungers has been developed and in use for many years and should present no serious difficulties.

The water displaced by plungers would not produce any unbalancing effect as all of the plungers are of the same size and the weight of water displaced by plungers that are down would be uniformly distributed over the area of the lower lift. The weight of water displaced by plungers under the lift going down necessitates a small additional per cent of pressure to force the one lift down and the other up after they pass each other at the middle of the stroke.

The greater part of the paper deals with plunger supported mechanical lifts since most of the earlier development has been along this line, but there is no reason to believe cable supported lifts may not be preferable for some applications. Either plungers, cables or combinations of the two may be found most economical for particular cases. In fact a combination of cable and plunger support has been incorporated in a design for a pair of freight elevators where conditions call for a long and narrow lifting platform. As Mr. Moseley suggests, the balancing and support of the lifts will be taken care of by the cables, while excess loads and control will be handled by plungers and pumping plant.

Mr. Frankland questions the statement that, there would be considerable difficulty in maintaining the equilibrium of lock chambers 660 ft. long, 70 ft. wide, carrying 30 ft. of water and supported on cables, as designed for the Erie-Ontario Canal. The author believes that sufficient grounds for this statement are evidenced in the fact that in every design of proposed and actual installations, all much smaller in size, this balancing problem has been recognized as the most difficult consideration. In all such designs, separate mechanical devices have been proposed and, while vertical screw shafts and motor control has been used successfully in the Dortmunder-Ems lock, such mechanism or brakes and motors on the sheave-pulley shafts are apt to be a large and expensive part of the whole installation.

Difficulties involved in the use of plungers for maintaining the equilibrium of lifts have been mentioned, such as the compressibility of water, expansibility of pipes, entrained air in the system and leakage. Water is so nearly incompressible that variations in volume due to different pressures need not be considered. Change in volume due to expansibility of pipes may be reduced to a negligible point by the use of piping of the proper strength to take care of variations of pressure due to eccentric loading. Free air can be eliminated from pure water without difficulty and by the avoidance of air pockets in the piping, no appreciable change in volume would result from entrained air. If, as Mr. Frankland believes, there could be no excess load at one end of a lock tank, then there could be no variations of pressure tending to reduce the volume of water under various plungers and his criticisms on the above points would have no weight. A small per cent of leakage around plunger packing is unavoidable, but can easily be supplied by an accumulator system. This leakage will be fairly uniform among the various plungers even when there happens to be a considerable variation in pressure due to eccentric loading as neither friction of plungers nor leakage increases in proportion to increase in pressure.

Mr. Frankland questions the possibility of there ever being an excess load on one end of a lock tank. The probability of this condition has been recognized in every design of lift locks. For example, I may quote from Mr. Brown's discussion describing the design for the Cohoes locks, "Each counterweight was sufficiently heavy to provide for the maximum possible variation of load, which was assumed to be that due to such an angle of the water surface as would cause a spill over one end of the tank."

Since in the design favored by Mr. Frankland, the cables supporting one end of one lock run to the same end of the other lock, any excess load on one end of one lock, tending to force it down, would certainly react on the same end of the other lock tending to lift it up. It is equally certain that such force, having tilted the tank ever so slightly out of level, would be cumulative in effect, as long as water runs down hill.

Attention may be called to the fact that the stretching of cables used for supporting and overcoming these unbalanced loads will present a far more serious problem than the compressibility of water or the expansibility of pipes.

The counterweights for the accumulator system of plunger locks of Mr. Harris' design have been considered sufficiently to know that they would be a comparatively small part of the whole installation. Since they are needed solely for supplying leakage around plungers, their capacity, in proportion to the size of the locks, would be as the amount of leakage is to the total amount of water in the plunger system. Such counterweights can hardly be compared with those of a design that calls for counterweights equivalent in weight to the weight of one of the lock chambers full of water, some 50,000 tons.

The author does not pretend to be an authority on lift bridges and, as there are no plunger lift bridges to compare with those of the cable supported design, cannot furnish any data as to their relative economy. It seems possible, however, that a design making it feasible to use very much lighter end towers (since they would not have to carry the weight of counterweights, sheave pulleys, cables, etc.), using the towers simply as guides for a plunger supported bridge and placing the counterweight under the roadway, might be worthy of consideration. It is not believed that there would be any serious difficulty involved in laying ten 2- or 3-in. pipes under the channel between the plungers on one side and the counter-weight on the other.

Mr. Frankland criticizes the writer's reference to overhead trusses connecting the towers of lift bridges. Recognizing the fact that later designs have made it possible to eliminate the overhead truss, still the very substantial structure connecting the towers observed many times in crossing the Halstead Street bridge in Chicago can hardly be dispelled as an illusion.

The author is indebted to Mr. Brown for bringing up several points for discussion that evidently have not been covered in sufficient detail. It appears desirable to point out the difference between purely "mechanical devices" which have been proposed for the "stabilization of variably loaded structures" and the balancing principle which is inherent in the method of support in Mr. Harris' design.

Referring to the description of the shelves (Fig. 6) both supported and maintained in balance by the same supporting members, it is apparent why a distinction is made between this principle and designs involving separate mechanical devices for stabilizing, such as counterweights underneath the lift or hung by cables at each side, longitudinal shafting and gears, vertical screw shafts at each corner operated by motors, or dynamic braking mechanism on sheave-pulley shafts. The balancing principle described by the author involves no extraneous stabilizing device, but is inherent in the method of supporting the loads, be it by cables or plungers.

It is agreed that the principle set forth in Mr. Brown's discussion is very old, i.e., "If a body is connected by flexible members of constant length, from at least three supporting corners to a single point, then when said point moves vertically the body will move vertically and remain parallel to its primary position." But the writer desires to point out that this is far from being the principle under discussion.

Mr. Brown's principle no doubt was one of the most ancient mechanical devices and was certainly made use of by our grandfathers when they hung a shelf by three or four cords running up through pulleys and down to a single point on a counterweight. Such shelves served the purpose for which they were intended, but the idea would hardly be feasible for balanced freight or lock lifts. The writer maintains that the principle he describes is new and can only be applied by using the same number of supporting members at each point of support of each lift as there are points of support to the lift, and connecting those supporting members as previously described to the other balancing lift or counterweight. Giving an example of the principle he has in mind, Mr. Brown says, "When two bodies are to be kept parallel, at least three corners on each may be connected to a single point on the other." This idea might suffice for two shelves with a uniformly distributed load. But assume two such bodies in form of shelves with three points on each shelf supported by three cords running to a single point on the other shelf. (The single point on each shelf would logically be over the center of gravity of the shelf.) If the load on each shelf were concentrated at one of the three points of support and sufficient additional force were applied to that point on one shelf to depress it, the center of the other shelf must be raised an equal distance through the medium of the single cord, but there is nothing to prevent the loaded corner of the second shelf from tipping down, since there is a couple formed by two forces acting in opposite directions through the center and one corner of the shelf.

In order to make clear the distinction between the two principles, consider the same two shelves with *three* cords supporting each of the three corners. Let the three cords at *one* corner of one shelf be connected to the *three* corners of the other shelf, instead of at one single point, and let the cords at the remaining corners be similarly connected. Then if we depress any corner or part of one shelf, we necessarily apply the lifting force at all three corners of the other shelf instead of at a single point and regardless of the location of the load on the shelves, all parts of each one must move equal distances.

As concisely as it is possible to state the proposition in one paragraph, I would put it as follows:

If each of two bodies is supported by flexible members of constant length, and if there are as many supporting members at *each point* of support as there are *points* of support to the body, and if *one* member from *each* point of support of the first body is connected to *one* point of support of the second body, and if the remaining members of the first body are similarly connected to the remaining points of support of the second body, then the vertical movement of any part of one body will cause an equal movement of the other body as a whole and no part of either body can move up or down without all parts of the other body moving an equal distance in the opposite direction.

The writer desires to correct the impression given by Mr. Brown in his reference to the Hoffman lock, where he says, "The author appears to be in error in assuming that the lock is unbalanced." No such assumption was made. There is no question about the lock being balanced in the sense of its being in stable equilibrium. This lock is a single-lift structure and cannot be termed a balanced lift as compared with the pair of lifts in balance with each other in Clark's design. The statement made by the author was that the Hoffman lift, "fails to take advantage of the counterbalancing effect that obtains in the balanced lifts built by Clark."

Reference may be made to *The Engineer*, London, of Jan. 3, 1902, and of April 17, 1896, for more detailed descriptions of this lift. It will be seen that the large motive power is only partially required for overcoming friction, for there is very little friction due to the lift floating freely on tanks in the wells, and is not required for taking care of variations of the level of the canal, but that the power is necessary for control and for lifting the lock in overcoming the effect of the decreasing buoyancy as the structure connecting the lock chamber and the tanks rises out of the water in the wells.

A description of the design of this lock before completion states that the lock was to be caused to fall or rise by the addition or removal of water from the lock. The later description after it was put in operation states that about 400 hp. is required in overcoming the inertia in starting, about 200 hp. being required after the structure is in motion.

The Cohoes locks were to be comparatively long and narrow and were admirably suited for support along their longitudinal center line, depending on side walls for guides to sustain transverse overturning moments. In Mr. Brown's design for these locks, it was necessary to include two separate stabilizing counterweights sufficiently heavy to provide for the maximum possible variation of load. If these locks had been supported on nine plungers, three at each of the points of support and interconnected, as in the design of Mr. Harris, the system would have required no stabilizing counterweights. Of course with nine plungers, the area of each plunger and pipe connection would be only one-third as large as those of the three-plunger design.

Under some conditions greater economy in the amount of steel used in the lock structure would be obtained by using a greater number of smaller plungers more economically spaced if, by doing so, an automatically balanced system is the result.

The author stands on his previous assertion that Mr. Harris has solved a problem, actually in a very simple way, which has never been applied by others and is willing to leave it to those of unprejudiced minds who may be interested in the future application of mechanical lifts.

No. 1707

THE DESIGN OF RIVETED BUTT JOINTS

By Alphonse A. Adler, Brooklyn, N. Y.
Member of the Society

In this paper Schwedler's graphical method of designing riveted joints is analytically treated by the author, who states the fundamental assumptions employed and submits brief evidence for their justification.

A general equation is derived to determine the pitch in any row, and another to determine the efficiency in ideal cases. The design of cover plates is also considered.

Actual joints are calculated, using commercial dimensions, and the close agreement found between the ideal and calculated efficiencies seems to indicate that the scheme of analysis is consistent. The design of a quadruple-riveted joint furnishes the data for the single-, double- and triple-riveted joints by simply omitting the extra rows of rivets.

IN the design of riveted joints certain assumptions are made the justification of which is ascertained from their agreement with the results of experiments. Among the more important of these assumptions are the following:

a The tensile resistance of the joint is directly proportional to the net area under stress

b The shearing resistance of the joint is directly proportional to the total cross-sectional area of the driven size of rivets

c The bearing resistance of the joint is directly proportional to the total projected area of the driven size of rivets

d There is no bending stress in the rivets

e The frictional resistance of the joint is independent of the strength.

2 Of the foregoing, the first assumption is perhaps the one which involves the greatest discrepancy. This subject was studied by Coker[1] by means of an optical method. A perforated plate of xylonite was placed between the polarizer and analyzer of a pair of Nicol prisms. By this means Coker showed the intensity of stress around the rivet hole. An analytical treatment of this problem was

[1] *Engineering* (London), March 28, 1913, p. 439.

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

given by Suyehiro [1] and his results show fair agreement with those obtained experimentally by Coker. Suyehiro further shows that if the hole in a plate is plugged, the resultant stress around the rivet hole is very much less. In the latter case it corresponds to a riveted joint when the rivets are driven, as only in this case can the plate be loaded. The stress intensity between the rivet holes, nevertheless, is not uniform.

3 Assumptions *b* and *c* are common in structural and machine design and are dealt with at length in the more important texts on strength of materials. Assumption *d* is also common in structural design. Since the rivet completely fills the hole and the plates are comparatively rigid, there is little chance for bending and hence the bending stress is negligible.

4 The last assumption forms the basis of another method of designing joints the advantages of which are pointed out by Bach.[2] Briefly stated, joints in this country are designed for strength and checked for tightness, while in the method proposed by Bach the procedure is reversed and quite different from that in the former case.

5 After all, if actual joints are riveted up and tested to destruction, the data so obtained yield the maximum values of the stresses in tension, shear and bearing. If these values are then used in actual designs the process becomes reversible and errors made originally in the assumptions are automatically canceled. Indeed, this is the general plan followed in structural design today. For the limited sizes of plate and rivets used in boiler joints it should give results sufficiently reliable to inspire confidence. However, additional experimental data will always be useful.

ANALYTICAL TREATMENT

6 As is customary in riveted-joint design, the shearing resistance of a rivet is equated to the crushing resistance in order to find the smallest permissible diameter of rivet. Hence if d is the diameter of the rivet in inches, f_s' the shearing resistance in lb. per sq. in. per single surface in double shear, f_c' the crushing (or bearing) resistance in lb. per sq. in. in double-shear bearing, and t the thickness of the plates connected in inches,

$$\frac{\pi d^2}{2} f_s' = dtf_c' \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

[1] *Engineering* (London), August 14, 1914, p. 231.

[2] Bach, Die Maschinen-Elemente, chapter on Nietverbindungen.

from which

$$d = \frac{2tf_c'}{\pi f_s'} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

In other words, if d is chosen in accordance with Eq. [2] the rivet is equally likely to fail in shear or crushing because of the condition imposed in equating the shearing and bearing resistances.

7 For reasons to follow, assume a strip of plate of width w inches to be bent around the rivet somewhat like the link of a chain but of rectangular cross-section. If the resistance of this link under tension is equal to either the shearing or the crushing resistance of the rivet and if R denote this resistance,

$$R = 2wtf_t$$

or

$$w = \frac{R}{2tf_t} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

where f_t is the tensile resistance of the plate in lb. per sq. in.

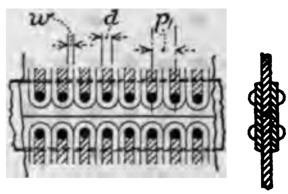


FIG. 1 SINGLE-RIVETED BUTT JOINT

8 The idea of conceiving a plate to be divided into hypothetical tension strips the resistance of each of which is equal to the strength of a rivet, first occurred to Schwedler[1], who used it as the basis of a graphical method. Unwin[2] has applied this method to boiler joints, but it is cumbersome and does not lend itself to slight changes in the assumed data without entailing a comparatively large amount of effort. An attempt to avoid this led to an analytical treatment which was published by the writer in 1916.[3] It was found later that the steps in this analysis could be concisely expressed by simple general equations, and these are the subject of this paper. A sufficient part of the article referred to is repeated here in order to insure continuity of treatment.

[1] Ueber Nietverbindungen. Lecture by J. W. Schwedler before the Architekten Verein zu Berlin; reprinted in their Wochenblatt, Nov. 22, 1867, et pp. 451, 461 and 472.

[2] Machine Design, vol. 1.

[3] *Power*, August 1, 1916.

. [7]

9 Fig. 1 shows a single-riveted joint in which the cover plate nearest the observer has been removed for convenience. The tension strips are shown around the rivet. The portion between the tension strips (shown shaded) could be cut out of the plate without impairing its strength. Of course in an actual boiler this could not be done, since this metal is required to enclose the contents.

10 Fig 2 shows a commercial quadruple-riveted joint with the distance between the rivet rows greatly exaggerated and in which the rivets are not staggered. The tension strips are numbered for convenience. It will be found that, starting below with any strip, it may be traced around a rivet and back again, thus showing that each strip has a particular rivet the resistance of which it adds to

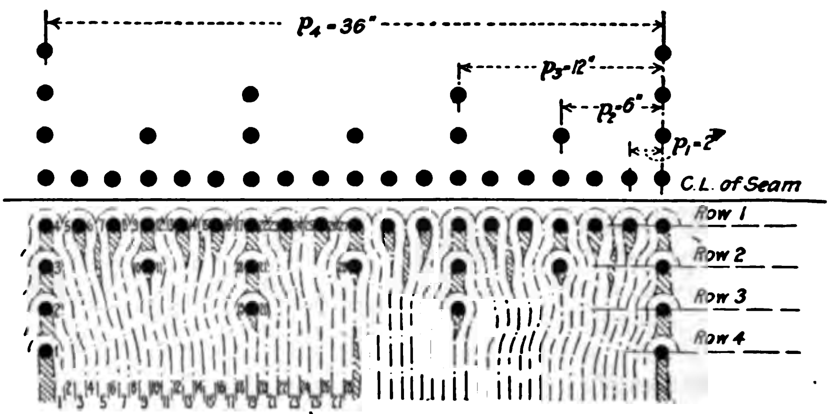


FIG. 2 LAYOUT OF A QUADRUPLE-RIVETED BUTT JOINT

the total resistance of the joint. The problem therefore rests on finding the maximum resultant strength.

11 If w is the width of a strip and d is the diameter of the rivet, then from Fig. 1 the most economical pitch for the first row of rivets occurs when the tension strips just touch each other. Denoting this pitch by p_1,

$$p_1 = 2w + d \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

12 The shaded area is metal available for rivets in subsequent rows. Thus in each pitch there is available a strip of width d. If m_1 represents the available metal per inch, then

$$m_1 = \frac{d}{p_1}$$

But since the amount of metal required to insert extra rivets in a second row is a strip of width $(2w + d)$, the pitch of the second row is

$$p_2 = \frac{2w + d}{m_1}$$

Since, however, from Eq. [4] $p_1 = 2w + d$ and $m_1 = \frac{d}{p_1}$, the pitch p_2 may be written

$$p_2 = \frac{p_1}{d/p_1} = \frac{p_1^2}{d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

13 Similarly, there is available for rivets in a third row a width of metal d in each distance p_2, or the width of available metal m_2 from the second row per inch of width of seam is

$$m_2 = \frac{d}{p_2}$$

Hence the pitch of the third row is

$$p_3 = \frac{2w + d}{m_2}$$

and replacing the values of numerator and denominator as before,

$$p_3 = \frac{p_1}{d/p_2}$$

But since p_2 is given in terms of p_1 by Eq. [5], this substitution results in

$$p_3 = p_1 \div \frac{d}{p_2}$$

$$= p_1 \div \frac{d}{p_1^2/d}$$

or

$$p_3 = \frac{p_1^3}{d^2} \quad , \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

14 In the same manner

$$m_3 = \frac{d}{p_3}$$

$$p_4 = \frac{2w + d}{m_3} = p_1 \div \frac{d}{p_3} = p_1 \div \frac{d}{p_1^3/d^2}$$

or

$$p_4 = \frac{p_1^4}{d^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

15 Therefore it will be seen in Eqs. [4], [5], [6] and [7] that the subscript of p in the left-hand member is the same as the exponent of the numerator p_1 in the right-hand member and is one greater than the exponent of d in the denominator. For a general equation, let n signify the number of the row; then

$$p_n = \frac{p_1^n}{d^{n-1}} \qquad [8]$$

16 To try a simple check, the equation should hold for the first row. Let therefore

$$p_1 = \frac{p_1^1}{d^0}$$

or

$$p_1 = p_1$$

which as it stands conveys little information; but recourse to Eq. [4] shows it to be equal to $2w + d$.

17 A general equation for the efficiency of a Schwedler joint is also possible. Thus, in an ideal joint all manners of failure are equally likely. Choosing the most convenient form for the equation, take the case for the efficiency in tension of the last row:

$$\begin{aligned} e_n &= \frac{(p_n - d)\, tf_t}{p_n tf_t} \\ &= \frac{p_n - d}{p_n} \\ &= 1 - \frac{d}{p_n} \qquad [9] \end{aligned}$$

As p_n becomes very large by increasing the number of rows, the efficiency approaches unity or 100 per cent.

18 A slightly different form might be obtained for the general equation of the efficiency. Since p_n in Eq. [9] may be replaced by its value from Eq. [8],

$$\begin{aligned} e_n &= 1 - \frac{d}{p_1^n \div d^{n-1}} \\ &= 1 - \frac{d^n}{p_1^n} \\ &= 1 - \left(\frac{d}{p_1}\right)^n \qquad [10] \end{aligned}$$

This equation expresses the same result as Eq. [9]. For example, the ratio d/p_1 is always less than unity, and the fraction raised to any positive power will approach zero as n becomes large. Again, the efficiency approaches 100 per cent as the number of rows is increased.

19 For high-efficiency joints the cover plates must be designed from fundamental principles rather than from the empirical rules given in certain textbooks. On the plate the tension strips must all pass through the last row of rivets. On the cover plates the condition is just the reverse, that is, the strips all pass through the first row of rivets. Since the pitch and tensile stress are fixed, the required area of metal may be obtained by suitably determining the thickness.

20 The total load on a strip of plate of width p_1 is

$$\frac{p_1 t f_t e}{100}$$

where e is the actual efficiency of the joint in per cent. The resistance of two cover plates is

$$2\,(p_1 - d)\,t_c f_t$$

where t_c is the thickness in inches of one cover plate. For equal strength these must be equated; hence

$$2\,(p_1 - d)\,t_c f_t = \frac{p_1 t f_t e}{100}$$

from which

$$t_c = \frac{p_1 t e}{200\,(p_1 - d)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [\mathbf{11}]$$

APPLICATION OF THE METHOD

21 Assume a plate $\frac{1}{2}$ in. thick. Let f_t = 60,000 lb. per sq. in., f_s' = 45,000 lb. per sq. in., and f_c' = 100,000 lb. per sq. in. From Eq. [**2**],

$$d = \frac{2 \times \frac{1}{2} \times 100{,}000}{\pi \times 45{,}000} = 0.707 \text{ in.}$$

The value of R in Eq. [**3**] is obtained from the shearing or bearing resistance in Eq. [**1**]. Thus for shear,

$$R = \frac{\pi \times (0.707)^2 \times 45{,}000}{2} = 35{,}300 \text{ lb.}$$

From Eq. [**3**]

$$w = \frac{35{,}300}{2 \times \frac{1}{2} \times 60{,}000} = 0.588 \text{ in.}$$

Then from Eqs. [4], [5], [6], and [7]

$$p_1 = (2 \times 0.588) + 0.707 = 1.883 \text{ in.}$$

$$p_2 = \frac{(1.883)^2}{0.707} = 5.03 \text{ in.}$$

$$p_3 = \frac{(1.883)^3}{(0.707)^2} = 13.4 \text{ in.}$$

$$p_4 = \frac{(1.883)^4}{(0.707)^3} = 35.7 \text{ in.}$$

The corresponding efficiencies for these ideal joints are, from either Eqs. [9] or [10],

$$e_1 = 1 - \frac{0.707}{1.883} = 62.5 \text{ per cent (Single-riveted)}$$

$$e_2 = 1 - \frac{0.707}{5.03} = 85.9 \text{ per cent (Double-riveted)}$$

$$e_3 = 1 - \frac{0.707}{13.4} = 94.7 \text{ per cent (Triple-riveted)}$$

$$e_4 = 1 - \frac{0.707}{35.7} = 98.0 \text{ per cent (Quadruple-riveted)}$$

22 To design commercial joints from the foregoing, choose, say, $d = 0.75$ in., $p_1 = 2$ in., $p_2 = 6$ in, $p_3 = 12$ in., $p_4 = 36$ in., as shown in Fig. 2. This will afford a joint having simple ratios of rivet pitches from row to row. The calculated efficiencies for commercial riveted joints are found in the usual way. A simple calculation will show that $R = 37{,}500$ lb. in bearing. Hence no rivets will fail in shear since the shearing resistance is 39,000 lb. Thus, for a quadruple-riveted joint where the unit pitch is 36 in., it is necessary to consider —

a Bearing resistance of all rivets in a 36-in. strip.
b Tensile resistance of row 4
c Tensile resistance of row 3 plus bearing resistance of row 4
d Tensile resistance of row 2 plus bearing resistance of rows 3 and 4
e Tensile resistance of row 1 plus bearing resistance of rows 2, 3 and 4.

The values of these resistances are as follows:

a $28 \times 37{,}500 = 1{,}050{,}000$ lb.
b $35.25 \times 60{,}000 \times \frac{1}{2} = 1{,}057{,}500$ lb.
c $(33.75 \times 60{,}000 \times \frac{1}{2}) + (1 \times 37{,}500) = 1{,}050{,}000$ lb.

d $(31.50 \times 60{,}000 \times \frac{1}{2}) + (4 \times 37{,}500) = 1{,}095{,}000$ lb.
e $(22.5 \times 60{,}000 \times \frac{1}{2}) + (10 \times 37{,}500) = 1{,}050{,}000$ lb.

23 It will be seen that the liability of rupture will occur under items ***a***, *c* or ***e***. The resistance of an unperforated strip 36 in. wide is $36 \times 60{,}000 \times \frac{1}{2} = 1{,}080{,}000$ lb. The efficiency is consequently

$$e = \frac{1{,}050{,}000}{1{,}080{,}000} \times 100 = 97.2 \text{ per cent}$$

The thickness of the cover plates is, from Eq. [**11**], approximately

$$t_c = \frac{2 \times \frac{1}{2} \times 97.2}{200 \times 1.25} = \frac{25}{64} \text{ in.}$$

24 To make a triple-riveted joint, omit the rivets in row 4 of Fig. 2. A similar set of calculations will show that in a 12-in. strip failure is likely to occur through —

a Bearing resistance of all rivets in a 12-in. strip
b Tensile resistance of row 3
c Tensile resistance of row 2 plus bearing resistance of row 3
d Tensile resistance of row 1 plus bearing resistance of rows 2 and 3.

These resistances have respectively the following values:

a $9 \times 37{,}500 = 337{,}500$ lb.
b $11.25 \times 60{,}000 \times \frac{1}{2} = 337{,}500$ lb.
c $(10.5 \times 60{,}000 \times \frac{1}{2}) + (1 \times 37{,}500) = 352{,}500$ lb.
d $(7.5 \times 60{,}000 \times \frac{1}{2}) + (3 \times 37{,}500) = 337{,}500$ lb.

The lowest resistance of the joint is for items *a*, *b* and *d*. The initial strength of the plate is $12 \times 60{,}000 \times \frac{1}{2} = 360{,}000$ lb., hence the efficiency is

$$e = \frac{337{,}500}{360{,}000} \times 100 = 93.7 \text{ per cent}$$

The thickness of the cover plates is approximately

$$t_c = \frac{2 \times \frac{1}{2} \times 93.7}{200 \times 1.25} = \frac{3}{8} \text{ in.}$$

25 For a double-riveted joint omit rows 3 and 4 of the quadruple joint and the unit strip becomes 6 in. wide. In this case failure may occur through —

a Bearing resistance of all rivets in a 6-in. strip
b Tensile resistance of row 2
c Tensile resistance of row 1 plus bearing resistance of row 2:

Numerically these become

uniformly distributed among the rivets of a given joint. To prevent too serious a deviation from this premise, the cover plates might be designed in the following way: Let it be assumed at the start that each rivet is properly driven; then, in an ideal case, the same metal-to-metal contact exists among all the rivets. For this condition to prevail after loading requires that the same stress exist in the cover plates as exists in the connected plate. Under these conditions the deformation under load will be the same for both cover plates and connected plate. In commercial joints, departures from the ideal case arise which may be somewhat obviated by scalloping the cover plates for triple- and quadruple-riveted joints, as shown in Fig. 3. This will decrease the rigidity of the cover plate and to some extent prevent the rivets of the outer rows from taking an undue share of the whole load.

DISCUSSION

Sherwood F. Jeter (written). Professor Adler's paper interests me for two reasons; first, because I criticised his article referred to as having been published in *Power*; second, on account of my general interest in the general subject of riveted joints. Anyone interested should read the original article in *Power* of August 1, 1916, page 159, my criticism of it on page 281 of the issue of August 22, 1916, and Professor Adler's rejoinder on page 432 of the issue of September 19.

My criticisms of the paper under discussion are as follows:

First, the introduction of Schwedler's graphical method appears to complicate rather than aid in an analytical treatment of the subject.

Second, the general formula for the pitch of rivets, as derived by Professor Adler, while correct, is not in the simplest form or the one most useful to the designer.

Third, the author does not show how the design of a maximum efficiency joint, or the principles involved in the design of such a joint, may be made useful to the boiler designer in arriving at the proportions of a commercial joint of the highest efficiency.

Fourth, for the best results in designing a commercial joint with several rows of rivets, both as regards rivet spacing and strength, it is not advisable to design such a join with more rows of rivets than required and then omit the rows that are not considered necessary, as is done in the paper.

Fifth, joints of maximum efficiency are not practical for use in boiler construction, even with only one row of rivets, except for the thinner plates.

In what follows, I expect to show the above statements to be facts.

The letters used in the formulæ will have the following meanings:

P = The pitch of rivets, inches. Where there is more than one row, this refers to the pitch on the outer row.
t = The plate thickness, inches.
d = The diameter of the rivet hole, or driven diameter of rivet, inches.
C = The crushing strength of the plate, lb. per sq. in.
T = The tensile strength of the plate, lb. per sq. in.
S = The shearing strength of the rivets, lb. per sq. in. of sheared surface.
E = The joint efficiency.

The efficiency of a single-riveted joint with double butt straps is expressed by one of the three following formulæ, the one giving the minimum value, of course, determining the efficiency.

Equation [1] is based on the breaking of the net section of the plate between the rivet holes; formula [2], on the crushing of the plate in front of the rivets; and formula [3], on the shearing of the rivets in double shear.

$$E = \frac{(P-d)tT}{PtT} = 1 - \frac{d}{P} \qquad [1]$$

$$E = \frac{Ctd}{PtT} = \frac{Cd}{PT} \qquad [2]$$

$$E = \frac{0.7854d^2(2S)}{PtT} \qquad [3]$$

Examining Eq. [1] and [2], it is evident that as the value of P increases, E increases by [1] and decreases by [2]. Therefore, starting with the smallest possible value for P, without considering the interference of the rivet heads, where P would equal d, that is the rivet shanks would touch each other (in which case the efficiency would be zero by [1] and $\frac{C}{T}$ by [2]), and increasing the value of P, the efficiencies by these two formulæ approach each other and become equal at some higher value for P.

Equating [1] and [2], the value of P, in terms of C, T and d, is found where the efficiency by [1] and [2] will be equal, thus:

$$\frac{Cd}{PT} = 1 - \frac{d}{P}$$

or

$$P = d\left(\frac{C}{T} + 1\right) \qquad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \qquad [4]$$

It is evident that a single-riveted joint cannot have a higher efficiency than when P has this value, for a change in the value of P will result in a lower efficiency by either Eq. [1] or [2].

It is seen by examining Eq. [3], that the efficiency due to the shearing of the rivets, for any finite value of P, may be anything desired, providing the value of d is not limited.

To determine the value of d, if the efficiency by Eq. [3] is to be the same as for Eq. [2], equate these, as follows:

$$\frac{Cd}{PT} = \frac{0.7854d^2(2S)}{PtT}$$

or

$$d = \frac{tC}{0.7854(2S)} \qquad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \qquad [5]$$

Since the value of d, as given by Eq. [5], is such that the crushing of the plate and the shearing of the rivets are equally possible, any increase in d beyond this value would cause the shearing of the rivets to cease to be a factor. This is so, since the shearing strength increases as the square of d, while the crushing strength increases directly as d. Therefore, to design a single-riveted joint of maximum efficiency, it is only necessary to select rivets of such size that:

$$d = > \frac{tC}{0.7854(2S)} \qquad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \qquad [6]$$

and space them so that:

$$P = d\left(\frac{C}{T} + 1\right)$$

Since the efficiency of such a joint will be expressed by either Eq. [1] or [2], that is the shearing of rivets will not be a factor, the efficiency may be written from Eq. [1],

$$E = 1 - \frac{d}{d\left(\frac{C}{T} + 1\right)}$$

or

$$E = 1 - \frac{1}{\frac{C}{T} + 1}$$

If Fig. 4 represents a single-riveted joint arranged for maximum efficiency, the portion of the plate in front of the rivets (equivalent to a strip of width d for each pitch), shown shaded, is not required for transmission of stress to the rivets. Therefore, the efficiency of such a joint may be increased by the addition of another row of rivets outside the first. To secure maximum results from this added row of rivets, the rivets must be so spaced that there will be just enough of these unused strips included in a pitch of the added row as required for a pitch of rivets on the first row. This is true, since it requires a width of plate equal to a pitch length on the first

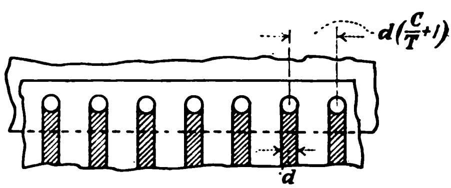


FIG. 4. SINGLE-RIVETED JOINT ARRANGED FOR MAXIMUM EFFICIENCY

row to include a rivet hole and the necessary plate to transmit stress to the rivet to be driven in it.

Calling the pitch of rivets on the second row P_2, to comply with the above requirements,

$$P_2 = P \times \frac{P}{d}$$

or from Eq. [4],

$$P_2 = d\left(\frac{C}{T} + 1\right)\left(\frac{C}{T} + 1\right)$$

or

$$P_2 = d\left(\frac{C}{T} + 1\right)^2$$

Similarly for a third row,

$$P_3 = d\left(\frac{C}{T} + 1\right)^3$$

and for m rows of rivets

$$P = d\left(\frac{C}{T} + 1\right)^m \quad \text{[7]}$$

where P is the pitch in the outer row.

From the method of design, these joints are equal in strength by any method of failure to be considered; therefore, the efficiency may be given by Eq. [1], thus,

$$E = 1 - \frac{1}{\left(\frac{C}{T} + 1\right)^m} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [8]$$

As will be noted from Eq. [7], the pitch of a joint of maximum efficiency varies directly as the value of d, and the smallest value for P will result when d just equals the value determined by Eq. [5].

It will be noted that the increases in pitch from row to row are determined by the ratio of C to T. If this ratio should be a whole number, then there would be a simple ratio for the spacing of rivets from row to row. Since, with boiler-plate material, the ratio between the crushing strength and the tensile strength is not usually a whole number, maximum-efficiency joints will not have symmetrically spaced rivets in the different rows.

It will be noted that Eq. [7] expresses the value for pitch on the first row as well as that for any other row. Professor Adler's general formula for pitch (Eq. [7] of his paper) requires a separate determination for the pitch of the first row, which is involved with the determination of two other values, specified by him as w and R. The above formula, Eq. [8], for efficiency, is for the same reasons more simple and easier to apply than the one which the paper provides. The value $\left(\frac{C}{T} + 1\right)^m$, when found, serves equally well for the determination of the pitch or the efficiency.

The above, I consider, covers the first two points on which I have undertaken to criticise this paper.

When Professor Adler shows how to design a commercial joint it would appear to the reader that he merely selected the next larger commercial rivet-hole size and the next whole number for pitch, above the values he derived from a joint of maximum efficiency. The particular values chosen by Professor Adler for illustration give the best results.

To design a commercial joint of the highest efficiency, a rivet-hole diameter should be chosen that will comply with the conditions specified by Eq. [6], keeping in mind that the smaller the rivet-hole diameter the smaller the pitch. Next determine the pitch of the rivets on the different rows by Eq. [7]. Decide the maximum spacing of rivets that will permit tight calking and select the row of rivets where this pitch is most closely approached without being exceeded. This will be the row to which the edges of the outer straps will have to be cut. Arrange the rivets in the first or second row (and other outer rows) beyond the row having a calking pitch, so that there will be a simple ratio or ratios between the pitches of succeeding rows, the ratio or ratios chosen to be as nearly as possible to the value of $\left(\frac{C}{T}+1\right)$.

To show how this method works out in practice, assume the same values as are used in the paper for C, T $(2S)$ and t, of 100,000, 60,000, 90,000, and 1/2, respectively. By Eq. [6],

$$d = > \frac{1/2 \times 100{,}000}{0.7854 \times 90{,}000} = 0.707$$

the same as Professor Adler finds.

The next larger commercial rivet-hole size is 3/4 in., therefore, by Eq. [7], for a joint of maximum efficiency, the pitches would be:

$$\text{For first row, } 3/4\left(\frac{100{,}000}{60{,}000}+1\right) = 2 \text{ in.}$$

$$\text{For second row, } 3/4\left(\frac{100{,}000}{60{,}000}+1\right)^2 = 5.33 \text{ in.}$$

$$\text{For third row, } 3/4\left(\frac{100{,}000}{60{,}000}+1\right)^3 = 14.21 \text{ in.}$$

$$\text{For fourth row, } 3/4\left(\frac{100{,}000}{60{,}000}+1\right)^4 = 37.89 \text{ in.}$$

The pitch on the second row, of 5.33 in., would not permit tight calking with any strap thickness likely to be used with $\frac{1}{2}$-in. plate, but since Professor Adler shows an even wider pitch along the calking edge, it will be assumed, for the sake of comparison, that this pitch could be calked. Spacing the rivets in the two outer rows with the simple ratios of 1 to 3 and 1 to 2, since these are the nearest to 1 to 2.66, which is the ratio required for maximum efficiency, the design of joint will be as shown in Fig. 5. There are 25 rivets in a unit section of this joint, and calculating the strength by the different modes of failure, as is done in the paper, would result as follows:

Crushing strength of 25 rivets:

a. $25 \times 37500 = 937{,}500.$

Tensile strength of plate at outer net section:

b. $31.25 \times .5 \times 60{,}000 = 937{,}500.$

Strength of plate along next to outer row plus crushing of one rivet:

c. $30.5 \times .5 \times 60{,}000 + 37{,}500 = 952{,}500.$

Strength of plate along next to inner row plus shearing of three rivets:

$$d.\ 27.5 \times .5 \times 60,000 + 3 \times 37,500 = 937,500.$$

Strength of the solid plate:

$$32 \times .5 \times 60,000 = 930,000.$$

Therefore, the efficiency would be

$$\frac{937,500}{960,000} = 97.65 \text{ per cent.}$$

There is no need to consider failure at the inner row of rivets, since the spacing of the rivets along the two inner rows was such that failure was equally likely on either: therefore, the result would be the same as by method *d* if the calculation had been made.

While I do not wish to be understood as advocating this abnormal joint for use on a boiler, I do contend that Fig. 5 shows a

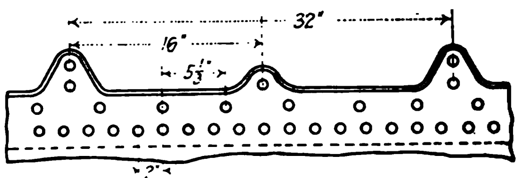


FIG. 5. QUADRUPLE-RIVETED JOINT

more practical joint than the one Professor Adler has designed, and one having a higher efficiency. The method of design just given covers my third point of criticism.

If only three rows of rivets were to be used in a commercial joint and it was considered that the 5.33-in. pitch on the second row would permit proper calking, then the rivets in the third row could be spaced 14.21 inches apart. Such a joint would be like Fig. 6, and it would, of course, be a maximum efficiency joint and have an efficiency of

$$1 - \frac{1}{\left(\frac{100,000}{60,000} + 1\right)^3}$$

since there are three rows of rivets used; that is, the efficiency would be 94.72 per cent.

Again, if the joint was to be double-riveted only, then the calking edge of the strap would be straight, that is, the same as Fig. [6] with the projections used to include the third row of rivets omitted.

This joint would also be a maximum-efficiency joint and the efficiency would be

$$1 - \frac{1}{\left(\frac{100,000}{60,000} + 1\right)^2}$$

or 85.93 per cent. The above indicates that to design a commercial joint of the highest efficiency the procedure I have suggested should be followed rather than the method proposed by Professor Adler.

Coming to my fifth point of criticism, Fig. 7 gives the relationship between rivet diameters and plate thicknesses that are required for joints of maximum efficiency, where the crushing strength is 95,000 lb. per sq. in., and the shearing strength is 44,000 for single and 88,000 lb. per sq. in. for double shear. The point of intersection of a horizontal line, representing the rivet diameter, and a vertical line representing the plate thickness, must lie to the left and

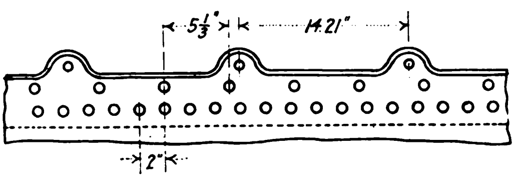


FIG. 6. TRIPLE-RIVETED JOINT

above either one or the other of the solid lines, which one depending on whether the rivets are in single or double shear. The dotted curved line shows the average or usual relationship between rivet diameters and plate thicknesses selected by designers of experience.

It will be seen that the range of plate thicknesses is very limited for maximum efficiency joints, even where the rivets are in double shear.

Without doubt, the most practical way to design commercial joints is to decide on an arrangement of rivets that will best meet the conditions of construction and make a diagram giving the efficiencies of such joints. By the aid of such diagrams, the joint that will best meet the practical requirements for calking, the maximum size of rivet that may be driven with the apparatus available and other practical considerations of a similar character may be selected.

THE AUTHOR. Considering the first of Mr. Jeter's criticisms, I must call his attention to the paragraphs leading up to his Eq.

[7]. It reads thus: "If Fig. 4 represents a single-riveted joint . . . ." Now the notion here expressed is precisely the same as used in my paper, and is Schwedler's analysis. It leads to an expression which in the notation of my paper is $p_2 = ntd$ I have left the equation in this form in my paper whereas Mr. Jeter replaces p_1^2 by its value

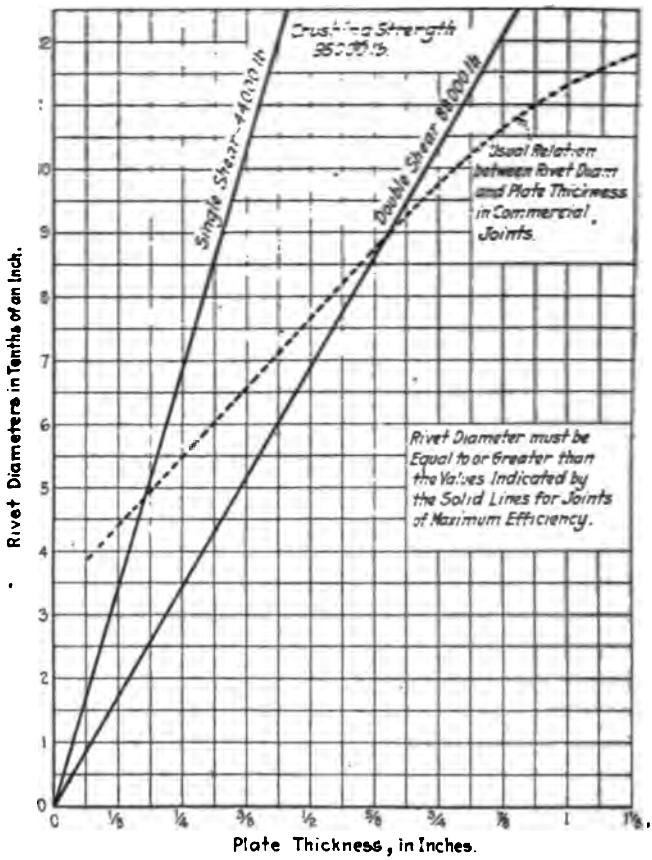


FIG. 7. RELATION BETWEEN RIVET DIAMETERS AND PLATE THICKNESS FOR JOINTS OF MAXIMUM EFFICIENCY

and arrives at his Eq. [7]. Since Mr. Jeter's transformations are correct his final Eq. [7] is equivalent to mine. Thus, I do not see the force of his first criticism.

Taking the second point, the Eq. [8] given by me contains only d and p_1. Now it is necessary to know the value of d in any case and since no joint may be designed with less than one row of rivets the value of p_1 must also be known. I do not see how any equation can be simpler. Furthermore, had Mr. Jeter used the value 0.707 in. for d instead of $\frac{3}{4}$ in. in showing the use of his Eq. [7], he would have obtained ideal pitches numerically the same as mine. It is incorrect to use commercial rivet diameters in equations based on ideal considerations.

For the third point of criticism, the commercial mode of procedure is to calculate the ideal joint and approximate the practical joint. It is impossible to derive an equation that will give an answer in commercial dimensions. The approximation of the practical joint is not a serious matter for anyone with a little experience. If the outer pitch is too great for calking, scallop the cover plates or decrease the pitch. Any change in the ideal joint is made at a sacrifice in efficiency.

For the fourth point, it is not necessary to design a quadruple joint and omit the extra rows to obtain, say, a double-riveted joint. When p_1 and p_2 are found the problem is finished. The illustration given in the paper shows the similarity of the joints and the possibility of standardization.

Mr. Jeter's last point requires additional evidence on his part. He has not shown why designers of experience use his so-called "usual relationship." There certainly must be some reason for it and, if there is, it must be possible to state it.

No. 1708

ECONOMICAL SECTION OF WATER CONDUIT FOR POWER DEVELOPMENT

By Cary T. Hutchinson, New York, N. Y.
Member of the Society

In this paper the author deduces a formula for determining the best slope and size of a conduit for supplying water to a power plant, this formula taking into account the construction costs, the value of the power recovered, the rate of returns expected on expenditures made, etc. The resulting relation between the best area and the flow in second-feet for chosen values of factors is shown in a logarithmic chart, and it is pointed out how this chart may be modified so that the relation for other costs and unit prices can be easily determined.

THE literature of water-power engineering accessible to the writer does not include a discussion of the method of determining the economical section of a water conduit for supplying water to a power plant.

2 In what follows a formula for the best slope and size of conduit is deduced, which takes into account in a practical manner the construction costs, the value of the power recovered, and the rate of returns expected on expenditures made, as well as the other physical conditions of the problem. The resulting relation between the best area and the quantity of water is shown in Fig. 2, for chosen values of factors entering the problem, and it is pointed out how, by a simple modification of the graphs of this figure, the relation for other costs and unit prices can be easily determined. One interesting result is that for a flume, or for any conduit, for which the increment cost of increasing the capacity by a relatively small amount is proportional to the surface the best speed of flow of the water is constant, independent of the size of the conduit.

3 The economical section is evidently that resulting in the greatest net earnings of the power plant under the conditions controlling the market where the power is delivered. Inasmuch, however, as this section must be determined in advance of complete

Presented at the Spring Meeting, Detroit, Mich., June 1919, of The American Society of Mechanical Engineers.

knowledge of market conditions, it is clear that only an approximation can be made, and that a ready method to determine the variation in net earnings for a large range of sections and shapes of water conduit may be useful.

4 Assuming that a certain shape and slope of water conduit is fixed upon provisionally, the question is whether some change either in the slope or in the shape or size of the section will result in an increase in net earnings. Any increase in the dimensions of the conduit will obviously entail an increase in construction cost, and hence an increase in annual charges. This increase in annual charges is limited, practically speaking, to interest, amortization and profit, inasmuch as only small changes in a quantity which itself is a small part of the total are under consideration. For instance, under ordinary conditions the loss in the water conduit may vary from, say, 5 per cent to 10 per cent of the total power; it is a variation of possibly 25 per cent one way or the other in this 5 or 10 per cent that is involved.

5 It is therefore evident that no increase in operating charges, or maintenance, or repairs need be considered, and that the changes in design of the conduit should carry charges only for interest, amortization, and profit. An allowance for profit on the additional expenditure must be included, since every dollar invested should earn its share of profit as well as its fixed charges.

6 The increase in power resulting from an increase in the size of the conduit brings in a certain addition to gross earnings. Against this, in theory, should be charged the costs of operation and maintenance on the additional equipment and machinery required to deliver this power to the market; but for the same reasons stated in considering the water conduit, all these charges against the additional gross earnings may be ignored in this analysis, as they are negligible in amount, due to the fact that the increase in the power output is small. There would, in fact, be no increase in operating charges, and under practical conditions there would be no increase in equipment, and therefore no increase in fixed charges on equipment.

7 The matter then reduces to the comparison of the additional gross earnings from the power recovered by an increase in the size of the conduit on the one hand, and the additional interest, amortization, and profit on the cost of the enlargement of the conduit on the other.

8 The determination of additional power is simple, involving

merely the overall efficiency from the water to point of delivery. A consideration of the value of this increased power is a matter of judgment on the part of the engineers and executives of the enterprise, giving attention to the market conditions under which the power is sold, and particularly to the load factor.

9 The determination of the additional cost of the conduit, however, is more difficult, inasmuch as this cost depends in theory not only on the area of the cross-section of the conduit, but also upon its shape; that is, upon the hydraulic radius or the wetted perimeter. The relation between the area and the wetted perimeter differs, for example, for a rectangular, a circular, or a hexagonal conduit, and cannot be expressed in a simple equation to cover all shapes of conduit. The practical way to handle the problem is to fix upon one shape of conduit, determine the economical area and slope for this shape, and then follow out a similar procedure for such other shapes as may be practicable in the case under consideration. This determination being made for the several possible shapes, the best result is then selected.

10 The procedure indicated in the foregoing general discussion can be expressed symbolically as follows:

Let Q = flow in sec-ft., taken as constant
L = length of conduit, ft.
A = area of conduit, sq. ft.
s = slope of conduit, ft. per ft. of length
r = hydraulic radius of conduit, ft.
w = wetted perimeter of conduit, ft.
v = speed of flow, ft. per sec.
C = constant in the Chézy formula
e = efficiency from water to point of delivery $\times$ 0.085.

Then the power loss p in the conduit in kilowatts will be

$$p = eQsL. \qquad [1]$$

11 In this equation s is the variable, and any change of p is due to a change of s and is expressed by

$$dp = eQL\,ds. \qquad [2]$$

12 If m is the annual value of one kilowatt under the ruling conditions, then

$$mdp = meQL\,ds \qquad [3]$$

is the added gross (and net) earnings due to the change in s.

13 As to the cost of increasing the capacity of the conduit: the flow is assumed to be given by the Chézy formula

$$v = \frac{Q}{A} = C\,(rs)^{0.5}. \qquad [4]$$

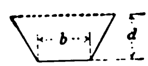
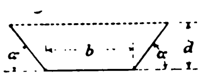
14 The best size of conduit is to be determined for a known value of the flow Q; that is, in Equation [4] Q is to be taken as constant. In this case,

$$\frac{dQ}{ds} = 0$$

and

$$\frac{dA}{ds} = -\frac{2}{5}\frac{A}{s};\quad \frac{dw}{ds} = -\frac{1}{5}\frac{w}{s};\quad \frac{dr}{ds} = -\frac{1}{5}\frac{r}{s}. \qquad [5]$$

TABLE 1 VALUES OF SECTION CONSTANT a FOR VARIOUS SECTIONS

Shape of Section	Hydraulic Radius	Cross-Section	Section Constant	
			a	$\sqrt{a}$
Semicircle, radius = r	$r/2$	$\pi r^2/2$	2π	2.51
Square, side = d	$d/3$	d^2	9	3.00
Half-square, depth = d	$d/2$	$2d^2$	8	2.83
Hexagon, half-full, depth = d 	$d/2$	$\sqrt{3}\,d^2$	$4\sqrt{3}$	2.55
Prism: 				
Tan a = 1, 1.5, $b = 4d$	0.72 d	5.5 d^2	10.6	3.26
Tan a = 1 2, $b = 10d$	0.83 d	12 d^2	16.0	4.00

15 With Q constant, a change in s can be offset by a change in either A or r, or in both; that is, either the size or the shape of the conduit can be varied to keep Q constant.

16 There is no way of expressing a general relation between A and r, but for any chosen shape, as, for example, a rectangle or semicircle, the area is proportional to the square of any linear dimension; that is,

$$A = ar^2 = wr = \frac{w^2}{a}. \qquad [6]$$

17 The value of the section constant a varies, but for usual forms the differences are not great, and the influence of changes in a

on the economical section is slight; in fact, it can be shown that for the best section

$$A \sim a^{1/7}$$

for the conditions of Equation [7a]. Table 1 gives values of a for the usual shapes. For preliminary calculations $a = 9$ may be used.

18 The cost of a water conduit can be expressed as a constant, representing the cost of a large part of the preliminary work and plant, plus an amount depending on the size and surface area. In general, the cost per foot may be expressed by

$$D = k_0 + kA^n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

where D = cost per foot, dollars

k_0 = constant part of cost per foot

k = a constant

n = an exponent whose value lies between 1 and 0.5.

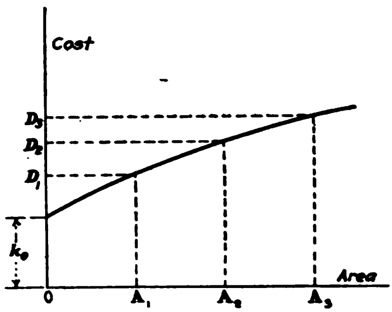


FIG. 1 RELATION BETWEEN SECTIONAL AREA OF CONDUIT AND COST

19 In any specific case, when all the conditions are known, estimates of the total cost per foot of the conduit should be made for three or more different cross-sections; plotting these values will enable both k and n to be determined.

20 For example, let Fig. 1 represent the cost per foot for a certain conduit; then k_0 is given at once by the curve, and from

$$\frac{D_2 - k_0}{D_1 - k_0} = \left(\frac{A_2}{A_1}\right)^n$$

n is determined. Each pair of points should be used and the value of n found. The values of n and k_0 being known, k may be obtained from

$$k = \frac{D_1 - k_0}{A^n}$$

21 Other methods could be used. The gist of the matter is that the accurate way is to make detailed estimates for several cross-sections and determine the constants from an analysis of these estimates.

22 Two extreme cases simplify the formula: First, when the increment cost is proportional to the area, as in a heavy rock cut, then

$$D = k_0 + kA \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7a]$$

and, second, where the increment cost is proportional to the surface, or the wetted perimeter, as for a flume, then

$$D = k_0 + kA^{0.5}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7b]$$

These are considered later.

23 If i represents the total rate of returns expected on all expenditures on the property, including interest, amortization and profit, then

$$I = iL\,(k_0 + kA^n), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [8]$$

gives the total returns from this investment, and a change ds in s calls for a change in returns of

$$dI = niLkA^{n-1}\frac{dA}{ds}\,ds \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

or, from Equation [5],

$$dI = -\frac{2\,niLkA^n}{5\,s}\,ds. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [10]$$

24 This saving, due to an increase in s, must be at least equal in value to the power lost, and indeed should exceed it by some margin; this margin can be included in the overall rate of return i, and therefore

$$dI = mdp \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [11]$$

25 Substituting in [11] from [10] and [3], there results

$$5\,mcQs = 2\,kniA^n$$

Substituting further from [6] and [4], namely,

$$s = \frac{v^2}{C^2r} = \frac{v^2}{C^2\,(A/a)^{0.5}}, \quad \text{and} \quad Q = Av,$$

gives finally

$$A^{n-0.5} = \frac{2.5\,mca^{0.5}}{nikC^2}\,v^3. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [12]$$

This may also be written

$$A^{(n+2.5)} = \frac{2.5\,mca^{0.5}}{nikC^2}\,Q^3 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [13]$$

If

$$N = \frac{2.5\,mea^{0.5}}{nikC^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [14]$$

then

$$A^{(n+2.5)} = NQ^3 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [15]$$

26 The best way to handle this equation for engineers is by logarithmic plotting. From [15]

$$\log A = \frac{\log N}{(n + 2.5)} + \frac{3}{(n + 2.5)} \log Q \quad . \quad . \quad . \quad . \quad [16]$$

When n is known, this can be readily plotted for any range of Q desired. As an illustration, assume

$$m = \$10,\ e = 0.67 \times 0.085 = 0.057$$
$$i = 0.15, \quad C = 120$$

TABLE 2 VALUES COMPUTED FROM FIG. 2

Q	A	V	r	s (Ft. per 1000)
100	3	4.35	1.60	0.830
500	100	5.00	3.34	0.520
1,000	191	5.22	10.50	0.180
2,500	450	5.56	16.70	0.128
10,000	1650	6.08	33.30	0.077

then

$$N = \frac{10^{-3} \times 2}{nk} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [17]$$

If, further, $n = 0.75$ and $k = \$0.10$,

$$N = 10^{-2} \times 2.67$$

and

$$\log A = \frac{\log 10^{-2} \times 2.67}{3.25} + 0.925 \log Q$$
$$= -0.485 + 0.925 \log Q \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [18]$$

27 Fig. 2 is the logarithmic graph of Equation [18] for values of Q from 100 to 10,000, in four parts; for the line BC the ordinates are to be multiplied by 10 and the abscissæ by 100; for CD, by 100 and 100; for DE, by 100 and 1000; for EF by 1000 and 1000 — all as indicated by the figure. From this figure, Table 2 is readily computed.

28 Two special cases are of particular interest: First, when $n = 0.5$ and the increment cost is proportional to the surface; this would approximate the case of a flume or a concrete-lined canal in earth. Here

$$A^3 = NQ^3$$

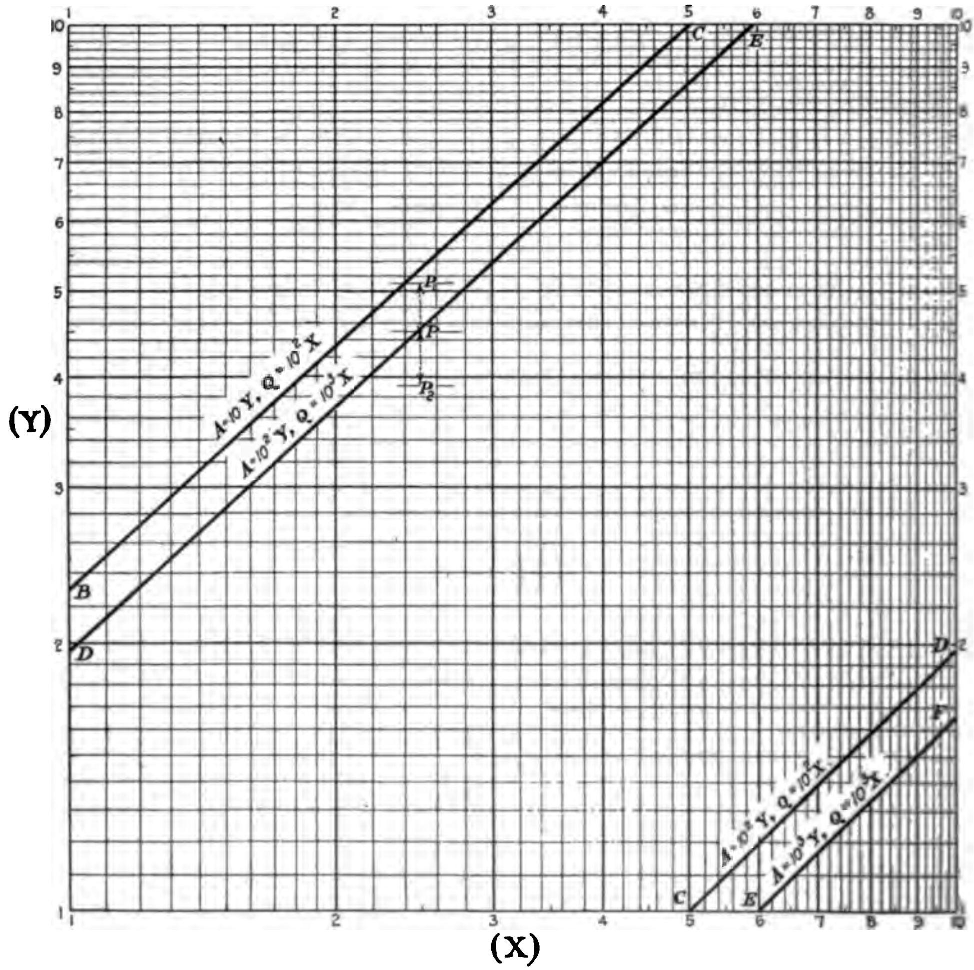


FIG. 2 LOGARITHMIC GRAPH OF EQUATION [18] FOR VALUES OF Q FROM 100 TO 10,000 SEC-FT.

and, since $v = Q/A$,

$$v^3 = \frac{1}{N} \qquad [19]$$

or there is one best speed of flow independent of the size of conduit. This is a somewhat surprising result.

29 The second case is when the increment cost is proportional to the amount of excavation, as for a costly rock cut; here $n = 1.0$, and

$$A^{3.5} = NQ^3 \qquad [20]$$

30 This can be solved either by plotting, as in Fig. 2, or as follows:

$$A^{0.5} = Nv^3 \qquad [21]$$

and

$$Q = Av$$

Calculate $A^{0.5}$ and A from [21], assuming values for v; then find Q, which can be plotted to A.

TABLE 3

N	Q	A	v
$10^9 \times 4.00$	2500	510	4.90
$10^9 \times 2.67$	2500	450	5.56
$10^9 \times 1.78$	2500	395	6.32

31 Variations in A resulting from other values of the unit costs than those used in plotting Fig. 2 can be easily taken into account without replotting these curves. Put

$$N' = fN$$

then

$$\log N' = \log f + \log N$$

and a length equal to $\log f/(n + 2.5)$ added to the ordinate of the curve at any point will give the value of Y for $N' = fN$. If f is less than unity the length is to be subtracted. For example, for the point P of Fig. 2, $Q = 2500$, $A = 450$, $v = 5.56$. If $f = 1.5$, then P_1 is the point where $PP_1 = \dfrac{\log 1.5}{n + 2.5} = \dfrac{\log 1.5}{3.25}$. If $f = 1/1.5$, P_2 is the point. These values are given in Table 3.

32 The usefulness of this analysis is limited by the accuracy of the determination of n, and this in turn depends upon the definite knowledge of construction costs.

No. 1709

MEETINGS SEPTEMBER-DECEMBER

MEETINGS OF SECTIONS

ATLANTA

October 28: Smoke Conditions in Atlanta, Cecil P. Poole and Department of Public Works, Earl F. Scott.

November 24: A Broader Field for the Engineer, Dean D. S. Kimball.

December 16: Informal dinner meeting with Mr. C. W. Rice.

BALTIMORE

November 13: Treatment of Wastes in the Canning Industry, Dr. J. H. Shrader, and Some Problems in the Canning Industry, J. C. Talliaferro.

BIRMINGHAM

October 11: Excursion to the Fairfield Works of the Tennessee Coal, Iron & Railroad Co.

November 22: A Broader Field for the Engineer, Dean D. S. Kimball.

BOSTON

November 19: Excursion to Hood Rubber Co. followed by dinner at Harvard Club. Addresses: Employee and Service Department of Hood Rubber Co., by Dr. R. S. Quinby, and Rubber from the Crude to Manufactured State, by Alfred A. Glidden (illustrated). Discussion: Shall the Engineer be Licensed?

BUFFALO

October 6: Fundamental Americanism, S. A. Botsford, and Our Country's Call, M. E. Cooley.

October 14: Manufactured Weather, J. E. Bolling.

November 18: Peace Memorial Bridge, Messrs. Eckert and Bagley.

CHICAGO

September 30: City Zoning as it pertains to the requirement for residential and manufacturing districts, by Mr. Whiten, Chairman, Zoning Committee, Cleveland, O.

October 21: Political Problems.

October 29: Power Supply of the Future.

November 21: Successful Method of Improving Human Relations, Chas. V. Scribner, Results of an Americanization Program, Alex. D. Bailey. Discussion: A National Department of Public Works.

November 24: (Joint meeting A. I. E. E., I. E. S., and W. S. E. on Industrial Economics.) Personal Efficiency of the Employee, Harold Elmert. Electrical Power as a Factor in Effecting Economies and Increasing Production, Geo. H. Jones. Reaction of Labor to Intensive Lighting, Edwin W. Tillson.

CINCINNATI

October 20: (With Engineers' Club of Cincinnati). The Engineers' Place in Labor, Dr. Ira N. Hollis.

CLEVELAND

September 9: All day meeting. The Trackless Train for Moving Materials (Illustrated with motion pictures). Engineering Problems of Cleveland's Rapid Transit Development. Inspection trip by sepcial train, the first over the New Rapid Transit Line, stopping at the Shaker Heights Club for luncheon. Dinner at Hotel Cleveland. Noted Hydro-Electric Developments in Italy and South America (Illustrated) by O. M. Smart, specialist in hydro-electric engineering, detailed by the U. S. Government to investigate the recent water power progress made in the Italian Alps. Pertinent address on a phase of the problem of capital and labor by Dr. Willis A. Moore.

September 30: The Engineer's Place in Safety Work, L. A. DeBlois, Engineer of the E. I. Du Pont deNemours Company, Wilmington.

November 25: (With Cleveland Engineering Society.) The Open Court Trial of Patent Cases, Hon. W. L. Day.

COLORADO

September 13: Informal dinner to Dean Charles Russ Richards.

October 24: Dinner and addresses. Building Marine Engines and Ship Machinery One Mile Above Sea Level, Wm. Lester, and Government's Preparation for Educating Disabled Soldiers, Professor J. A. Hunter.

November 28: Industrial Education in Denver Public Schools, Jos. Y. Parce. Discussion on Industrial Democracy and Industrial Education.

December 26: Dinner meeting.

CONNECTICUT

November 19: The Motor Truck in War, Col. F. H. Pope and Lt. Col. A. J. Slade. Motorization of the World's Traffic, K. G. Martin.

Hartford Branch

October 20: Engineering Problems Involved in Distribution of Water in the City of Hartford, Caleb M. Saville.

November 24: Precision Gages, C. W. Weingar.

Meriden Branch

November 7: Conflicting Claims on Piston Rings, Maurice A. Michaels, and The Manufacture of Shot Guns, Walter A. King.

November 21: Application of the Vacuum Tube to Radio Telegraphy, Harold P. Donle.

December 19: Manufacture of Phonograph Records, Fred L. Wood.

New Haven Branch

October 15: Joint meeting with the Winchester Engineering Club: address by F. O. Williams.

October 16: Industrial and Technical Points of the Rubber Industry, Charles R. Haynes.

EASTERN NEW YORK

October 27: Dinner, and talk by E. W. Rice, Jr.

INDIANAPOLIS

October 24: Industrial Unrest, W. C. Rogers; Improving Human Relations, in Industry, C. F. Scribner; Results of Americanization Program Carried Out by Commonwealth Edison Co., A. D. Bailey; Training of Students in Industrial Management, Prof. B. W. Benedict; Industrial Medicine, Dr. O. P. Geier.

October 25: General Consideration of Present Status of Research in Industrial Life of the Country, C. F. Hirshfeld; Administration of Industrial Research Affairs, R. E. Carpenter; Industrial Research, Walter A. Spear; Research in Nela Park Laboratory, E. P. Hyde; Present Status of Fatigue of Metals Phenomena, Professor H. F. Moore; Design of Purdue Tractor Testing Plant, C. J. Benjamin; Research, George S. Hessenbruch; and Pure Science and Engineering Research, Dean J. R. Allen.

LOS ANGELES

December 8: Address by A. H. Goldingham.

MID-CONTINENT

October 30: All-day meeting at Bartlesville, Okla. Report of Committee: What Should Be the Content of a Course to Fit Young Men to Become Petroleum Engineers? Addresses: Construction and Operation of Pipe Lines for Transportation of Natural Gas, C. E. Brock; A Standard for Gasoline, Kerosene and Motor Fuel Oil, Dr. E. DeBarr; Problems Confronting Engineering Colleges, Professor A. A. Potter; Effect of Compressed Air or Gas on the Production of Petroleum Wells, W. S. Smith; New Problems for Engineers, Dr. Ira N. Hollis; Natural Gas Gasoline Plants, F. E. Rice; and Appraisement of Oil and Gas Properties, O. J. Berend.

October 31: Inspection trip by automobile.

MINNESOTA

November 3: The Preparation and Use of Powdered Coal, Alonzo B. Kenyon.

December 1: (With Student Br. of Univ. Minn.) Problems Confronting Young Engineers, Dean L. W. Jones.

NEW YORK

September 17: Industrial Unrest, Dr. William M. Leiserson. Published in full in MECHANICAL ENGINEERING for November, 1919.

October 9: Steel Carvings, Geo. J. Foran and The Human Factor in the Operation of Industry, Dr. Henry R. Seager. Abstracts in MECHANICAL ENGINEERING for November 1919.

November 26: Position of the Engineer in Modern Industry, Charles Ferguson, and Management in Relation to Capital and Labor, E. W. Hulet.

ONTARIO

November 14: Dinner, followed by address by General C. H. Mitchell.

December 18: Thirty Years' Progress in Boiler Construction, N. Quesnal.

PHILADELPHIA

October 28: Powdered Coal and Its Future, H. C. Barnhurst.

November 25: Color Photography, Henry Hess. Informal talk by J. A. Steinmetz.

December 11: Oil as a Fuel, Henry Thomas and E. H. Peabody.

ST. LOUIS

October 15: Address by Dean D. S. Kimball.

October 16: Industrial Medicine, Dr. Otto P. Geier.

SAN FRANCISCO

October 23: Flow of Oil in Pipe Lines, Dr. W. F. Durand, and Oil Pipe Line Design and Economics, Herbert W. Grozier.

WASHINGTON

November 19: Joint Meeting under the auspices of the Society of Washington Engineers.

November 20: War Inventions, Col. C. H. Hilton, and War Inventions and their Application to Industry, F. C. Colwell. Discussion: Department of Public Works.

WASHINGTON STATE

November 24: Organization meeting, election of officers and adoption of constitution.

WORCESTER

September 30: Methods of Sewage Disposal (illustrated), Harrison P. Eddy.

October 28: Costs from Any Angle, Wm. R. Bassett.

November 25: National Department of Public Works, Dr. George F. Swain.

December 16: Power, Its Production and Distribution as Applied to Industrial Plants (illustrated), R. J. S. Pigott.

THE ANNUAL MEETING

Attendance at the Annual Meeting of the Society, held at the Engineering Societies' Building, New York, December 2–5, reached a record total of 2116, of which 1346 were members and 770 guests. Anticipating a large attendance, a Sub-Committee of the Committee on Meetings and Program, composed of local members having the social arrangements in charge, prepared to utilize all the available space in the building. Besides the Society's headquarters on the eleventh floor, the meeting rooms on the fifth and sixth floors and the auditorium were made use of, and on Thursday evening, the annual reunion night, members also gathered in the spacious foyer on the first floor.

Although the convention nominally opened on Tuesday, December 2, many were present earlier for Council and committee meetings, Local Sections conferences, etc., twenty-eight delegates being in attendance from the Local Sections of the Society.

The formal opening was on Tuesday evening, when President Cooley gave his presidential address on the activities of the Society and national scope of the work of the engineer. Following his address the announcement was made of the election of officers for the ensuing year. President-Elect Fred J. Miller was introduced by President Cooley and responded with a speech of acceptance.

Honorary membership was then conferred upon Charles de Freminville, Consulting Engineer, Creusot Works, France, who was present and delivered an address of acceptance, and announcement was made of the election to honorary membership of Auguste C. E. Rateau, Chairman, Board of Directors, Rateau, Battu and Smoot Company, France. At the conclusion of the exercises there was held a reception by the Society to the President, President-Elect, ladies, members and guests.

During the Business Meeting a tablet was unveiled of the late Frederic R. Hutton, for twenty-three years Secretary of the Society. This tablet, an engraving of which is reproduced herewith, was executed by Victor Brenner and now hangs in the reception hall of the Society headquarters. The tablet was presented with remarks by Major William H. Wiley, Treasurer of the Society, a close friend of

MEMORIAL PLAQUE TO THE LATE FREDERIC R. HUTTON

Professor Hutton, and Acting Chairman of the Committee appointed for the preparation of this memorial. Major Wiley spoke as follows:

GENTLEMEN OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS:

In presenting this tablet, as Acting Chairman of the Memorial Committee appointed for that purpose, I was firmly of the belief that no words of mine would be fitting on such an occasion in speaking of a man so well known to the entire Society and whose work, as its Secretary and President, was a large part of the history of this organization, but the rest of the Committee unanimously differed from my judgment, hence I will yield to them and make a few brief remarks in tendering this memorial.

When Dr. Hutton became Secretary of the Society in 1883 the membership was 554. I was elected Treasurer in 1884 and the amount of money passing through my hands that year was, in round numbers, $6,500. With his usual energy shown in every enterprise with which he was connected, Dr. Hutton labored in season and out of season to increase the membership of the Society by every means in his power, and used wonderful tact in all such action. Naturally a wonderful success resulted from his efforts.

The Society showed its appreciation of him and of his work by electing him to the presidency in 1908, at which time the membership was 2,929 and the amount of money passing through my hands, as Treasurer, was nearly $54,000. I think I shall not say too much when I assert that a great part of the increase in membership was due to his untiring efforts. At the present time, the year ending September 30, 1919, the membership has increased to 11,082 and the money passing through my hands, as Treasurer, $375,430. I mention these figures to show that the growth fostered by Dr. Hutton was maintained by his successor.

My associations with Dr. Hutton, as Treasurer, were naturally intimate, and most pleasant. His energy is known to all of us, and as his neighbor in the Catskill Mountains for over twenty years, I can say that he threw his efforts in that section to building up and increasing the activities of the Union Chapel with which he was connected. He was busy at all times, and devoted a great deal of valuable time to this work, and, like his actions for the interests of the Society of Mechanical Engineers, his efforts in the Mountains were marked by signal success. I could dwell at great length on the work of Dr. Hutton, both professionally and socially. He has been sadly missed in this Society as well as in Twilight Park, his summer home. Notwithstanding he went to the Mountains for a well-deserved rest, yet when an appeal for any charitable object reached him, he did not hesitate to volunteer to aid as a reader or a speaker or a contributor. I am just glancing at these matters in passing to pay a deserved tribute to one who did so much for others, but the speakers who are to follow me will go more into the details of his work.

I cannot close more fittingly than to apply to our late Secretary and President, the words inscribed on the tomb of a celebrated architect who was buried near one of the finest structures he had ever erected — "If you wish to see his work, look around you."

There was widespread interest during the convention in the discussion upon the reports of the Aims and Organization Committee of the Society and the Joint Conference Committee of the Founder Societies, and in the conference of the Local Sections' delegates.

Numerous committee meetings were also held, including those of the A.S.M.E. Power Test Codes and its Sub-Committee on General Instructions; Research Committee and its Sub-Committee on Fluid Meters; the Committee on Steel Roller Chains; the Committee on the Standardization of Shafting; and the Standardization Committee.

On the social side mention should be made in particular of the gathering at the ladies' reception and tea on Thursday afternoon, always one of the pleasantest occasions of the Annual Meeting.

The excursions to points of interest about the city were well attended, particularly that on Friday to the Curtiss aeroplane plant and flying field at Garden City where exhibition flights were given and special arrangements were made whereby members who so desired were able to undertake flights. As usual the meeting of representatives of student branches was held and following the convention on Friday evening there were numerous college reunions. An event of interest to many was an evening devoted to an account of the work of John Ericsson and Cornelius H. DeLamater, the founder of the famous DeLamater Iron Works, the occasion being the eightieth anniversary of the meeting of these two noted engineers and the thirtieth anniversary of their death. Several societies joined in these exercises.

The Sub-Committee for the 1919 Annual Meeting consisted of R. V. Wright, General Chairman, and R. F. Jacobus, Frederick A. Scheffler, E. Van Winkle, F. T. Chapman, Hazen G. Tyler, H. J. Marks, Edric B. Smith and Mrs. H. C. Spaulding, chairmen in charge of branches of the Sub-Committee's activities.

PROGRAM

Tuesday Morning, December 2

Registration of members and guests at headquarters. Meetings of Council and Society's Committees.

Monday Evening

PRESIDENTIAL ADDRESS AND RECEPTION

Presidential address, M. E. Cooley, President of the Society. Report of Tellers of Election. Introduction of President-Elect. Conferring of Honorary Membership upon Charles de Freminville, Consulting Engineer, Creusot Works, France, and announcement of election to Honorary Membership of Auguste C. E. Rateau, Chairman, Board of Directors, Rateau, Battu and Smoot Company, France.

Reception by the Society to the President, President-Elect, ladies, members and guests. Dancing.

Wednesday Morning, December 3

BUSINESS MEETING

Discussion on Amendments to Constitution. Reports of Standing and Special Committees. Awards of Junior and Student Prizes. Memorial to Prof. Frederic R. Hutton and William Kent. Exercises in Honor of Members who Served in the War.

Reports of Aims and Organization Committee and of the Joint Conference Committee of the Founder Societies.

Wednesday Afternoon

SIMULTANEOUS SESSIONS

APPRAISAL AND VALUATION

Joint Session with the American Society of Refrigerating Engineers.

Ice Plant Depreciation, George E. Wells.

Depreciation of Insulation, John E. Starr.

Appraisal and Valuation Methods, David H. Ray.

Fundamental Principles of Rational Valuation, James Rowland Bibbins.

Development of Project, by the late F. B. H. Paine.

The Construction Period, H. C. Anderson.

Price Levels and Value, Cecil Elmes.

GAS POWER SESSION

The Hvid Engine and its Relation to the Fuel Problem, E. B. Blakely.

Kerosene as a Fuel for High-Speed Engines, L. F. Seaton.

Combustion of Heavier Fuels in Engines of Constant-Volume Type and Super-Induction Type Engines, Leon Cammen.

Oil Pipe Lines, S. A. Sulentic.

Wednesday Evening

De Lamater-Ericsson Commemoration.

Thursday Morning, December 4

THE INDUSTRIAL SITUATION IN RELATION TO PRESENT CONDITIONS

The Causes of Industrial Unrest; and the Remedy, Frederick P. Fish, Chairman, National Industrial Conference Board.

Wage Payment, A. L. De Leeuw.

Systems for Mutual Control of Industry, William L. Leiserson, formerly Chief, Division of Labor Administration, Working Conditions Service, U. S. Department of Labor.

What may we Expect of Profit Sharing in Industry? Ralph E. Heilman, Dean, School of Commerce, Northwestern University, Chicago.

Thursday Afternoon

SIMULTANEOUS SESSIONS

MACHINE DESIGN SESSION

Reliability of Materials and Mechanism of Fractures, Charles de Freminville.

Tests on Dredging Pumps used in Inner Harbor Navigation Canal, New Orleans, W. J. White.

A Prefected High-Pressure Rotary Compressor, C. B. Lord.

Turbo-Compressor Calculations, A. H. Blaisdell.

A New Type of Hydraulic-Turbine Runner, Forrest Nagler.

POWER MACHINERY SESSION

Emergency Fleet Corporation Water-Tube Boilers for Wood Ships, F. W. Dean and Henry Kreisinger.

Flow of Water through Condenser Tubes, William L. DeBaufre and Milton C. Stuart.

Air Pumps for Condensing Equipment, Frank R. Wheeler.

Thermal Conductivity of Insulating and other Materials, T. S. Taylor.

TEXTILE SESSION

The Economical Use of Steam in Textile Plants, George H. Perkins.

Topical Discussion on Teaching and Training Textile-Mill Employees.

Thursday Evening

Lecture on The Future of Aviation, by Colonel E. A. Deeds, Dayton, Ohio.

Lecture on Present Development of the Military Airplane, by Colonel Thurman H. Bane, Chief of Engineering Division of Air Service.

Friday Morning, December 5

SIMULTANEOUS SESSIONS

TRANSPORTATION SESSION

Scientific Development of the Steam Locomotive, John E. Muhlfeld.

Motor-Transport Vehicles for the United States Army, John Younger.

MACHINE SHOP SESSION

Common Errors in Designing and Machining Bearings, C. H. Bierbaum.

Lubrication of Ball Bearings, H. R. Trotter.

Thread Forms for Worms and Hobs, B. F. Waterman.

Illustrated Talk on Electric Arc Welding, H. L. Unland.

GENERAL SESSION

SLOW-SPEED AND OTHER TESTS OF KINGSBURY THRUST BEARINGS, H. A. S. Howarth.

OCTAVAL NOTATION AND THE MEASUREMENT OF BINARY INCH FRACTIONS, Alfred Watkins.

MODERN ELECTRIC FURNACE PRACTICE AS RELATED TO FOUNDRIES IN PARTICULAR, W. E. Moore.

AN INVESTIGATION OF STRAINS IN THE ROLLING OF METAL, Alfred Musso.

Friday Afternoon

COUNCIL MEETING.

No. 1710

A SURVEY OF THE SOCIETY'S ORGANIZATION

PRESIDENTIAL ADDRESS 1919

By Mortimer E. Cooley, Ann Arbor, Mich.
President of the Society

IT is my pleasant duty on this occasion to tell you something of the doings of the Society during the year while I have been its President. And I thought you would also be interested to hear something about the way in which the Society does its work. For most of you, I feel certain my story will be like the opening chapters of a new book. It will, in fact, be from the opening chapters of an old book, familiar to you in appearance but, I dare say, one you have scarcely ever opened. I mean the Year Book of the Society.

First, however, let me tell you how much I have enjoyed the year and how much I appreciate the honor you conferred on me when you entrusted me with the responsibilties of administering the affairs of the Society. It is truly a great responsibility and growing greater with each year's growth of the Society — growth both in its membership and in its activities.

I remember well those first years, back in the early eighties when our members were only a few hundred, and our activities confined principally to two meetings, one in the spring, the other in the winter. The contributions were chiefly the papers read at the two meetings and the discussion they provoked. Today there are over 12,000 members, 72 committees engaged in the work of the Society, 17 committees representing the Society or other organizations, 31 local sections, and 47 student branches. These are directed by a Council of 22 members assisted by the chairmen of the six Standing Committees of Administration and a paid Secretary with 76 assistants. Thus has the Society grown in the 39 years since it was organized in 1880.

The machinery of this great national society is really somewhat complicated. I assume that you are as ignorant of it as I was when

An address delivered at the Annual Meeting 1919, of The American Society of Mechanical Engineers.

I took my place at the throttle valve. Ah! happy thought! How easy to explain when the right analogy comes to one. As you know a throttle valve has two functions, one to start, the other to stop. At first I thought its most important function had to do with starting. But then I was guileless. I did not know my colleagues on the Council. Throttle valve is just the right word. How many times we would in our discussions have settled the affairs of the Nation had something gone wrong with the stopping function of that throttle valve; but as it happened the valve never got out of order; only individuals got "out of order."

Probably only a few of you know what the Council is. I need not hesitate to speak openly and with freedom, as knowing me as well as they do, it would not have surprised me had there been not a single member of the Council present here this evening. The fact that a number are present is evidence of the discipline to which they have been subjected; or it may be evidence of the possession of a conscience still active in pointing the way of duty. But let that pass. The Council is composed of the President, six Vice-Presidents, nine Managers, the Treasurer, and five Past-Presidents. They are a fairly quiet and orderly, not to say, modest lot of men — starting with the President at the top. But those past-presidents at the bottom! What shall I say of those past-presidents? And to think that I, too, shall so soon graduate and become one of them. But so runs the machinery of the Society. The last five presidents have a place on the Council. Thus each president has five years in which to make amends for any bad judgment he may have exercised during his one year of being first in responsibility. While these 22 men do all the voting, they do not do all the talking. There are the six chairmen of Standing Committees to be reckoned with on the floor in debate. That makes 28. But that is not all — far from it. There is also the Secretary. He is a spouting fountain — of ideas for advancing the interests of the Society, and the watchdog of its activities. No one of you knows how great is his responsibility to keep the President functioning properly. The Treasurer of the Society is the only member of the Council who sits tight and says nothing.

Now that I have paid my respects to these gentlemen, my most immediate colleagues, let me say of them that it has been a rare privilege to be associated with such men in the great work of the Society. It is no mere honor as some have thought, to be a member of the Council. It means hard and conscientious work and vast quantities of it. If you could see these gentlemen sitting long hours around the Council Board, sometimes two full days at a time, you would realize what good servants of the Society they are. And let me tell you that were it more generally known how like horses these men have to work there would be fewer candidates, when the nominations for officers of the Society are made. The Council has held in all, during the past year, eleven meetings, one of them in Pittsburgh, two in Detroit and one in Indianapolis.

To my colleagues of the Executive Committee I would pay especial tribute. They have been ever faithful in responding to calls for some emergency action. And many times I fear they have thereby been greatly inconvenienced. I was told I must choose nearby men in order to get them together, and even then I would rarely have a quorum. But I wanted to go afield a bit in my selections for the Executive Committee and bethought me of a way to break down tradition. So I invited the Executive Committee to meet me at dinner. It worked like a charm. Not a man was absent. And at none of the many meetings we have held has there ever lacked a quorum. My heart goes out in gratitude to those loyal men who have so wholeheartedly helped the President over so many difficult places.

Next in our administration machinery come the six Standing Committees of Administration — Finance, Meetings and Program, Publication and Papers, Membership, Constitution and By-Laws, and Local Sections. I do not quite know how to tell you of my appreciation of the thirty men on these six committees. The amount of time and energy they give to the Society's work is enormous — appalling I nearly said, but that word refers to the President who must keep in touch with their several activities. The Society should know and appreciate the great care required and given so splendidly by the groups of five gentlemen composing each of our six standing committees of administration.

To aid in its work, the Meetings and Program Committee is assisted by six sub-committees, each charged with the responsibilities of securing papers and preparing programs on particular subjects for sessions at the Annual and Spring Meetings of the Society. These sub-committees are really special committees in that they may be changed from time to time to meet the need of new subject matter for presentation at the Society's meetings.

Next come the Standing Committees — House, Library, Public Relations, Research and Standardization. The Research Committee supervises all research activities, collaborates with kindred committees of other institutions and obtains results of researches conducted in other countries. It has at present seven sub-committees which are really special committees, each engaged in the investigation of some particular subject, such as the cutting action of machine tools, fuel oils, lubrication, fluid meters, worm gearing, bearing metals, and heat transmission. The latter acts jointly with a similar committee of the American Society of Refrigerating Engineers. The report of the Research Committee by its Chairman, Professor Arthur M. Greene, Jr., read at the Detroit meeting, merits wide attention. It deals with the subject comprehensively.

The Society has naturally a large number of special committees to deal with particular problems. These are formed as occasion demands and discharged with the acceptance of their reports. Some of them are appointed annually and thus become standing committees for the year; such for examples as the Committee on Junior and Student Prizes. Others are more in the nature of permanent standing committees, such as the Committee on Increase of Membership and on Student Branches. The latter is now called Committee on Relations with Colleges.

During the past year this Committee on Relations with Colleges has, under the direction of its Acting Chairman, Past President Ira N. Hollis, made a comprehensive study of the Society's relations with educational institutions and of student and junior prizes. The report of this committee is very complete and well worth attention. It is the result of much hard work involving extended correspondence and study of other society organizations. The Council in accepting this report put itself on record as favoring prizes other than honorary membership, now accorded to persons of acknowledged professional eminence. Life membership, in addition to being acquired for a sum of money, could be awarded for the best contribution to mechanical engineering found in the papers for one year; Junior and Student Prizes, for papers of exceptional merit; a medal, for notable invention or some striking improvement in connection with the industries; honorable or special mention, for conspicuous contributions to engineering either of a practical nature or in literature; scholarships or fellowships, for exceptional attainments in college work; a medal or special mention, for exceptional achievement by Junior and Student members.

Rear Admiral George W. Melville left a bequest of $1,000 to the Society for the award annually of a medal for the best thesis on mechanical engineering; and another gentleman has this year made a similar contribution. The Society allowed him $1500 for expenses to France as the Society's official representative at the Franco-American Congress but he paid his own expenses and added of his own money $1000, making his contribution $2500. I refer to Past-President Charles T. Main.

The National Societies can well afford to further their relations with colleges. It is impossible to do too much toward helping to shape the ideals and ambitions of the youth of our country who are soon to take our places in the engineering world. The colleges would welcome the counsel of men in actual contact with the world's affairs. These engineers are giving little attention, or none at all, to the preliminary training of the young men they take on their staffs as assistants. They are, or should be, vitally interested; and they could if they would, be of great help in shaping the curricula of engineering colleges. Whatever the practicing engineer may think of his brother the professor-engineer — and his thoughts are, I fear, sometimes none too flattering — he must not confuse his brothers with clairvoyants. While college professors may be conceded to have wonderful imaginations, they stop short of being occult. The advice of the practicing engineer would be eagerly welcomed. If he thinks of us of the near-cloth as being diviners, let him at least show his hand to be read.

It cannot be said in these days that the college man works for money. His real compensation comes from the success of his students after they leave college and enter the practical world. Their training must still, and for several years, be in progress. Their teachers only are different. If we are to achieve greatest success in our profession of engineering, that is, have it stand at the forefront in world affairs, you to whom are turned over the young men from our colleges must do your part. When you realize what your part is, you will be both willing and anxious to help the college man in his part of the work. The four years in college should not be separated, as by a wall, from the years which follow. There should in those after years be the same inspiration to high ideals of life — those quite apart from money. Our country — the whole world, never needed high ideals more than it does today.

But let me get on with my description of the Society's machinery. The Boiler Code Committee with its five sub-committees, still occupies a front seat in the Society's activities. The fact that our Boiler Code has been incorporated in the laws of thirteen states and seven municipalities, is a source of great satisfaction and pride. It indicates a good work, well done. The Boiler Code sub-committee on Railway Locomotive Boilers is very active. The railroads themselves, through the Interstate Commerce Commission, have now sought the opportunity to coöperate in this work.

The Committee on Power Test Codes, reorganized in 1918 with 18 sub-committees, is a bee-hive of industry, and bids fair to rival the Boiler Code Committee for the Society's favor. Some of its sub-committees have already practically completed their work and submitted reports. Another year should get the important work of this committee well on towards completion.

Progress in various degrees is being made by other special committees. The Committees on Standardization of Feedwater Heaters, Shafting, Steel Roller Chains, have all reported progress. The Committee on Protection of Industrial Workers has recently completed jointly with the United States Bureau of Standards, an elevator code in which the Safety Committee of the National Association of Manufacturers participated. The special Screw Threads Committee and the Committee on Flanges and Pipe Fittings have been active. An international convention was recently called to meet in Paris to adopt a standard for pipe threads. This Society's official delegate is M. Laurence V. Benet, who resides in Paris; and to assist M. Benet technical advisors have been appointed from the Society's membership in various American industries engaged in the manufacture of pipe and fittings.

It will interest you to learn that standardization in general has recently taken a distinct step forward. The American Engineering Standards Committee is now, after being for several months the subject of active debate, an accomplished fact. While it is recognized that there are still several changes that could well be incorporated in its Constitution, it was considered better to get started and let experience determine the exact nature of those changes. The Committee starts off with representatives from the American Society of Civil Engineers, the American Institute of Mining and Metallurgical Engineers, The American Society of Mechanical Engineers, the American Institute of Electrical Engineers, the American Society for Testing Materials, and from the Government Departments of War, Navy and Commerce. The constitution provides for admitting to membership on the Committee, representatives from other societies and associations. This Committee does not create standards. That work is left to be done as now, by bodies especially interested in some particular standard. But the Committee gives its stamp of approval and serves as the medium through which standards are passed out to the world. One of the most useful functions of the American Engineering Standards Committee will be to coöperate with similar bodies in other countries in the formation of international standards; for instance, with the industries of Great Britain through the British Engineering Standards Association. When this Committee is functioning completely, such matters as the coming meeting in Paris, of the Commission on an International Standard for Pipe Threads, would be handled by the American Engine ring Standards Committee.

It will be remembered that last year at the Annual Meeting, President Main, after delivering his address, fled the country. I did not at first know the reason for his precipitate haste. I could scarcely imagine that it was because of the things he said in his address, although I now know that the President can say and do things amply justifying him in seeking safety. It was, however, an honorable mission that called him away. He went abroad as this Society's representative on a Committee of the four Founder Societies which was invited by the Société des Ingeneurs Civils and the Committee of the French Engineers Congress, with the official approval of the French Minister of Armament, Public Works and Commerce, to study with French Engineers the rehabilitation of France after the war. Out of the service rendered by this Committee has come about the appointment of a Franco-American Committee in which this Society is represented by Charles T. Main and George W. Fuller.

Other exchanges of international courtesies that have afforded us satisfaction and pleasure during the year occurred in connection with the Ottawa meeting of the Engineering Institute of Canada in February and the Detroit meeting of this Society in June. Past-President Hollis went to Ottawa as our special representative and Fraser S. Keith, Secretary of the Engineering Institute of Canada, came to Detroit as special representative of the Institute.

The James Watt Centenary celebration was held in England on September 16th, 1919. In response to an invitation to participate, this Society sent as honorary Vice-Presidents Messrs. J. Wilfrid Harris, Harry F. L. Orcutt, R. Sanford Riley, and Wilson E. Symonds.

This Society has had the pleasure of participating in extending courtesies to members of several delegations from foreign countries who have come to this country on various missions. Official delegations have come from France, Belgium and Switzerland. In fact, our distinguished guest of the evening, M. de Freminville, is a member of the French Engineering and Economic Mission which was invited to this country by the Chamber of Commerce of the United States. M. Schneider, the illustrious head of this Commission, was recently entertained by our sister society, the Mining and Metallurgical Society of America.

An interesting and important part of the mechanism of the Society's machine is the groups of members, known as Society Representatives, who represent this Society on other bodies. There are 17 of these groups, and the men who compose them are those who have been conspicuous in the Society's work, or have qualifications especially fitting them for the particular duties involved. These other bodies and committees are the American Association for the Advancement of Science, American Engineering Standards Committee, Classification of Technical Literature Committee, Cost of Electric Power Committee, Engineering Council, Engineering Foundation Board, International Committee on Screw Threads, International Standard for Pipe Threads, John Fritz Medal Board, Joseph A. Holmes Memorial Board, National Research Council, Naval Consulting Board of the United States, Standards for Graphic Presentation, United Engineering Society — its Board of Trustees and its Library Board, United States Bureau of Mines Advisory Committee, Western Society of Engineers, Washington Award Committee. The names themselves sufficiently describe the character of work involved making unnecessary any detailed description. The reorganization of the National Research Council on a peace basis includes its affiliation with similar bodies of other nations. The Chairman of our own Committee on Research has a place on the National Research Council.

The Regular Nominating Committee and the Tellers of Election must also be mentioned. The Nominating Committee was in the old days chosen by the President alone. But President Jacobus in 1917 inaugurated the policy, followed since, of inviting the Local Sections to make nominations from which the President could choose. But a new plan is to be followed in the future.

The Nominating Committee is to be elected by the voting membership: and tomorrow morning, the Society will decide on the machinery. The proposal is to utilize the Local Sections organization for conducting the primary for nominations, every voting member being assigned to a local section for that purpose. The nominations are to be confirmed at the Local Sections Conference to be held at the Annual Meeting, after which the election will take place at the Annual Business Meeting. Thus will be insured, it is believed, a thoroughly democratic election.

In 1918 there were twenty-one Local Sections. Now there are thirty-one, an increase of ten during the year. Three of them are on the Pacific Coast. One local section in New England has five branches. This growth of Local Sections and extension of section responsibilites is one of the marked features of the Society's development. Like a wide spreading tree, the Society must have its roots and branches wide spread if it is to grow and thrive. The roots must reach out into new earth and the branches out into sunshine. It is my firm belief that by increasing the importance of its sections the Society will add to its strength and vigor and render the greatest service to its membership.

As I see future development of the profession as a whole, it is important that the members of local sections of our own and other national societies should be also members of sectional and local engineering societies which embrace in their membership all engineers whether they be members of national societies or not; and which include those who work with engineers as members of their staffs, and who may not possess the qualifications for membership in the national societies. This, happily, is the plan being followed. Cleveland is a good example. There is now an arrangement, approved for trial by our Council, whereby our members who are also members of the Cleveland Engineering Society may have part of their dues to this Society remitted; that is, there is a division of dues. No doubt, considerable latitude will be permitted Local Sections in order to encourage them to work out plans for improving their own organizations. With increase in importance of Local Sections work, there will come the necessity of providing more permanent local headquarters. It has been suggested and I believe wisely, that financial assistance should be given to local building projects through the aid of societies owning the Engineering Building here in New York, and, by loans from surplus reserves. This subject may well have our earnest consideration.

One of the Sections Committee's recommendations is a Travelling Secretary whose duty it shall be to visit Local Sections and assist them in developing their plans. It is a good suggestion; but I would add to it. The President's duties have become very heavy. Notwithstanding that I have given one-quarter or one-third of my time to the duties of the office, I am sure that I have not done more than one-half of what the President should do. Among other things, it has been impossible to respond to the many cordial invitations to visit local sections. It has also been impossible to respond to many invitations from other national bodies to be present and participate in their deliberations and ceremonial functions. You will agree with me that these are proper and important duties of the President, and the dignity of the Society requires that they should be performed.

What I would add, therefore, is that the proposed travelling secretary be the President's representative. He could be styled "Assistant to the President." He should be a man of parts and worth a salary of at least $5,000 per year. With such assistance the President could perform many of his duties in whatever part of the country he might come from; that is, it would not be necessary for him to spend so much of his time at national headquarters.

This assistant could go to the President's home and stay a week or two at a time if necessary. He could represent the President at Local Section headquarters; or by lightening the burden of routine work, enable the President to go. Nor is this all. Many of the President's duties now necessarily fall on our Secretary, already overburdened with his own duties. The assistant to the President could relieve the Secretary of many of these, but not all of course. Certain very important duties could be performed by no one with more grace or with more complete regard for the proprieties and requirements, than by our Secretary.

In taking up the duties of President, it became evident to me after a time that pending the report of our Committee on Aims and Organization, it would be inexpedient to suggest or inaugurate any changes in the existing policies of the Society. It seemed rather a period during which the existing activities of the Society should be fostered and made as fruitful of results as possible. With this thought in mind, I undertook a study of committee work. As was to be expected with such a large number of committees, some of them were more active than others. Inactivity was not necessarily the fault of a committee. Some of the sub-committees were required to work only when called on. The work of other committees was temporarily suspended on account of the war. In a number of instances suspended committee work has been resumed.

One result of my investigation was to find frequently the same man serving on a number of committees, and only the older and more experienced members of the Society on many of the committees. While in some cases these conditions were warranted as tending to give the best results, it seemed to me that for the good of the Society it was desirable to increase the number of individuals engaged on committee work by limiting one's membership to a few committees and by adding some of the younger men. In this way more general interest would be stimulated in the Society's work and the younger men would be trained for more responsibility later. Accordingly, Local Section officers were requested to canvass their membership for men willing to accept committee appointments. There is now on file a considerable list of men not only willing to serve but who have promised if appointed to attend to the work.

Another result of my investigation was to find the membership of many of the committees confined to within reaching distance of national headquarters here in New York. It is absolutely necessary, in a number of cases, to have at least a majority membership living nearby. The Executive Committee, the Finance, the House, the Library and the Membership Committees are examples. It is not advisable, in general, to have members of a committee come from widely separated parts of the country. It should be possible for them to get together without too great sacrifice of time consumed in travel, and without too great expense, which in most cases must be borne by the members personally.

It appeared that a solution of the difficulty might be found in choosing committee members with respect to localities. For instance, an automobile committee might be centered in Detroit and embrace Buffalo, Cleveland, Toledo, Jackson and Port Huron. A petroleum committee might be wholly from the Oklahoma or the California oil districts. A machine-tool committee might be wholly from the Ohio or the New England district, and similarly for committees on other subjects. A natural way with the increased importance of our local sections would be to choose the membership committee wherever practicable from the members of a particular local section. The local section itself could be held responsible for the work of the committee. The membership of such committees need not be confined to this Society. Indeed, it might be well, and in the interests of the Society as well as the work, to appoint some non-members. If they become greatly interested they might become members later. Committee work is of first importance in the Society, and should be well cared for. When it is of high order great credit comes to the members of the Committee and to the Society. It is the good work done by its members that enables the Society to occupy and hold its high place in the world. The honor of the Society is the honor of its members. The Society can give back only what its members put in both in kind and in quantity.

Of equal importance to that of Local Sections, if not greater, is the Society's publications. It is through them that the Society gives back to its members what they contribute in money and in work. Its journal has been rechristened and is now called MECHANICAL ENGINEERING. It is the medium through which the membership at large can keep in touch with all Society activities, and become promptly informed on important engineering topics discussed at the annual and spring meetings and at the Sections' meetings throughout the country. During the year the Society has acquired the ownership of "Engineering Index" which it has greatly enlarged, and in which now appear regularly reviews or titles of articles from over eleven hundred periodicals properly classified and indexed. The Council has recently authorized an increase in the scope of the journal so that it can be of still greater service to the membership.

With the desire to take part in and help solve, so far as it could properly, the industrial troubles of the country, the Council early in the year directed the appointment of a committee of seven to investigate and report on the feasibility of the Society actively participating in industrial questions. This committee made its report in February, advising that inasmuch as the constitution of the Society set forth as its object "to promote the Arts and Sciences connected with Engineering and Mechanical Construction" it was inexpedient to take any active part as a society in industrial matters; but that it was highly important that the members as individuals or in their business capacity, should take such part as they could.

Believing that it would be helpful to the membership to receive the results of investigations of industrial conditions the Council in April authorized affiliation with the National Industrial Conference Board and appointed a representative on the Board. The propriety of the Society holding such relations being later questioned the Council in October directed the appointment of a committee to report on the question of retaining our membership. This Committee, while unanimously commending the work of the National Industrial Conference Board was of the opinion that the

Society should not hold membership in any other society or organization. The Council, therefore, withdrew from the National Industrial Conference Board at its November meeting held on December 1.

By far the most important work of the Society during the year has been that of its Committee on Aims and Organization. Inasmuch as the report of this Committee has been sent out to the membership and is to be discussed and disposed of at this annual meeting, I have not deemed it necessary, or even proper to do more than call your attention to it and to urge you to take part in its consideration when presented. Suffice to say that certain recommendations of the committee have already had the attention of the Council since they were favorably considered by the Society at its spring meeting in Detroit. I refer particularly to the development of Local Sections, the encouragement of Professional Sections and the enlargement of the scope of the Society's journal, MECHANICAL ENGINEERING. We are all most appreciative of the work of this Committee. Theirs was a hard task, and while they would have welcomed an opportunity to carry their work further and perfect it, the Council thought that the Society would do well if it agreed upon and adopted even a few of the large number of recommendations made.

A most important part of the work of our Committee on Aims and Organization was its coöperation with similar committees of the American Society of Civil Engineers, the American Institute of Mining and Metallurgical Engineers, and the American Institute of Electrical Engineers. A joint committee of these four founder societies has held several meetings to consider and agree upon certain fundamental things as the basis for closer coöperation in the future. Without in any way sacrificing their identity and purpose, and leaving the chief aim of each to be pursued independently, it has been agreed that a comprehensive national organization or council of engineers should be created to be composed of representatives of local, state and regional societies. Further, that this organization be so framed as to include allied technical societies; and that coöperation of these several bodies in technical activities, and affiliations of local organization for general public service and welfare work be encouraged. This joint committee also recommends that the bill to create a National Department of Public Works receive the active support of all engineers.

Every thoughtful engineer will recognize the importance of these recommendations. The engineer has in the past been too much of an individualist. He is at heart an idealist. He lives in the things he creates, and often his chief reward is the satisfaction he enjoys from work well done rather than pecuniary. It is a splendid quality of heart and mind that enshrines the engineer. It harks back to those earlier days when respect for one's achievements and personal character measured success rather than the dollars amassed. But to keep one's place in these modern days requires that the game be played in accordance with prevailing rules. Thus the engineer must perforce conform to these rules if he would exert his fullest influence. It is organized rather than individual effort that gets ahead today. Only the individual who can direct and control organized effort stands out as an individual. He is the general in command of forces.

The one criticism of the engineering profession, as I see it, is that there are too few generals. The positions of high command too often go to others who are rarely themselves engineers but who constantly make use of engineers in framing and executing their plans. That is to say, the engineer, while he has the knowledge and skill required to plan and build structures utilizing Nature's material forces for the benefit of mankind, has apparently given but little indication that he possesses the knowledge and skill required to handle human forces for the benefit of mankind. It would appear that that kind of leadership does not appeal to the engineer. He has made the instrument, even composed the music, which others have played to sway the multitude. In short, the engineer, if he has the qualities for leadership, has not been leading, except in his own field and amongst his own kind where of course he stands preeminent.

The world today needs a Moses to lead it out of the Wilderness. Moses must have been an engineer. That job of his in the Red Sea would indicate that he was an hydraulic engineer. Can the engineering profession aspire to such leadership as is today needed? No one knows. We think it can. But it would appear that the engineer has not been thought of in that connection. Why? Simply because he has in the past taken so little part in pubic affairs outside his profession. Here then lies the opportunity, the first step being to organize nationally and do the things recommended by our Joint Committee of the four national engineering societies.

One of the things recommended is the active support of the bill in Congress to create a National Department of Public Works.

Such a department would at once bring the engineering profession before the eyes of the world. It is such a normal and proper move that one would think it would carry itself. But it will not. You can be sure of that. It is forty years since the effort to secure such a department was first made. It failed as have all subsequent efforts for want of being properly pushed. And who is to push? You, gentlemen of the engineering profession. Not one, nor a few, but all of you and all together, and at this particular time.

I am now come to near the end of what I had in mind to say to you. There are other things that should be added to make my story of the year's doings complete. But something must be left for my successor in office to say next year. A president's address is not, I fear, the best medium of conveying a message to the multitude. It partakes more of the nature of an embalming process of ideas. It will find its mausoleum in the TRANSACTIONS. Some thousands of years hence it will be discovered, and the world of that day will comment on it much as we now do on the papyrus from the tombs of Egypt. But I have had great pleasure, notwithstanding, in preparing my address.

Our worthy editor has for several weeks been keen to discover the title of my address. He was somewhat perturbed when I informed him that there was to be no title; that I proposed to be free, up to the last moment to write what I pleased. The fact that I have succeeded in eluding him I consider some accomplishment, as you would agree did you know him as well as I do.

And now my adieus. It has been for me a happy year. I am grateful to you for your confidence, and your good will. It has brought forth the best that is in me. While I have many times failed to measure up to what I consider a proper specification for a president of The American Society of Mechanical Engineers, I have the satisfaction of feeling that I have stretched myself to the utmost in an effort to render all the service of which I was capable.

No. 1711

STEAM USE IN TEXTILE PROCESSES

By George H. Perkins,[1] Boston, Mass.
Member of the Society

The following paper, presents tables of comparative figures showing the importance of the use of steam for the manufacturing processes in the textile industry. Analyses are given of the various uses to which steam is put and suggestions are made for research and improvement of economy.

THE collective textile industries of the United States consume 9,662,600 tons of coal annually and require for their operation a total primary power of 2,495,000 hp. of which 66.8 per cent, or 1,665,900 hp., is developed by steam prime movers.

2 Tables 1 and 2 give figures compiled from the United States Census of Manufactures for 1914, showing the distribution of coal and primary power to the seven principal manufacturing groups included in the textile industry. These figures serve to emphasize the magnitude of the steam requirements of this industry, which are to be considered in this paper with particular reference to the heat demands of textile processes.

RELATIVE ECONOMY OF TEXTILE INDUSTRIES

3 The relative fuel economy of the different textile groups is shown in Table 3, derived from data given in Tables 1 and 2. The items are listed in the order of comparative performance as figured on a common basis of power production and show clearly the effect on the over-all economy of the extensive use of steam in the various processes of manufacture.

4 The cotton goods and cordage groups rank first, owing to their small demands for process steam. The relatively good economy shown by the woolen and worsted group may be attributed to a more general utilization of exhaust steam in processes and

[1] Consulting Engineer, 34 Batterymarch St.

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

the better balance usually existing between the steam demands of prime movers and processes.

TABLE I COAL CONSUMPTION AND STEAM POWER REQUIREMENTS OF THE TEXTILE INDUSTRIES

Industry	Anthracite 1000 Tons of 2240 lb.	Per Cent of Total Anthracite	Bituminous 1000 Tons of 2000 lb.	Per Cent of Total Bituminous	Steam Power 1000 Hp.	Per Cent of Total Steam Power
Carpets and Rugs.......	74.7	3.1	193.7	2.7	29.5	1.8
Cordage, Jute, Twine and Linen................	91.9	3.8	195.2	2.7	65.8	4.0
Cotton Goods...........	313.5	13.0	3634.2	50.0	1011.3	60.7
Dyeing and Finishing [1]...	490.6	20.5	896.6	12.4	111.5	6.6
Hosiery and Knit Goods..	118.1	4.9	484.3	6.7	80.8	4.9
Silk Goods..............	1053.7	44.0	249.9	3.5	78.3	4.7
Woolen, Worsted and Felt	257.4	10.7	1608.8	22.0	288.7	17.3
Total..............	2399.9	100.0	7262.7	100.0	1665.9	100.0

[1] Exclusive of that done in textile mills.
Fuels other than coal have been omitted as the equivalent total is small.

TABLE 2 DISTRIBUTION OF PRIMARY POWER IN THE TEXTILE INDUSTRIES

Industry	Total Primary 1000 Hp.	Steam Power 1000 Hp.	Water Power 1000 Hp.	Internal Combustion Engines 1000 Hp.	Rented Power	
					Electric 1000 Hp.	Other 1000 Hp.
Carpets and Rugs.......	44.0	29.5	4.1	....	9.2	1.2
Cordage, Jute, Twine and Linen............	93.9	65.8	14.4	2.9	10.7	0.2
Cotton Goods..........	1585.9	1011.3	314.2	4.0	252.9	3.5
Dyeing and Finishing....	130.1	111.5	9.9	0.7	7.2	0.8
Hosiery and Knit Goods..	125.7	80.8	14.6	1.1	26.3	2.9
Silk Goods.........	117.0	78.3	7.6	1.8	23.8	5.5
Woolen, Worsted, and Felt	398.4	288.7	76.3	2.7	25.0	5.7
Total..........	2495.0	1665.9	441.1	13.1	355.1	19.8
Per Cent of Total.... ..	100.0	66.8	17.7	0.5	14.2	0.8

5 The dyeing and finishing group has a comparatively small power demand and a most extensive use for process steam. The relation of coal to power in the silk goods group appears out of proportion in comparison with the other industries and must be attri-

buted, in part, to the wasteful use of steam and a more extensive use of the lower grades of fuel.

6 The validity of the results given in Table 3 may best be shown by citing a number of specific examples of individual plants of different types, giving the actual fuel consumption and power production. This is shown in Table 4. These figures check well with the average for the respective groups as given in Table 3.

TABLE 3 COMPARATIVE FUEL ECONOMY OF THE TEXTILE INDUSTRIES

Industry	Equivalent Bituminous 1000 Tons of 2000 lb.[1]	Per Cent of Total	Steam Power 1000 Hp.	Per Cent of Total Primary Power	Tons Per Hp. Per Year	Lb.* Per Hp. Per Hr.[2]
Cotton Goods...........	3947.7	40.8	1011.3	63.7	3.90	3.12
Cordage, Jute, Twine and Linen................	287.1	3.0	65.8	70.0	4.35	3.48
Woolen, Worsted and Felt	1866.2	19.3	288.7	72.3	6.50	5.20
Hosiery and Knit Goods..	602.4	6.2	80.8	64.5	7.45	5.96
Carpets and Rugs.......	268.4	2.8	29.5	67.2	9.08	7.26
Dyeing and Finishing....	1387.2	14.4	111.5	85.8	12.50	10.00
Silk Goods.............	1303.6	13.5	78.3	67.0	16.70	13.36
Total.............	9662.6	100.0	1665.9	66.8	5.80	4.64

[1] Anthracite ton of 2240 lb. included as equivalent of Bituminous ton of 2000 lb.
[2] Based on 2500 hr. operation per year.

TABLE 4 ACTUAL ECONOMY OF INDIVIDUAL PLANTS

Kind of Plant	Total Coal Tons	Total Steam Power Hp.	Coal Per Hp. Per Year
Cotton goods, no dyeing...............	10,500	2,800	3.75
Cotton goods, with dyeing..............	13,000	3,000	4.33
Jute goods............................	9,400	2,200	4.28
Worsted yarns.........................	1,560	340	4.6
Woolen goods..........................	1,800	270	6.67
Bleachery and finishing................	10,200	800	12.75
Print works...........................	60,000	4,000	15.00

FUEL PER UNIT OF PRODUCT

7 It is still the practice in many textile plants to compare the total weight of coal burned with the weight of yardage of goods processed or produced. Such ratios are at best only rough measures

of the actual economy and are subject to wide variations, with fluctuations in the quantity or quality of the product, grade of coal, etc. As a matter of interest only, the following gives the average range of these ratios for a number of the industries:

Industry	Coal Per Unit of Product
Cotton goods	2 to 2.25 lb. per lb. of cloth produced.
Jute goods	1 to 1.25 lb. per lb. product.
Woolen goods	6 to 7 lb. per lb. of cloth finished.
Worsted yarns	3 to 3.25 lb. per lb. of yarn produced.
Wool scouring and combing	0.8 to 1.2 lb. per lb. of grease wool processed.
Bleaching and finishing	0.8 to 1.3 lb. per lb. of goods processed.
Print works	2.5 to 3 lb. per lb. of goods processed.

COAL USED BY PROCESSES

8 The approximate proportion of the total coal which is used by the processes in this industry is given in Table 5.

9 These figures are based on data obtained from the analysis of representive plants. While the requirements of individual mills in the same group will vary according to local conditions, it is believed that the figures presented are conservative and will serve the present purposes.

TABLE 5 PROPORTION OF TOTAL COAL USED BY PROCESSES

Industry	Total Coal (1000 Tons)	Per Cent Used by Processes[1]	Coal used by Processes (1000 Tons)
Cotton goods	3,947.7	6	236.8
Cordage, jute, etc.	287.1	6	17.2
Woolen and worsted	1,866.2	70	1,306.2
Hosiery	602.4	50	301.2
Carpets and rugs	268.4	60	161.0
Dyeing and finishing	1,387.2	75	1,040.3
Silk goods	1,303.6	60	781.8
Total	9,662.6	39.8	3,844.5

[1] These percentages include the proportional boiler losses and also credit for use of exhaust steam.

IMPORTANCE OF HEAT APPLICATIONS

10 The foregoing facts emphasize in general terms the importance of the problems of heat application to textile processes. These problems warrant far greater attention than they have yet received,

both from the standpoint of production efficiency and from the consideration of fuel economy. But little progress has been made along either line in comparison with the marked advances in the art of generating steam and power.

11 Many textile power plants are "saving at the spigot and wasting at the bung" and the average dye-house, bleachery or print-works is usually considered a necessary evil to be endured up to the limit of the boiler capacity of the plant. While production efficiency must be considered as of the first importance, the present fuel situation demands rigid conservation in the use of both coal and steam.

EFFECTS OF HEAT ON TEXTILE MATERIALS

12 The purposes for which heat is applied to textile materials in process are numerous and varied and only the principal ones will be summarized as follows:

Process	Material	Purpose or Effect of Heat
Scouring	Wool	To assist the action of the soap and alkali.
Carbonizing	Wools and shoddies	To free the acid from moisture to permit carbonization.
Combing	Worsted wools	To facilitate drawing through the pins.
Dyeing	All materials	To assist the fixation of dye-stuffs.
Kier Boiling	Cotton goods	To assist action of liquor in softening impurities.
Drying	All materials	Evaporation and removal of moisture.
Tentering	Cotton and worsted goods	To dry goods under tension retaining width.
Steaming	Woolen, worsted and print goods	To shrink, remove glaze and to set fabrics and colors.
Soaping	Print and dyed goods	Warm water removes surplus color and improves appearance.
Ageing	Print and dyed goods	To set or develop colors.
Washing	Woolen and worsted goods	Warm water removes oil and soap applied in previous processes.
Pressing	Woolen and worsted goods	With pressure, flattens surface for finish effect.
Calendering	Cotton goods	With pressure and friction, assists in producing luster.

CLASSIFICATION OF PRINCIPAL HEAT-USING PROCESSES

Application	Machines of Similar Use
Direct contact of materials with steam or water vapor	Steamers for woolen goods, print goods, etc. Ageing machines for print and dyed goods Yarn conditioning machines
Direct contact of materials with heated surfaces for warming stock, pressing, polishing, drying, etc.	Worsted comb steam chests Worsted back washers Plate presses for woolen goods Rotary presses for woolen goods Slashers and dressers Cylinder and can dryers Calenders for cotton goods Polishers for twine, thread, etc.
Drying of materials with air heated by direct or indirect radiation	Hot air slashers and dressers Cotton and wool stock dryers Carbonizing dryers Cloth dryers Shrinking dryers Tenter frames Dry rooms for yarn Back washer dryers
Direct use of steam for boiling or heating liquids and use of warm water	Dye tubs and dyeing machines Scouring bowls for wool Bleaching kiers Cloth washers Crabbing machines Starch and size cooking Soapers
Indirect use of steam for heating liquids	Bleaching kiers with closed heaters and circulating pumps. Jacketed kettles for size, starch or dyestuffs. Tubs and dyeing machines with closed submerged coils. Preheating of liquids for process or storage by closed heaters.

13 The direct or indirect heating of rooms containing processes requiring careful temperature and humidity control may also be mentioned under the general heading.

FACTORS AFFECTING PRODUCTION AND ECONOMY

14 Some of the various conditions commonly found, which affect either production or economy are as follows:

Production	Economy
Wet steam	Leakage from traps, vents, etc.
Poorly designed steam distribution system	Wasted drips

Incorrect arrangement of traps	Overheating or overcooling through careless operation
Deficient trap capacity	Radiation losses through dryer housings, etc.
Delays in heating liquor, dyes, etc.	Waste of exhaust steam or condensing water
Inefficient air circulation in dryers	Lack of hot water storage
Deficient dryer radiation	Failure to recirculate air in dryers

15 In addition to the data given above, there is usually little information available either of the steam requirements or demands.

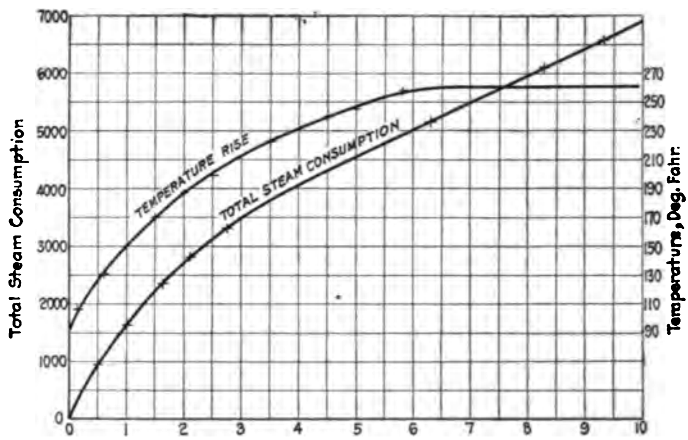


FIG. 1 TEMPERATURE RISE AND STEAM CONSUMPTION IN KIER TEST.

Fig. 1 shows graphically the results of a test on a 3-ton kier with outside heater and pump circulation. The rate of temperature rise and the variation in the total steam consumption are shown throughout the entire boil.

While considerable progress is being made in means and methods for correcting the conditions mentioned, there is still room for much improvement.

16 Many of the larger mills are making complete steam surveys of their plants periodically to check use against demand and also to enable steam costs to be properly apportioned to the various processes. Modern steam meters give reliable results and invariably reveal many surprising conditions which require attention and which cannot be disclosed by any other means.

TEST RESULTS

17 The results of tests made at different plants are given in the following tabular form, and in Figs. 1 and 2, merely to illustrate

common practice and not as standards of high performance. They will however be found useful for comparative purposes.

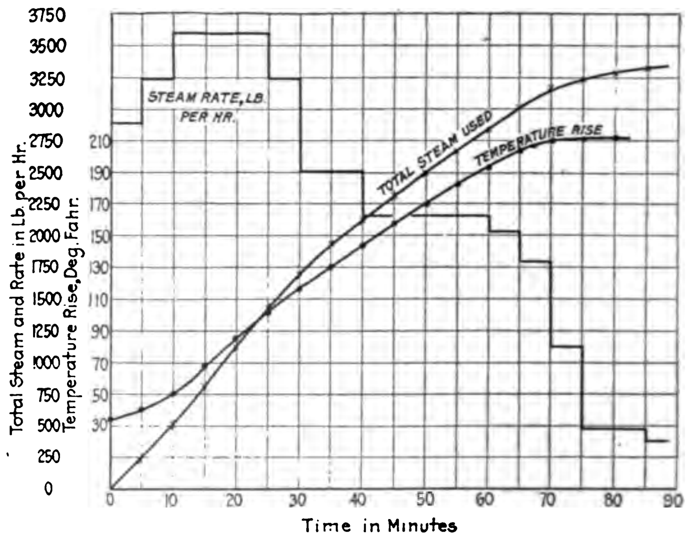


FIG. 2 TEMPERATURE RISE AND STEAM CONSUMPTION IN DYE-TUB TEST.

TABLE 6 KIER TEST (SODA BOIL)

See Fig. 1

Size and kind of kier	4 ton, pressure type, injector circulation
Weight of dry cloth	8000 lb.
Kind of cloth and wt in yd per lb, muslin 56 x 52	7.25 yd.
Weight of water entering in goods	8000 lb.
Weight of liquor	16800 lb.
Initial temperature	80 deg.
Final temperature	245 deg.
Steam pressure, gage	25 lb.
Time of boil	10 hr.
Total steam used	8250 lb.

APPROXIMATE HEAT BALANCE

	Pounds of Steam	Per cent of Total
Heating wet goods	2264	27.5
Heating liquor	2971	36.0
Heating kier	238	2.9
Radiation losses	1860	22.5
Vent evaporation and unaccounted for losses	917	11.1
Total	8250	100.0

This test gives the steam consumption for a ten-hour boil, on a 4-ton pressure kier with injector circulation. The approximate steam distribution is also given in the form of a heat balance. The large loss through radiation should be noted.

TABLE 7 DYE TUB TEST

See Fig. 2

Galls of liquor boiled	1845
Pounds of liquor boiled	15300
Initial temperature	34 deg. fahr.
Final temperature	210 deg. fahr.
Total temperature rise	176 deg. fahr.
Time of total temp. rise, min.	70
Total steam used during total temperature rise, lb.	3230

The above table and Fig. 2 give the results of a test on a dye tub during the period of the total temperature rise. The curves in Fig. 2 show the rate of temperature rise, rate of steam flow in pounds per hour and also the total amount of steam used. It should be noted that fully one-half of the total steam used is consumed within the first thirty minutes and below a temperature of 120 deg. Fahr.

SUGGESTIONS FOR INVESTIGATION AND IMPROVEMENT

18 It is hoped that the outline given of this subject will help to stimulate greater attention to this important matter.

19 The following suggestions are offered as concrete lines for useful and practical study and research.

a Further investigation of conditions of temperature, humidity and air circulation for drying various materials with maximum production and minimum heat consumption

b Development of more efficient arrangement of radiation and air circulation in dryers

c Automatic control of the production rate of dryers, through conditions resulting from drying results obtained

d Further development and application of temperature control devices to heat using processes

e More effective utilization of exhaust steam in processes, heating water storage, etc.

f Recovery of heat from spent and rejected liquors

g Development of insulating materials to be used under conditions where equipment is subject to extreme moisture conditions or to mechanical injury

h Use of supplementary air circulation with can dryers.

20 The field is a broad one with many varied and complex problems, but the coöperative effort of manufacturers, machine builders and engineers should bring about much needed improvement in the present average practice.

TABLE 8 TENTER FRAME TESTS

Aver. weight of cloth, yd. per lb.	5.4
Duration of test, hr.	7.0
Steam pressure, gage	30 lb.
Outside temperature	70 deg. fahr.
Temp. of air supplied, dry bulb	163 deg. fahr.
Temp. of air supplied, wet bulb	85 deg. fahr.
Temp. of return air, dry bulb	138 deg. fahr.
Temp. of return air, wet bulb	84 deg. fahr.
Air supplied, cu. ft. per min.	8700
Weight of cloth dried per hr., lb.	650
Weight of water evaporated per hr., lb.	106
Per cent of water in wet cloth	14

	Without air recirculation	With air recirculation
Steam condensed per hr., lb.	1122	433
Steam per lb. of dry cloth, lb.	1.71	0.67
Steam per lb. of water evap., lb.	10.5	4.1
Lb. of cloth dried per lb. of steam	0.58	1.5
Yd. of cloth dried per lb. of steam	3.23	8.1

This table gives the results of tests on a tenter frame both with and without recirculation of air. The economy resulting from recirculation should be especially noted.

TABLE 9 CAN DRYER TESTS

Test No.	1	2	3	4	5
Wt. of goods, yd. per lb.	1.45	2.5	5.15	8.10	10.85
Kind of goods	Shoe Lining	Shoe Lining	Sheeting	Twill	Lawn
Steam pressure in cans, gage	5.7	4.7	8.1	5.5	5.3
Number of cans	47	47	47	23	13
Size of cans	23 in. x 100 in.	23 in. x 100 in.	23 in. x 100 in.	23 in. x 141 in.	36 in. x 124 in.
Material of cans	Copper	Copper	Copper	Tinned Iron	Copper
Total can surface, sq. ft.	2350	2350	2350	1625	1265
Cloth dried per hr., yd.	2393	4420	10,135	17,750	12,730
Cloth dried per hr., lb.	1650	1768	1968	2191	1173
Per cent of water in wet cloth	48.8	51.8	51.7	46.1	44.4
Water evaporated per hr., lb.	1570	1892	2112	1873	938
Steam condens. per hr., lb.	2462	2625	3424	2530	1310
Steam per lb. dry cloth, lb.	1.49	1.48	1.74	1.15	1.12
Steam per lb. water evaporated, lb.	1.57	1.39	1.62	1.34	1.40
Number of strings	2	2	2	4	4
Average speed yd. per min.	20	36.5	84	74	50
Yards of cloth per lb. steam	0.97	1.68	2.96	7.00	9.7
Steam per hr. per sq. ft. can surface	1.05	1.12	1.46	1.56	1.04

This table gives the results of 5 tests on can dryers with a range of weights of goods from 1.45 to 10.85 yards per pound. It should be noted that the amount of steam used per pound of water evaporated is comparatively uniform throughout the tests.

TABLE 10 CLOTH DRYER TESTS

	Uninsulated Housing Woolen suiting	J-M No. 1 Type Insulated Housing Woolen suiting
Weight of goods, oz. per yd.	13 oz.	13 oz.
Weight of wet goods processed per hr., lb.	900	1047
Weight of goods dried per hr., lb.	523	609
Yards dried per hr., lb.	644	747
Weight of water evaporated per hr., lb.	377	438
Per cent moisture in wet cloth	41.9	41.9
Heating surface, sq. ft.	2240	2240
Steam pressure, gage	42.5	43.5
Temperature in dryer	135 deg. fahr.	189 deg. fahr.
Weight of steam condensed per hr., lb.	875	738
Steam condensed per sq. ft. of radiation	0.39	0.33
Steam per lb. of dry cloth, lb.	1.67	1.21
Steam per lb. of moisture evaporated	2.32	1.85
Yards dried per lb. of steam	0.735	1.01

TABLE 11 CARBONIZING DRYER TEST

	Uninsulated Housing Woolen and cotton thread waste	J-M No. 1 Type Insulated Housing Woolen and cotton thread waste
Weight wet stock processed per hr., lb.	402	585
Weight stock dried per hr., lb.	266	391
Weight of water removed per hr., lb.	136	194
Per cent moisture in wet stock	32.4	33.3
Heating surface, sq. ft.	2488	1952
Temperature in dryer	279 deg. fahr.	261 deg. fahr.
Steam pressure, gage	80	80
Weight of steam condensed per hr., lb.	1044	432
Steam condensed per sq. ft. of radiation, lb.	0.42	0.22
Steam condensed per lb. of dry stock, lb.	3.93	1.11
Steam condensed per lb. of moisture removed, lb.	7.68	2.23

Tables 10 and 11 give the results of tests on a woolen cloth dryer and a carbonizing stock dryer and show clearly the effect of using insulated housings.

NOTE: The author wishes to acknowledge the contribution of the above tests on insulated and uninsulated cloth and stock dryers from the H. W. Johns-Manville Co.

DISCUSSION

CHARLES T. MAIN emphasized the desirability of there being as much information as possible of the type contained in the author's paper presented to the society. In the design of the power plant for textile mills engineers were often groping in the dark because of lack of information as to the steam requirements.

One of the greatest wastes in mills was that of the hot water. He cited the case of a mill in which he had installed at the request of the owners facilities for greater amounts of hot water than were customarily used. Upon being told after the mill was in operation that the power plant was inefficient, he reported, after conducting a boiler test to prove the efficiency of the steam generating apparatus that the fault lay in the fact that about twice as much hot water was being used as was ordinarily necessary.

With reference to the apparently large amount of coal used per horsepower in a silk mill as shown by the author's figures he said that he thought that silk mill operators were unwilling to use exhaust steam to the extent that it is used in a cotton mill for fear of damaging their product.

John C. Percy thought that the reason for the greater coal consumption of the silk mill might lie in the fact that less heat was available from the machinery and that a larger proportion of heat was necessary for heating the mill, there being insufficient exhaust steam for heating purposes.

The Author, in answer to a question, said that his figures as reported from the census included plants operated with other fuels than coal but that on reducing these to an equivalent coal basis, the total amounted to such a small percentage of the aggregate coal as to be negligible for his purposes.

He pointed out that the census figures were five years old and that present-day statistics would show material increases in the use of all fuels, particularly fuel oil.

No. 1712

THE THERMAL CONDUCTIVITY OF INSULATING AND OTHER MATERIALS

By T. S. Taylor,[1] East Pittsburgh, Pa.
Non-Member

Despite the fact that numerous thermal-conductivity measurements have been made, but very few data are available for such materials as are used in the construction of electrical machinery. To further investigate this field the author accordingly conducted a series of tests based upon two methods, both of which are explained in detail in this paper. The results obtained are tabulated for such materials as fish paper, fuller board, cambric, mica tape, various kinds of woods, asbestos, plate glass sheet steel, wool felt, etc.

Although numerous experimenters have been interested in making thermal-conductivity measurements, but very few data are available for such materials as are used in the construction of electrical machinery. The most important work on such insulating materials thus far available was done by Symons and Walker.[2] The materials tested by these authors were special and consequently the values obtained cannot be taken as applicable to similar materials used in the construction of electrical machinery in this country. The investigations herein described were therefore undertaken to obtain values of the thermal conductivity of insulating and other materials, the values of which are of direct interest to those concerned with the heat problem in electrical apparatus. Results have also been obtained for numerous other materials of general interest.

THE NORTHRUP THERMAL BRIDGE

2. As a preliminary step it seemed worth while to try the "Thermal Bridge" method suggested by Prof. E. F. Northrup.[3] The

[1] Westinghouse Research Laboratory, Westinghouse Electric and Manufacturing Company.

[2] See Journal of the Institute of Electrical Engineers (London), vol. 48, p. 674, 1912.

[3] See American Electrochemical Society, Journal no. 24.

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

bridge used is shown in Fig. 1. It consisted of two soapstone cylinders each $4\frac{5}{8}$ in. in diameter, one being $8\frac{1}{2}$ in. long and the other 3 in. long. Each cylinder consisted of an inner core $1\frac{5}{8}$ in. in diameter surrounded by a coaxial cylinder having a wall thickness of $1\frac{3}{8}$ in. The two faces along MN separating the two parts of the apparatus were ground so as to fit very closely. A spiral groove was cut in

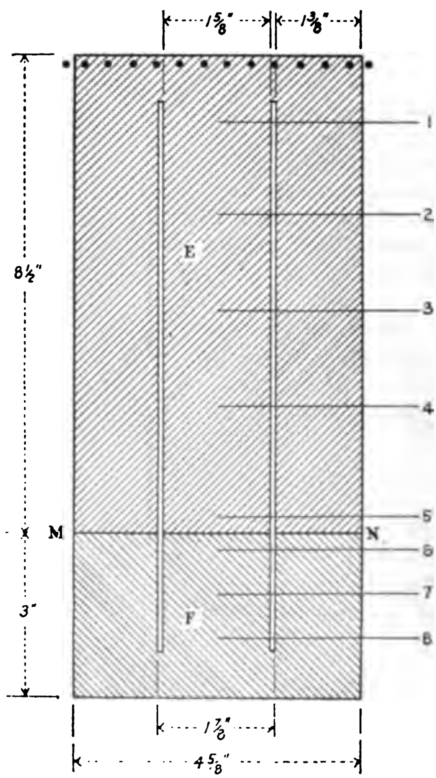


FIG. 1 DIAGRAMMATIC SKETCH OF THE NORTHRUP THERMAL BRIDGE

the top of the longer cylinder E and a heater wire placed in this groove. Small holes, 1, 2, 3, etc., were drilled at right angles to the axis of the cylinders through the outer wall and into the central core as indicated. Copper-constantan thermocouples made of 0.005-in. wire were inserted in these holes for the purpose of determining the temperature gradient along the cylinders. The lower part of the apparatus F was placed on a brass box which served as a cold-temperature reservoir when kept filled with water and ice or when water was kept circulating through it. The heat generated at the top would flow down the core and outer wall through the junction *MN* to the reservoir. The purpose of the core and surrounding wall was to insure a uniform flow of heat through the core. Felt was placed around the outer cylinder to further prevent undue loss of heat from the surface of the apparatus.

3 If the distance from thermocouple 1 to thermocouple 5 is made the same as that from 2 to 8, and the conditions are such that a uniform temperature drop exists along the core, the thermal conductivity of a material placed in *MN* can be determined in terms of soapstone. First suppose the above conditions to exist when there is no specimen in *MN*. Then the temperature drop will be uniform along the apparatus, as can be tested by the thermocouples. When a sample is inserted in *MN* and the temperatures of 1, 5, 2 and 8 are measured, the drop between 2 and 8 exceeds that from 1 to 5 by an amount equal to the drop through the sample. The soapstone equivalent of the sample is then readily calculated from the temperature drop through the sample and the temperature drop per unit length along the soapstone. This is true provided that the drop is uniform along the soapstone and that there is no temperature drop across the junctions of sample and soapstone at *MN*. In the actual experiment the procedure was slightly different from that outlined as the distance from thermocouple 1 to 5 was not exactly the same as that between 2 and 8. As seen in Fig. 1, several thermocouples were inserted along the cylinder; and then from the curve showing the relation between temperature and distance from the upper thermocouple (Fig. 2) the soapstone equivalent of the sample under test was readily determined. The thermoelectromotive forces of the thermocouples were measured by means of a thermocouple potentiometer and their equivalent temperatures obtained by reference to a calibration curve previously determined for the thermocouple wire used.

4 If the two parts of the apparatus are fitted as closely as possible together, thus having no material between them at *MN*, and the temperatures as indicated by the thermocouples measured, curve *A*, Fig. 2, represents the temperature distribution along the cylinders. The positions of all thermocouples are measured from the uppermost one. As is seen, there is a very marked temperature drop at the division between the two parts of the apparatus, equivalent to 2.5 deg. cent. for this particular temperature distribution. This is about 4.4 per cent of the total drop between thermocouples

1 and 8. A sheet of paper 0.001 in. thick was then inserted, which caused a slight increase in this temperature drop although it by no means doubled it. This being observed, several things were tried to see whether the drop could be eliminated either entirely or in part. It was found that by putting vaseline or glycerine between the surfaces *MN*, this division drop was almost, if not entirely, eliminated. This was true at least for relatively small temperature gradients. Curve *B*, Fig. 2, illustrates this point very clearly. The

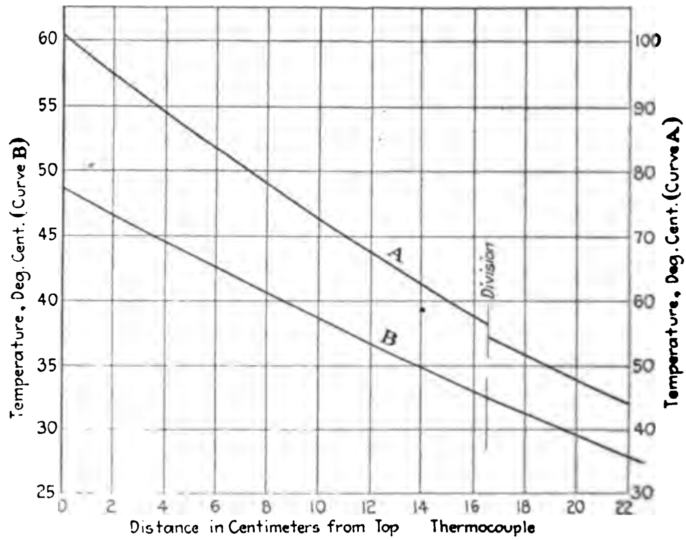


FIG. 2 CALIBRATION CURVES OF THE NORTHRUP THERMAL BRIDGE

curve shown was obtained when the division *MN* was well lubricated with vaseline.

5 To test whether the vaseline soaked into the soapstone sufficiently to affect the results, two rings and disks were cut from the same piece, one ring and one disk being thoroughly soaked in hot vaseline; the temperature gradients were then obtained both with the vaseline-soaked ring and disk inserted in *MN* and again with the unsoaked ones in the same position. No noticeable difference was found to exist between the two cases as the temperature gradients along the apparatus and through the disks were identical.

SOAPSTONE EQUIVALENT OF PARAFFINED FISH PAPER

6 Disks and rings were cut from 0.015-in. paraffined fish paper so as to fit the core and surrounding ring at the junction *MN* of the apparatus. The temperature gradient along the apparatus was determined when different numbers of sheets (disks and rings) were in position *MN* for various temperature differences between the hot and cold ends. A typical set of observations is represented by the curves in Fig. 3. The ordinates are the temperatures in degrees centigrade corresponding to the respective distances of the thermocouples 1, 2, 3, etc., measured from couple 1. Curve *A* was obtained when the sample was composed of 6 sheets and Curve *B*

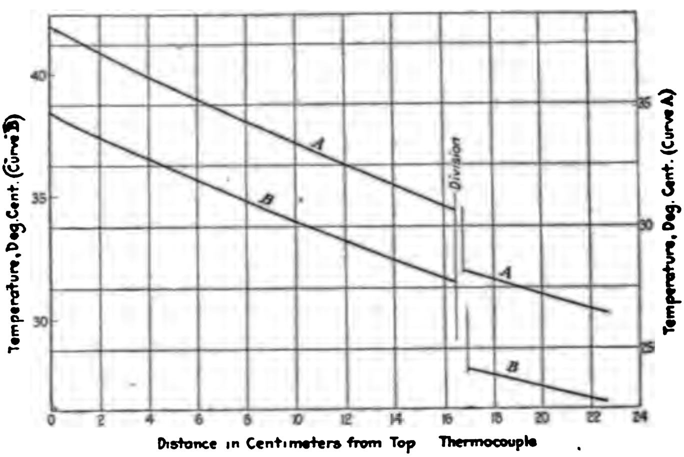


FIG. 3 CURVES ILLUSTRATING TEMPERATURE GRADIENT FOR PARAFFINED FISH PAPER

for 10 sheets of 0.015-in. paraffined fish paper. Vaseline was used between the sheets in order to diminish the division drop as much as possible. Sufficient pressure was applied to the top of the apparatus to insure good contact between the surface of the constituents of the sample and the soapstone. The values of the soapstone equivalent of a thickness of one inch of paraffined fish paper obtained from the curves in Fig. 3 were 32.8 in. and 32 in., respectively. That is, one inch of paraffined fish paper composed of sheets 0.015-in. thick would have a thermal resistance equivalent to the foregoing values of soapstone. These values of the soapstone equivalent of fish paper were obtained from the curves in Fig. 3 as follows: The drop through the sample (A) is 2.55 deg. cent. and hence the drop per centimeter is thus divided by the thickness of the sample, that is, 2.55/0.2286, or 11.15. The average temperature gradient obtained from the slope of the curve just above and below the sample is 0.350 deg. per cm. Therefore 1 cm. of the fish paper is equivalent to 11.15/0.350 or 32.8 cm. of soapstone, or one inch of fish paper will have the same thermal resistance as 32.8 in. of soapstone.

7 Similar results were obtained for other samples composed of different numbers of sheets. These results are given in Table 1, and all values refer to paraffined fish paper 0.015 in. thick. As can be seen from the table, this method gives a wide variation in results. Various factors produce these variations, one of which is due to the fact that the temperature does not fall uniformly along

TABLE 1 SOAPSTONE EQUIVALENT OF PARAFFINED FISH PAPER

No. sheets in sample	Total thickness of sample in in.	Aver. temp. of sample, deg. cent.	Total drop from couple 1 to 8	Drop in sample	Soapstone equivalent
5	0.077	29.7	11.5	2.58	37.4
2	0.030	29.6	10.6	1.05	37.5
6	0.088	33.3	11.7	2.55	32.8
10	0.155	25.7	11.8	3.66	32.0
4	0.060	30.0	11.2	1.70	27.2
15	0.221	28.0	12.0	4.25	24.7
20	0.305	28.5	14.4	6.30	28.2
15	0.228	28.6	14.0	5.45	33.4
10	0.150	27.7	12.6	3.58	27.2
50	0.75	48.5	55.7	10.5	32.6

the sample, heat being lost by radiation, conduction and convection from the inner core to the outside wall. This makes the slope of the curve, which determines the value of the temperature gradient, quite uncertain, especially just above and just below the sample.

8 The final results which depend upon these gradients vary considerably. This large variation in results made it quite evident that it would be advisable to attempt the work by use of some other method. The one described in the following paragraphs was used and found very satisfactory. The thermal-bridge method has the disadvantage also of being an indirect one which would necessitate a separate determination of the thermal conductivity of soapstone itself. This, however, would have been a small matter had the

results been in close and satisfactory agreement. Such results as are recorded in Table 1 by no means fulfill these conditions.

THE "THERMAL METER"

9 A sketch of the "thermal meter" used is shown in Fig. 4. It consisted essentially of an electric heater *H* constituting two hot equitemperature surfaces or sources of heat and two cooling chambers *E*, *E* (one on either side of the heater) or cold constant-temperature surfaces. The heat generated in *H* passed laterally through the samples *I*, *I*, of a given material to the cold reservoirs *E*, *E*. The heater was made from two disks of soapstone 9 in. in diameter and $\frac{3}{8}$ in. thick. Each disk had a helical groove of $\frac{3}{16}$-in

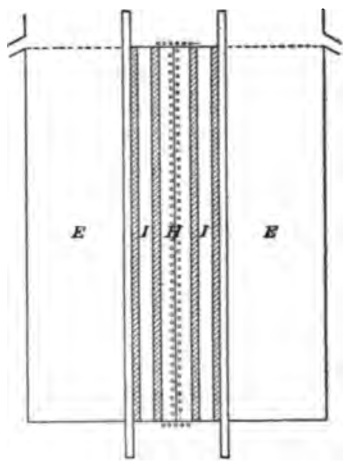


FIG. 4 DIAGRAMMATIC SKETCH OF THE THERMAL METER

pitch cut in one face. A heater wire, No. 21 constantan, was wound and cemented securely in the groove of each disk and then the iron disks were cemented together with the sides containing the heater wire adjacent. The two heating elements in the two disks were joined together at the center by means of a peg in one disk being pushed into the spring contact in the other. Potential leads were brought out from each heating element at points 2 in. from the centers. It was later found that potential leads fastened to the heating elements at the points where the wire started in the outer terminals of the spirals served equally well, since the temperature coefficient of resistance of the constantan wire is very small and furthermore the temperature of the heater was constant over its entire face. Extra turns of wire were wound around the outer edge of the heater in order to prevent the loss of the heat generated in the heating element proper through the edge of the heater. After several trials it was found that this procedure gave a heater which in use had a very constant temperature over its two faces, even up to the outer edge.

10 Two samples of the material to be tested were always used, each being 9 in. in diameter and from 0.1 to 0.75 in. thick, depending upon the nature of the material. One sample was placed on each side of the heater *I*, *I*, Fig. 4. Extra disks of lagging of the same material of the sample or something else suitable were placed on each side of the sample, as shown by the shaded portions in Fig. 4. This made the ultimate drop at high temperatures less than it otherwise would have been, and likewise gave a wider range of mean temperature. The faces of the cold reservoirs *E*, *E*, constituting the cold equitemperature surfaces, were made of heavy brass having a diameter of 10 in. The samples, heaters, and cold reservoirs were held securely together by means of bolts extending between these two plates. At first strong spiral springs were used around these connecting bolts to insure uniform pressure, but it was found later that equally satisfactory results could be obtained by merely turning down the nuts on the bolts till the samples, etc., were drawn tightly together. Thermocouples of 5-mm. copper-constantan wire were inserted on each side of the samples *I*, *I*, under test. Great care was taken in order to insure good contact between the sample and the thermocouple junction. Two couples were placed on each side of a sample, one at the mid-point and another about 1½ in. from the center. The electromotive force of the couples (the cold junction being always kept at 0 deg. cent.) was measured by means of a thermocouple potentiometer. The current in the heater was likewise measured by the same potentiometer by measuring the drop through a standard resistance placed in series with the heater. The potential drop per unit length of heater wire was also measured by means of the potentiometer. This necessitated placing a high resistance in parallel with the heater and then measuring the potential drop across a small fraction of this. The current was supplied by a storage battery and consequently remained quite steady. The cold equitemperature surfaces *E*, *E*, were maintained so by having water from the tap circulating through them continuously. The outer edges of the samples and heater were surrounded by felt in

order to prevent undue loss of heat from the edges of the heater and samples.

11 In order to facilitate the work a second apparatus having the same parts as the one described above was constructed. The only difference being that the heater, etc., were of square cross-section. This heater was made of soapstone slabs 12 in. square. The heater wire (No. 21 constantan) was wound back and forth in parallel slots $\frac{3}{16}$ in. apart in each of the parts of the heater. This made its construction quite easy. The elements of the two parts of this heater were joined in parallel as its resistance would have been too high for the available voltage had they been in series. The potential leads of this heater were joined at the ends of a wire in a single groove, thus measuring the drop in a 12-in. length of the wire. Four sets of potential leads were inserted (two in each half) and the average of the four potential drops used in calculating the drop per unit length of heater wire. As for the round heater, extra turns of wire were put around its outer edges to insure a uniform temperature source. Great care was taken to have the same amount of resistance in each element of the heater, since they were joined in parallel. The faces of the cooling reservoirs were of cast brass $13\frac{1}{2}$ in. by $13\frac{1}{2}$ in. by 1 in., and being heavy they did not buckle when bolted together over the heater and the materials tested. Identical results were obtained with the two pieces of apparatus for a given material.

12 The chief advantage of this method of determining thermal conductivities is the ease with which the quantities involved are measured. When heat passes continuously from one plane constant-temperature surface to another parallel to it, the quantity of heat flowing per second is given by the relation,

$$Q = \frac{kA(t_2 - t_1)}{d}$$

where k is the thermal conductivity of the intervening medium, d its thickness or the distance between the constant-temperature surfaces, A the area through which the heat passes, and t_2, t_1, the temperatures of the hot and cold surfaces, respectively. This formula readily lends itself to the calculation of k. The other quantities are readily measured in the above-described apparatus. Q is determined from the current in the heater wire and the potential drop per unit area of heater. The latter is readily calculated from the constants of the heater and the potential drops between the fixed leads previously mentioned. Half the heat generated must pass laterally through each sample. The distance d is the average thickness of each sample. The area A can be taken any desired value, preferably unity. This is possible only since the resistance of heating elements der unit length is constant, it being constantan and its temperature being practically constant. The temperatures t_2 and t_1 are the mean values of the temperatures for the hot and cold sides of both samples as determined by the thermocouples. The thermocouples (copper-constantan) were carefully calibrated so that the temperatures corresponding to the microvolts measured from the calibration curve could be readily obtained. One rather serious disadvantage of this method of measuring thermal conductivities is that it requires a long interval of time for the temperature equilibrium to be established, which is absolutely requisite in such measurements. For the apparatus used here no observations were ever taken under $3\frac{1}{2}$ hours' heating and most all measurements were taken after some 6 to 7 hours' heating. This latter time was quite sufficient for equilibrium of temperature distribution to be fully established.

13 The samples of the materials tested were usually composed of one or more sheets. Extreme care was taken to eliminate the air between the surfaces of the component sheets as much as possible by the use of vaseline, shellac, carpenter's glue, etc. By the use of such materials the drop between component sheets was made quite negligible or at least of the same order of magnitude as the drop through the same distance in the material itself. This was due to the fact that the thermal conductivity of such materials as glue, etc., when dried differs but little in order of magnitude from that of the sheet materials dealt with. In making up a sample that material was used to make good contact between the components which lent itself most readily to the case in question.

RESULTS OBTAINED BY THE THERMAL METER

14 Table 2 shows a typical set of results for samples made up of 7 sheets of 0.03-in. fuller board, 0.23 in. in thickness. By plotting the values of the mean temperatures as ordinates and the corresponding values of the thermal conductivity k as abscissæ it is possible to get a measure of the temperature coefficients of thermal conductivity. The values given in Table 2 if plotted will give a straight line whose equation is

$$k_t = k_0 (1 + at)$$

where k_t and k_0 are the thermal conductivities at temperatures t and 0, respectively, and a is the temperature coefficient. From the data give the value of a for this sample is found to be about 0.0030.

15 In the manner outlined above the thermal-conductivity measurements have been made for a large number of materials. The values obtained are recorded in Table 3 and are expressed both in calories per cm. per deg. cent. per sec. and in watts per in. per deg. cent. per sec. The values are also given for a definite temperature such as 20 deg. cent. as well as for an average range 20 deg. to 80 deg. The temperature coefficient is also recorded as determined

TABLE 2 THERMAL CONDUCTIVITY OF FULLER BOARD

Hot-side temp., t_2, deg. cent.	Cold-side temp., deg. cent	Mean temp., deg. cent.	Conductivity k, watts per in. per deg. cent. per sec.
29.15	18.10	23.62	0.00414
47.10	29.40	38.25	0.00432
59.45	37.15	48.30	0.00433
84.00	53.07	68.54	0.00458
102.60	65.05	83.82	0.00484
26.85	16.15	21.50	0.00407
40.87	24.20	32.54	0.00418
62.65	37.75	50.20	0.00436
115.20	70.30	92.75	0.00486

in Par. 14. It is to be observed that all samples did not show a temperature coefficient. This is doubtless due to the changing characteristics of the samples. Thus it is highly possible that the increase in the thermal conductivity due to an increase in temperature is counteracted by a corresponding increase in the thermal resistance due to the change in the surface contacts and likewise increase in air pockets. Besides measuring the thermal conductivity transversely for sheet material, measurements have been made as shown by Table 3 of the longitudinal conductivity along the laminations. The samples for this latter work were prepared for the one apparatus by winding up disks of the material 9 in. in diameter, and for the other apparatus by cutting the material into strips and then forcing them tightly together in a special press. By coating the edges of the strips while in the press with glue they could be held together in a square sample and later placed in the apparatus. It is interesting to note that the ratio of the longitudinal to the trans-

TABLE 3 THERMAL CONDUCTIVITIES

Material, thickness in in.	Thickness of sample, in.	Sp. gr.	Direction of heat flow	Temp., deg. cent.	Cal. per cm. per deg. cent. per sec. $\times 10^6$	Watts per in. per deg. cent. per sec. $\times 10^5$	Temp. coef. $\times 10^4$
FISH PAPER		1.06	Trans.	20	410	435	19
0.010	0.212			20-85	433	462	
0.010	0.748	1.06	Long.	20	1150	1222	19
				20-80	1215	1290	
0.023	0.222	1.03	Trans.	20	482	512	24
				20-85	517	548	
0.056	0.355	1.01	Trans.	20	567	602	21
PARAFFINED FISH PAPER		1.06	Trans.	20	460	489	18
0.007	0.211			20-80	483	513	
0.015	0.225	1.13	Trans.	20	520	553	
				20-90	525	558	
0.038	0.299	1.15	Trans.	15-30	494	525	
FULLER BOARD TREATED							
0.020, dark	0.230	1.39	Trans.	20	384	408	30
				20-90	418	444	
0.030, dark	0.500	1.15	Long.	20	1450	1540	50
				20-80	1650	1750	
0.056, green	0.227	1.09	Trans.	20	357	380	30
				20-100	396	421	
0.125, green	0.254	0.95	Trans.	20	339	361	16
				20-80	350	372	
FULLER BOARD UNTREATED							
0.015	0.232	1.38	Trans.	20	640	681	
				20-80	641	682	
0.30	0.216	1.26	Trans.	20	610	649	9
				20-80	628	667	
0.30	0.500	1.28	Long.	20	1500	1590	17
				20-80	1580	1680	
0.010	0.210	1.39	Trans.	20	622	661	10
				20-90	690	735	
0.056, light grey	0.217	1.15	Trans.	20	465	495	60
				20-60	515	548	
0.056, light grey	0.500	1.15	Long.	20	1520	1620	26
				20-80	1650	1750	
0.125, light grey	0.365	1.01	Trans.	20	347	369	33
				20-80	387	412	
FULLER BOARD							
0.125, light grey, soaked in transformer oil	0.365	1.01	Trans.	20	507	540	16
				20-80	543	577	
0.125	0.520	1.01	Long.	20-80	1230	1310	
VARNISHED CAMBRIC			Trans.	20	517	550	
0.009, tacky	0.263	1.17		20-95	544	578	10
0.009, tacky	0.694	1.17	Long.	20	1027	1093	
				20-100	1046	1113	5
0.009, dry	0.275	1.24	Trans.	20	516	549	
				20-90	532	565	9
CEMENT PAPER			Trans.	20	304	323	
0.015, plain	0.216	0.62		20-90	322	342	17
0.018, treated	0.211	1.02	Trans.	20	372	395	
				20-80	395	420	21
MICA TAPE							
0.006	0.201	1.06	Trans.	20-80	630	670	
0.008	0.229	1.12	Trans.	20-80	630	670	
0.006	0.768	1.03	Long.	20-80	3470	3680	
BLACK BIAS CLOTH							
0.009	0.205	1.26	Trans.	20-100	600	621	
0.009	0.782	1.26	Long.	20	915	975	
				20-100	1027	1091	50
CEMENT PAPER AND MICA							
No. 226	0.223	....	Trans.	20	443	472	
				20-80	462	491	14

TABLE 3 THERMAL CONDUCTIVITIES — *Continued*

Material, thickness in in.	Thickness of sample, in.	Sp. gr.	Direction of heat flow	Temp., deg. cent.	Cal. per cm. per deg. cent. per sec. $\times 10^6$	Watts per in. per. deg. cent. per sec. $\times 10^5$	Temp. coef. $\times 10^4$
CEMENT PAPER AND MICA — CONT.							
No. 227	0.1985	.....	Trans.	20	465	494	
				20–100	498	530	15
No. 247	0.225	.....	Trans.	20	501	533	
				20–80	522	555	16
No. 227	0.512	.....	Long.	20	2230	2370	
				20–80	2360	2510	20
MICARTA FOLIUM							
No. 249	0.233	.....	Trans.	20–100	553	588	
	0.569	.....	Long.	20–100	2700	2870	
KRAFT PAPER AND MICA							
No. 312	0.220	.....	Trans.	20–100	545	579	
	0.520	.....	Long.	20	2680	2830	
				20–100	2840	3020	1.
FISH PAPER AND MICA							
No. 230	0.195	.....	Trans.	20–100	483	514	
No. 232	0.233	.....	Trans.	20–100	475	505	
No. 233	0.237	.....	Trans.	20–100	451	481	
PRESSED MICA PLATES							
0.41 in. white	0.201	2.34	Trans.	20–100	623	663	
0.41 in. yellow	0.203	2.41	Trans.	20–100	550	585	
0.032 in. white	0.1915	2.32	Trans.	20–100	675	718	
0.032 in. yellow	0.1915	2.41	Trans.	20–100	580	617	
0.025 in. white	0.1995	2.43	Trans.	20–100	725	771	
0.025 in. yellow	0.1996	2.26	Trans.	20–100	612	650	
MICARTA	0.247	1.36	Trans.	20	606	645	
0.125 in.				20–90	620	660	6
HARD RUBBER	0.380	1.19	Trans.	25–50	380	404	
WHITE FIBER	0.383	1.22	Trans.	20	663	705	
				20–80	695	728	12
WOODS							
White Pine	0.519	0.45	Across grain	20–120	255	271	
White Pine	0.732	0.45	Along grain	30–80	613	652	
White Oak	0.516	0.60	Across grain	20–80	455	484	18
White Oak	0.754	0.60	Along grain	40–70	944	1003	
Maple	0.733	0.72	Along grain	20	1015	1078	
Maple	0.733	0.72	Along grain	20–80	1037	1100	8
Maple	0.508	0.72	Across grain	20–80	434	462	
ASBESTOS							
⅛ in. sheet	0.344	0.894	Trans.	22–80	395	420	
0.025 in. paper	0.306	0.98	Trans.	20	345	367	
				20–100	375	399	24
0.035 cloth	0.356		Trans.	20	666	708	
				20–80	685	728	14
Board	0.507	1.93	Trans.	20	1780	1890	
				20–90	1950	2080	14
PLATE GLASS	0.252	2.49	Trans.	20	1785	1900	
				20–100	1945	2070	18
	0.289	2.60	Trans.	20	1905	2024	
	0.289		Trans.	20–120	2016	2142	12
SOAPSTONE	0.715	2.87	Trans.	70–130	8000	8500	
SHEET STEEL							
0.0172 in M. A.	0.415	.....	Trans.	20	1370	1455	
Varnished				20–80	1430	1520	19
Unvarnished	1.48	.....	Long.	40–100	103000	109500	6
				40	101300	107700	
With asphalt	0.420	.....	Trans.	20	4710	5020	
paint on sheets			Trans.	20–80	4850	5160	10
Unvarnished	0.416	.....	Trans.	40	1480	1570	
0.0172 in.				40–100	1580	1680	25
Same with	0.425	.....	Trans.	40	6360	6750	
asphalt paint				40–100	6520	6930	9
W. A. SILICON STEEL							
0.014 in.	0.419	.....	Trans.	40–80	1270	1350	16
Varnished							
Unvarnished	1.44	.....	Long.	40–100	41800	44400	19
				40	39500	42000	
Same painted with asphaltum	0.422	.....	Trans.	40–80	4640	4920	10

verse conductivity is much greater for the mica combinations than for the other insulating materials. This is due to the influence of the mica, whose longitudinal conductivity is so much better than its transverse. The same ratio has its least value for such materials as varnished cambric and black bias cloth. For these materials there is less difference between the transverse and longitudinal construction than the mica compounds.

16 Results were also obtained for a number of granular and powdered materials. In order to make these measurements toroidal

TABLE 3 THERMAL CONDUCTIVITIES — *Continued*

Material, thickness in in.	Thickness of sample, in.	Sp. gr.	Direction of heat flow	Temp., deg. cent.	Cal. per. cm. per. deg. cent. per sec. × 10⁶	Watts per in. per deg. cent. per sec. × 10⁶	Temp. coef. ×10⁴
W. A. SILICON STEEL — CONT.							
Unvarnished	0.440	.....	Trans.	20	1340	1420	
				20–100	1470	1560	25
Same painted as above	0.443	.....	Trans.	20	4200	4470	
				20–100	4520	4810	17
SIL-O-CEL Brick	0.977	0.495	Trans.	30	242	258	15
				30–150	262	279	
Powdered	0.955	0.15		30	208	222	31
				30–150	242	258	
WOOL FELT dark grey	0.98	0.15	Trans.	40	149	159	76
				40–100	175	186	
GRAPHITE Solid	1.04	1.58		50	105500	112200	12
				50–130	110200	117200	
Powdered, through 20 mesh on 40 mesh	0.476	0.70		40	2850	3030	
				40–100	3200	3400	48
Powdered, through 40 mesh	0.476	0.42		40	922	980	
				40–100	1007	1080	40
Powdered, through 100 mesh	0.476	0.48		40	438	467	
				40–100	482	513	34
LAMPBLACK Eagle Brand Germantown	0.476	0.165		40	156	160	
				40–150	166	176	6
COAL DUST	0.476	0.73		30	265	282	
				30–150	298	317	23
IRON DUST AND SAND	0.377	1.14		30	460	489	23
				30–150	517	550	

rings were made of pine wood ½ in. wide, ½ in. thick and having an internal diameter slightly less than 9 in. The powdered materials were placed within these rings between sheets of paper glued to the sides of the rings. The thermocouples were attached adjacent to the material by means of shellac to the inner sides of the papers glued to the rings.

17 Results were also obtained for the transverse thermal conductivity of 0.0172-in. carbon sheet steel and for 0.014-in. silicon

sheet steel. The values obtained are slightly greater than those obtained by other observers for similar materials. Attempts were made to detect the change of transverse thermal conductivity of iron stampings with pressure, but the apparatus did not lend itself readily to this since the exact pressure applied could not be determined. Since this work has been done a testing machine has become available and a study will be made in the near future of the influence of pressure on the transverse thermal conductivity of iron stampings, and this work will be reported later. It is interesting to note that the transverse conductivity of iron stampings can be increased from 3 to 4½ times by coating the sheets with asphalt paint before putting them together. Consequently if the punchings in electrical apparatus could be assembled in groups, having a gum or other suitable material between the constituents so as to have better contact, the heat generated could be much more readily conducted away. This point will also be investigated when the work on iron stampings is resumed. The results for the longitudinal thermal conductivity of iron stampings were obtained by making up a form 12 in. by 12 in. by 1½ in. from strips 12 in. by 1½ in. of each of the two kinds of steel mentioned above. The strips were fastened tightly together by means of heavy bars and bolts into the form 12 in. by 12 in. by 1½ in. Both sides of these were ground smooth and the thermocouples inserted in small grooves in the faces, the thermocouple junction being actually pinched between the sheets of the material.

18 By comparing the values found for soapstone and 0.015-in. paraffined fish paper by this method it is seen that 1 in. of paraffined fish paper (made up of sheets) has a thermal resistance equal to 16 in. of soapstone. The same ratio was found to be 32 by means of Northrup's thermal bridge. The discrepancy is no doubt due to the loss of heat laterally in the bridge. This makes the slope of the curve, Fig. 3, much greater above the sample than below it. Consequently a larger ratio is obtained than would be the case were the temperature drop uniform along the apparatus. It is easy to show mathematically also that an apparatus such as that indicated in Fig. 1 should be at least five times as large in diameter as it is long in order to be little influenced by loss of heat from the sides. As can be seen, no such relations existed between the dimensions of the apparatus.

19 It is interesting to see the effect a layer of dust would have upon the internal temperature of a piece of apparatus. Thus

if a layer of cold dust is deposited upon a surface through which heat is passing the temperature within the surface will be raised 0.001/0.003 or $\frac{1}{3}$ deg. cent. for each watt of energy that passes through the surface in the form of heat. This is on the assumption that there is but little difference between the loss of heat from the two surfaces, which cannot be far from correct.

SUMMARY OF RESULTS

20 Experiments in the thermal conductivity of insulating and other materials have led to the following conclusions:

a The "thermal bridge" recommended by Professor Northrup has not been found satisfactory for determining the thermal conductivity of sheet materials

b Two "thermal meters," one of circular cross-section and the other of square cross-section, have been found entirely reliable for the measurement of the thermal conductivity of sheet and other materials

c By putting vaseline, glycerine, glue, shellac or a similar material on the division between two surfaces the thermal drop due to such division can be largely eliminated. This is particularly true for poor conductors

d The thermal conductivity has been measured for a large number of materials both across and along the laminations. For the poor conductors the ratio of the longitudinal to the transverse conductivity varies from 2 for black bias cloth to $5\frac{1}{2}$ for mica tape

e The temperature coefficient of thermal conductivity has been measured whenever the experimental results justified doing so

f Of the electrical insulating materials tested, those containing mica have in general the better thermal conductivity

g As a thermal insulator soft pine is the best of the woods tested and is but little inferior to dark-grey felt.

h The transverse conductivity of iron stampings can be increased some three or four times by the insertion of some suitable material between the stampings so as to make better thermal contact. This is for a pressure of about 50 lb. per sq. in. Nothing destroying the electrical insulation could be used, however. By using something between sheets the ratio of the longitudinal to transverse conductivity could be reduced to 20 to 25 instead of 80 to 100

i In general, the thermal conductivity of laminated products can be considerably increased by suitable impregnation so as to get rid of the air film

j Oil-soaking soft fuller board increases its thermal conductivity by about 50 per cent.

k To obtain the best thermal insulation for a given thickness it would be better to make it up of several thin sheets rather than of a single sheet. The effect is more pronounced for good conductors than for poor ones.

l Results were obtained for longitudinal conductivity of iron stampings. 0.0172-in. carbon sheet steel is about two and one-half times better than 0.014-in. silicon sheet steel. Carbon sheet steel has a longitudinal thermal conductivity about 80 times the transverse while silicon has but 32 times the transverse.

m A layer of dust, say, coal dust, upon the surface of a body will increase its internal temperature by $\frac{1}{8}$ deg. cent. per watt flowing through unit area.

21 The influence of pressure upon the thermal conductivity of sheet metal is to be studied soon and any further work on thermal conductivity will be done under definite pressures.

DISCUSSION

L. B. McMillan (written). The author has given considerable valuable information on the thermal conductivities of electrical insulating materials. Only a few heat-insulating materials are considered, however, and none of the more efficient ones.

In the case of electrical insulating materials high thermal conductivity is a desirable quality as it facilitates the dissipation of heat from electrical conductors, while obviously in the case of heat-insulating materials it is low conductivity that is desirable. Furthermore, the results are expressed in units convenient to the electrical but not to the mechanical engineer. Therefore it would seem that the paper would have been more likely to receive the attention of the engineers most interested had the title been more explicit.

No. 1713

EMERGENCY FLEET CORPORATION WATER-TUBE BOILER FOR WOOD SHIPS

BY F. W. DEAN, BOSTON, MASS.

and

HENRY KREISINGER, PITTSBURGH, PA.

Members of the Society

The following paper describes and reports tests upon the standard water-tube marine boiler designed by the United States Shipping Board Emergency Fleet Corporation. Thirteen hundred and fifty-two boilers were ordered, of which the first 706 were alike, after which 646 were ordered with slight changes including the use of four passes instead of three as employed in the original lot. In view of the large number of boilers needed, the scarcity of steel and the desire to secure competitive prices, the water-tube type was adopted instead of the Scotch marine type. This made it possible to have the boilers constructed in inland shops throughout the country and effected a reduction in the weight of steel of more than nine million pounds for the total order. The boilers had a grate area of 65.54 sq. ft., heating surface of 2500 sq. ft. and a commercial horsepower of 435 on the basis of the marine rating of 6 lb. of water to a square foot of heating surface per hour. Under test the three-pass type showed at first an efficiency of 60 per cent, which was later raised to about 73 per cent based upon combustible by certain changes which were effected. The four-pass type exceeded 74 per cent on two tests. These results were obtained when the boilers were hand-fired.

The Fuel Section of the Bureau of Mines made investigations of the process of combustion and of the temperature of gases as they flowed through the furnace and boiler. These investigations determined in a rational way what changes in the furnace and in the arrangement of the baffles were desirable to raise the efficiency of the boiler. Details of some of these investigations are given together with tables showing the complete results of the evaporative tests.

PART I DESCRIPTION OF BOILER AND RESULTS OF EVAPORATIVE TESTS

BY F. W. DEAN

IN 1917 the Emergency Fleet Corporation embarked upon a program of building a great fleet of wood ships for the purpose of quickly meeting a great emergency. Wood was selected not only

Presented at the Annual Meeting, December 1919, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

for accelerating delivery but for reducing the demand for steel which was needed for steel merchant and naval vessels.

2 At first it was intended to build 1000 wood ships of 3500 tons deadweight capacity each, and each ship would require two boilers containing about 2500 sq. ft. of heating surface each. The problem of securing 2000 good-sized boilers was a serious one and naturally caused careful consideration of the relative merits of water-tube and Scotch boilers. The natural inclination was to use Scotch boilers, but the boiler-making capacity of the country on the seacoast, or on waters tributary thereto, was insufficient to produce them in a reasonable time, to say nothing of the capacity of the mills of the country to produce the steel and steel plates required if Scotch boilers were used.

3 It was decided to use water-tube boilers, and to show the saving in steel caused by their adoption it will be sufficient to state that the weight of that metal in one of these boilers, with casing, is 41,200 lb., while for the equivalent Scotch boiler it is 110,000 lb. The program for 1000 wood ships was considerably reduced and instead of 2000 boilers, 1352 were ordered. The saving of steel for this number of boilers by the use of the water-tube rather than the Scotch type was more than 9,000,000 lb.

4 Furthermore, the adoption of the water-tube type rendered most of the boiler shops of the country available, at such inland places as Battle Creek, Mich., Chattanooga, Tenn., and Allentown, Pa., etc. All told, they were built by 19 different contractors.

5 Still further the competition coming from the adoption of the water-tube type, taking into consideration prices asked by some of the regular makers of marine water-tube boilers, resulted in a saving which is estimated to have been about $7,000,000 for the requirements of the program. While the Emergency Fleet Corporation has been accused of extravagance, credit for this piece of economy should be given to it, and this is due to the engineering department.

6 A great advantage of the design made by the Emergency Fleet Corporation came from the fact that all of the boilers for the wood ships using the first 706 boilers were alike and differed only slightly from the later orders of 646, which were all alike. The differences in the design came from changing the number of baffles from three to four and in using Key handhole caps instead of plugs with copper ferrules.

7 It is well to state at this point that the performance of the wood-ship standard boiler is most satisfactory in service.

DESCRIPTION OF THE BOILER

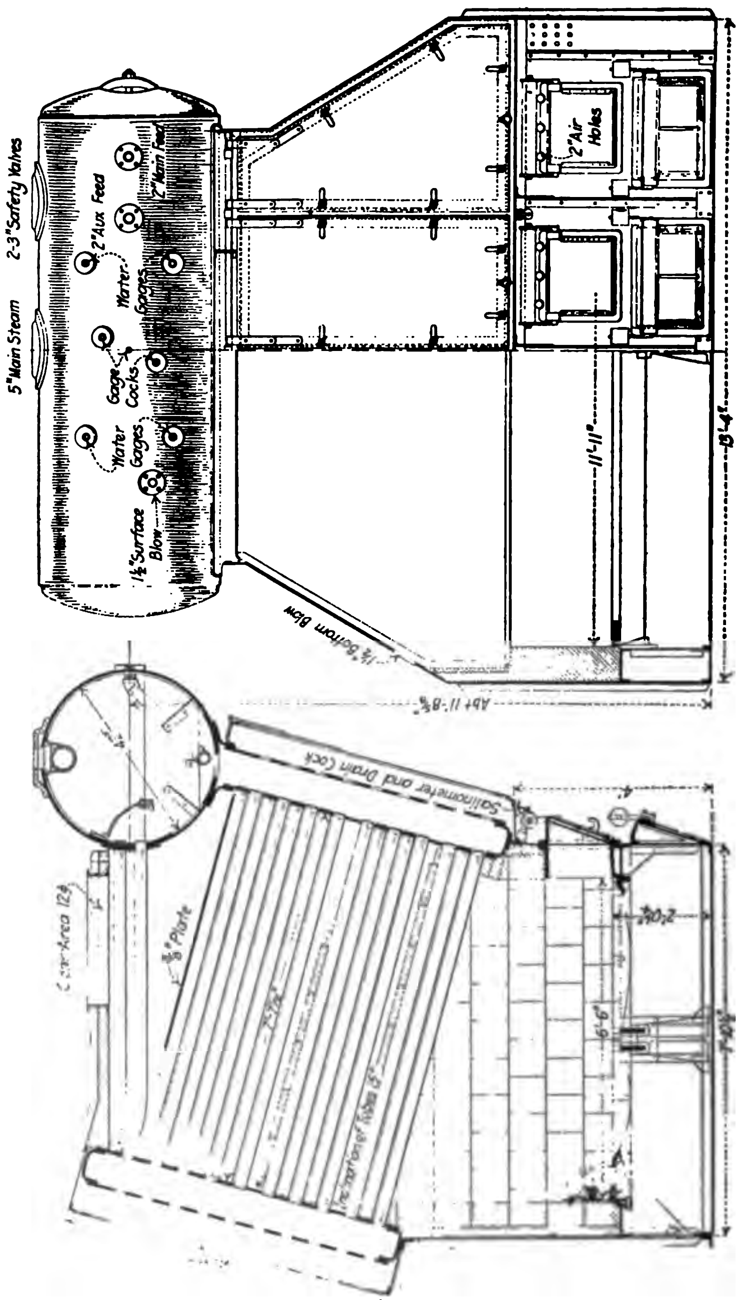
8 The boiler as first designed is shown in Fig. 1, and as later modified in Fig. 2. It consists of two headers, each composed of two plates, known as tube and handhole plates, connected at the edges by channel-shaped pieces, the two headers being connected by tubes which furnish most of the heating surface and form means of fastening the headers together. The front header is surmounted by a steam drum which is connected to the header by means of a flanged saddle which is riveted to both the drum and header. Holes in the bottom of the drum within the limits of the saddle furnish the means of connecting the interior of the drum and header, and two rows of so-called circulating tubes connect the upper part of the back header with the drum and serve to conduct the steam to it.

9 The header plates are stayed together by means of hollow iron staybolts, the holes being $\frac{3}{4}$ in. in diameter. The handholes in the first 706 boilers were closed by means of tapered plugs surrounded by thin copper ferrules. The plugs had a threaded shank secured by a yoke and nut as is usual for handhole plates.

10 While these plugs have in general given satisfactory service, it was soon recognized that the Key cap is better, and all boilers afterward ordered, amounting to 646, were provided with these caps.

11 The drum was made with the longitudinal joint where the circulating tubes enter, there being here an inside and an outside strap. The tubes pass through the shell and both straps, but the holes in the shell are larger than the tubes and the expanding occurs only in the straps. The drum is reinforced at the bottom by means of an inside strap in order to make up for the plate section cut away by the holes referred to.

12 The tubes are seamless hot-rolled steel. The baffles are of the longitudinal type, of which the first boiler tested had three, forming three passes as shown in Fig. 1. After some testing it was decided that it was best to place the lowest baffle on the lowest row of tubes in order to render the second and third rows of tubes more active as heating surface. In the first design these tubes did little good except such portion of them as extended across the first pass. When this change was being made it was seen that there was room for four passes and accordingly four baffles were inserted. This rendered the boiler somewhat more efficient, especially at the higher



One Boiler as drawn - Starboard. One Boiler to Other Hand - Port.

Fig. 1 Three-Pass Standard Water-Tube Marine Boiler for Wood Ships

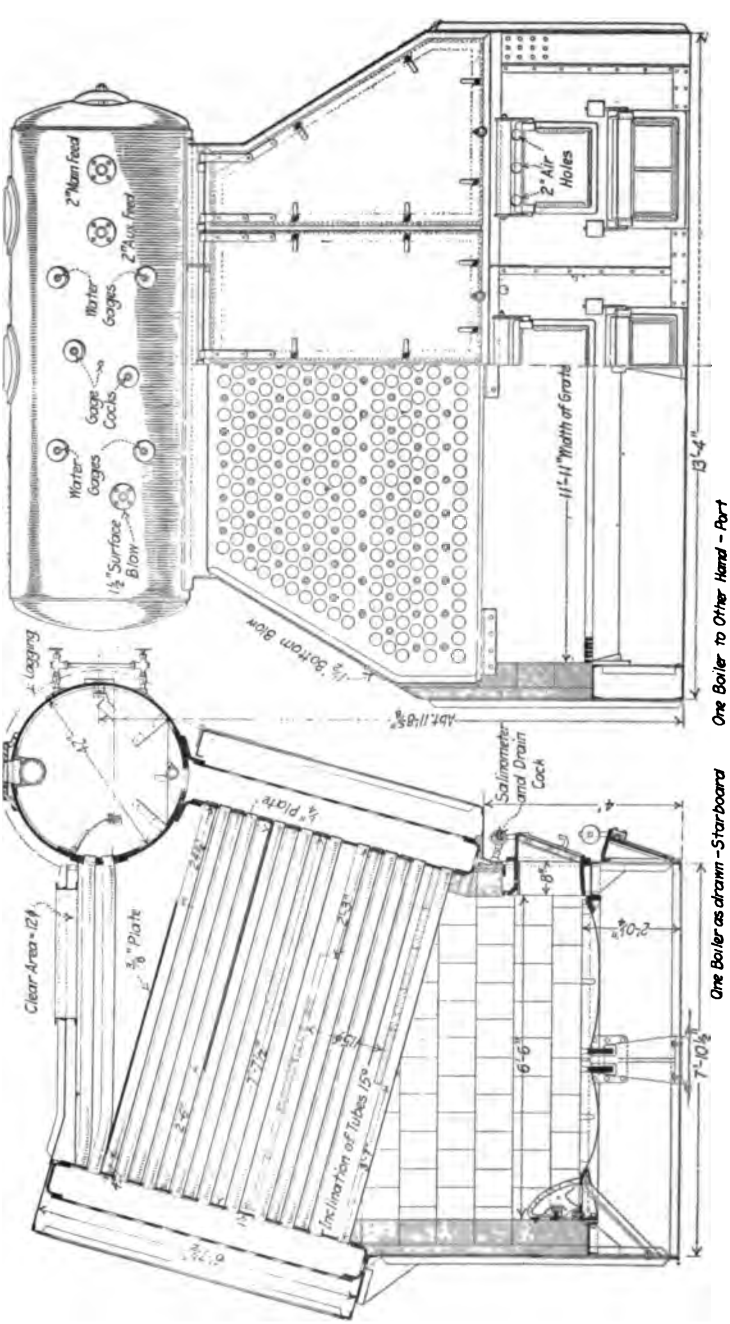


FIG. 2 FOUR-PASS STANDARD WATER-TUBE MARINE BOILER FOR WOOD SHIPS

channel base is bolted to the bottom of the ship and the casing holds the boiler firmly in place and is amply able to prevent movement from rolling or pitching at sea or in a collision, all of which have been proved by experience, including the latter, even without rolling or collision braces.

17 The front and back of the casing have sheet-iron doors covering the headers, and the space between the headers and doors is sufficiently great to accommodate soot blowers. These blowers, which are found to be of great importance, blow through the hollow staybolts.

18 The boilers are equipped with bottom and surface blows, a duplex safety valve, two water glasses, a set of three gage cocks, main and auxiliary feeds, a salinometer cock, a basket of zinc plates and an internal perforated feedpipe.

19 In the three-pass boiler there are four fire doors and four ashpit doors, but in the four-pass boiler there are three of each. These doors are of the in-swinging type. Above each door there is a hollow perforated lintel with air freely entering through holes in the casing, and on the sides of the doors there are perforated jambs with air supplies from the outside. All of these air openings are not only useful in protecting the parts from burning, but promote economy, for it is found that none too much air enters the furnace through them.

20 The grates are of the fixed type of double bars in two lengths, and have $\frac{1}{2}$-in. air and $\frac{1}{2}$-in. iron spaces.

21 Across the back end of the furnace there is a Wager bridge wall with narrow air spaces which saves nearly 2 per cent of coal, perceptibly diminished smoke, improved the gas analyses, saved the back brickwork and reduced the time and labor of cleaning the fires nearly 50 per cent.

DIMENSIONS OF BOILER

22 The following are the leading dimensions of the boiler:

Width of casing at floor level	13 ft. 4 in.
Length of casing at floor level	7 ft. $10\frac{1}{2}$ in.
Height of center of drum above floor	11 ft. $8\frac{5}{8}$ in.
Thickness of header plates	11/16 in.
Width of water spaces of headers	8 in.
Outside diameter of tubes	3 in.
Exposed length of tubes between headers	7 ft. $7\frac{1}{2}$ in.
Number of tubes between headers	388
Number of tubes between rear header and drum	21

Inside diameter of drum	42 in.
Thickness of drum plates	$\frac{5}{8}$ in.
Width of furnace	11 ft. 11 in.
Depth of Furnace	$6\frac{1}{2}$ ft.
Height of furnace at center	3 ft. 8 in.
Firing doors	15 in. by 18 in.
Width of grate	11 ft. 11 in.
Depth of grate without bridge wall	6 ft. 6 in.
Depth of grate with bridge wall	5 ft. 8 in.
Grate area without bridge wall	$77\frac{1}{2}$ sq. ft.
Grate area with bridge wall	$67\frac{1}{2}$ sq. ft.
Heating surface fire sides	2518 sq. ft.
Thickness of brick lining	$8\frac{1}{2}$ in.

The boiler tested was built by and tested at the Erie City Iron Works, Erie, Pa.

THE EVAPORATIVE TESTS

23 In contemplating these tests which, as before stated, were made for the purpose of studying the behavior of the boiler and, if possible, of improving it, it was recognized that the measurement of high temperatures would play an important part, and for this reason the services of the Bureau of Mines were sought. The Bureau very willingly agreed to assist, and assigned Mr. Henry Kreisinger, Member of the Society, to take charge of this part of the work. In time the field enlarged and he embraced in his work the gas analyses, the paths and velocities of the gases, the draft magnitudes and losses throughout the passes, the advisable quantity of air and position of its admission, as well as the temperature throughout the furnace and all parts of the passes. A description of his work will be given in his portion of the paper. The Bureau also determined the calorific values of the coal.

24 In the beginning it was decided to use Georges Creek Cumberland coal from the Big Vein Mine on all tests in order to have a standard coal of good and uniform quality, low volatile content, and high-fusing clinker.

25 It was also decided to make tests with fixed grates, shaking grates, firebox without a bridge wall, with the iron bridge wall already mentioned, a brick bridge wall covering the same area as the other, and several kinds of oil burners, using Mexican oil. Mexican oil was selected because that is the oil that will be used chiefly in the future.

26 The three-pass boiler was erected with the baffles in the positions and of the lengths in the original drawing, but the first

test showed that they were not sufficiently long. Several of the earlier tests were made with the grate of the full size, that is to say, without the iron or the brick bridge wall. By the addition of the iron bridge wall, admitting air around the fire doors, lengthening and otherwise changing the baffles, and studying the method of firing, the efficiency of the three-pass boiler was raised from about 60 per cent to about 71 per cent based upon dry coal, and to about 73 per cent based upon combustible. The efficiency of the four-pass boiler was about 72½ per cent based on dry coal, and based upon combustible exceeded 74 per cent on two tests. These tests were made when firing by hand, but when using the "Type E" stoker higher efficiencies were obtained.

RESULTS OF TESTS

27 Tables 1, 2, 3 and 4 show the general results of the tests of the three-pass and four-pass boilers under all of the conditions, both with hand and stoker firing. No results with oil firing are given because those tests are now in process.

28 While it was intended to test the boiler at the marine rating in general, which is to evaporate 6 lb. of water per sq. ft. of heating surface per hr., other rates were used, especially with the four-pass boiler. In Table 3, on Dec. 18, 1918, the four-pass boiler was worked at 131 per cent of marine rating and 229 per cent of land rating. Even at this high rate the efficiency was about 71 per cent, and but little below that of lower rates. With the stokers the boiler was operated at 162 per cent of marine rating and 282 per cent of land rating. At this rate the evaporation was 9.72 lb. per sq. ft. of heating surface per hr. from and at 212 deg., and the efficiency was 72.3 per cent based upon combustible. Based upon the total area of grate for hand firing the coal consumption was at the rate of 29.6 lb. per sq. ft. of grate per hr. The coal consumption per sq. ft. of heating surface per hr. on this test was 0.92 lb.

29 It will be observed that two of the tests given in Table 4 were of 24 hr. duration each, one of 22 hr., and one of 23 hr.

30 The conclusion arrived at is that the boiler is of excellent efficiency and that the four-pass boiler is well adapted to overloads. This is due to its having four passes. The only defect of the four-pass boiler is, as might be expected, that there is considerable absorption of draft in the passes, and the greatest loss was in the third pass from the bottom. The third baffle was lowered one tube, but

TABLE 1 RESULTS OF EVAPORATIVE TESTS OF THREE-PASS BOILER EQUIPPED WITH WAGER BRIDGE WALL AND HAND-FIRED

1	Date, 1918	June 12	June 13	June 14
2	Duration, hr.	10	10	10
	Dimensions and Proportions			
3	Grate area, sq. ft.	65.54	65.54	65.54
4	Heating surface, sq. ft.	2500	2500	2500
5	Ratio heating surface to grate area	38.20	38.20	38.20
	Average Pressures			
6	Gage pressure, lb.	179	178	179
7	Atmospheric pressure, lb.	14.28	14.41	14.41
8	Absolute pressure, lb.	193.28	192.41	193.41
9	Draft between damper and boiler, in.	0.53	0.58	0.63
	Average Temperature			
10	External air, deg. fahr.	59	64	64
11	Fire room, deg. fahr.	74	77	70
12	Feed water, deg. fahr.	204	209	207
13	Escaping gases, deg. fahr.	538	545	551
	Fuel			
14	Moist coal consumed per hour, lb.	1455	1411	1521
15	Moisture in coal per cent	2.19	2.39	2.02
16	Dry coal per hour, lb	1423	1380	1490
17	Dry refuse per hour weight, lb.	151	143	158
18	Dry refuse per cent,	10.60	10.35	10.65
19	Combustible per hour, lb.	1272	1237	1331
	Quality of Steam			
20	Moisture in steam, per cent	0.51	0.62	0.41
	Heat Value of Coal and Efficiency			
21	Heat value of a pound of dry coal, B.t.u.	14380	14996	14350
22	Efficiency of boiler based on dry coal, per cent	70.30	71.50	70.80
23	Efficiency of boiler based on combustible, per cent	72.00	73.20	73.00
	Water			
24	Water supplied to boiler per hr., lb.	14032	13822	14780
25	Dry steam generated, lb	13962	13836	14719
26	Factor of evaporation	1.057	1.062	1.064
27	Equivalent evaporation from and at 212 deg., lb.	14758	14685	15514
	Evaporative Performance			
28	Water evaporated per pound of dry coal, lb.	9.81	10.06	9.90
29	Equivalent from and at 212 deg., lb.	10.36	10.56	10.42
30	Water evaporated per pound of combustible, lb.	10.95	11.22	11.06
31	Equivalent from and at 212 deg., lb.	11.56	11.78	11.60
	Rate of Combustion			
32	Dry coal burned per sq. ft. of grate per hr., lb.	21.80	21.10	22.80
33	Dry coal burned per sq. ft. heating surface per hr., lb.	0.57	0.56	0.60
	Rate of Evaporation			
34	Water evap. from and at 212 deg. per sq. ft. h. s. per hr., lb.	5.90	5.80	6.20
35	Water evap. from and at 212 deg. per sq. ft. grate per hr., lb.	226	222	237
	Power of Boiler			
36	Commercial horse power for land use, hp.	428	422	450
37	Excess above commercial rating of 250 hp., per cent	71	69	80
38	Marine rating, wat. evap. from and at 212 deg. per hr., lb.	15000	15000	15000
39	Equivalent commercial hp. of marine rating, hp.	435	435	435
40	Excess above or below marine rating, per cent	1.80 below	3.20 below	3.20 above

TABLE 2 RESULTS OF EVAPORATIVE TESTS OF THREE-PASS BOILER EQUIPPED WITH WAGER BRIDGE WALL AND HAND-FIRED

1	Date, 1918	June 17	June 18	June 19	June 20
2	Duration, hr.	10	10	10	10
	Dimensions and Proportions				
3	Grate area, sq.ft.	65.54	65.54	65.54	65.54
4	Heating surface, sq. ft.	2500	2500	2500	2500
5	Ratio grate area to heating surface				
	Average Pressures				
6	Gage pressure, lb.	175.00	178.00	182.10	180.30
7	Atmospheric pressure, lb.	14.49	14.54	14.56	14.53
8	Absolute pressure, lb.	189.49	192.54	196.66	194.83
9	Draft between damper and boiler, in.	0.64	0.64	0.61	0.39
	Average Temperatures				
10	External air, deg. fahr.	74	68	58	61
11	Fire room, deg. fahr.	83	80	80	69
12	Feed water, deg. fahr.	211	212	211	216
13	Escaping gases, deg. fahr.	538	531	556	510
	Fuel				
14	Moist coal consumed per hour, lb.	1478	1436	1530	1274
15	Moisture in coal, per cent	1.76	2.02	2.09	2.04
16	Dry coal consumed per hour, lb.	1450	1406	1498	1248
17	Dry refuse per hour, lb.	152	138	132	109
18	Dry refuse in per cent	10.50	9.80	8.80	8.75
19	Combustible consumed per hour, lb.	1298	1268	1366	1139
	Quality of Steam				
20	Moisture in steam, per cent	0.40	0.40	0.30	0.30
	Heat Value of Coal and Efficiency				
21	Heat value of one pound of dry coal, B.t.u.	14409	14237	14441	14251
22	Efficiency of boiler based on dry coal, per cent	69.80	70.80	67.30	68.60
23	Efficiency of boiler based on combustible, per cent	72.00	71.50	68.30	69.00
	Water				
24	Water supplied to boiler per hour, lb.	14346	13915	14242	12094
25	Dry steam generated per hour, lb.	14284	13859	14196	12056
26	Factor of evaporation	1.049	1.048	1.050	1.045
27	Equivalent evaporation from and at 212 deg. per hour, lb.	14984	14524	14906	12599
	Evaporative Performance				
28	Water evaporated per pound of dry coal, lb.	9.87	9.84	9.50	9.65
29	Equivalent from and at 212 deg., lb.	10.32	10.33	9.96	10.02
30	Water evaporated per pound of combustible, lb.	11.00	10.90	10.40	10.55
31	Equivalent from and at 212 deg., lb.	11.56	11.46	10.92	11.05
	Rate of Combustion				
32	Dry coal burned per sq. ft. of grate per hour, lb.	21.10	21.50	22.90	19.00
33	Dry coal burned per sq. ft. of heating surface per hour, lb.	0.58	0.56	0.60	0.50
	Rate of Evaporation				
34	Water evap. from and at 212 deg. per sq. ft. h. s. per hr., lb.	5.94	5.81	5.96	5.04
35	Water evap. from and at 212 deg. per sq. ft. grate per hr., lb.	229	222	228	192
	Power of Boiler				
36	Commercial horse power for land use, hp.	434	421	432	365
37	Excess above commercial rating of 250 hp., per cent	74	68	73	46
38	Marine rating in water evap. per hr. from and at 212 deg., lb.	15000	15000	15000	15000
39	Equivalent commercial hp. of marine rating, hp.	435	435	435	435
40	Excess above or below marine rating, per cent	0.20 below	3.20 below	0.70 below	15 below

TABLE 3 RESULTS OF EVAPORATIVE TESTS OF FOUR-PASS BOILER EQUIPPED WITH WAGER BRIDGE WALL AND HAND-FIRED

1	Date, 1918	Dec. 19	Dec. 16	Dec. 17	Dec. 18
2	Duration, hr.	12.01	7.97	10.07	8.27
	Dimensions and Proportions				
3	Grate area, sq. ft.	66.53	66.53	66.53	66.53
4	Heating surface, sq. ft.	2500	2500	2500	2500
5	Ratio grate area to heating surface	37.58	37.58	37.58	37.58
	Average Pressures				
6	Gage pressure, lb.	200.50	200.00	197.30	198.60
7	Atmospheric pressure, lb.	14.67	14.67	14.68	14.70
8	Absolute pressure, lb.	215.17	214.67	211.98	213.30
9	Draft between boiler and damper, in.	0.34	0.44	0.78	1.56
	Average Temperatures				
10	External air, deg. fahr.	49	37	36	41
11	Fire room, deg. fahr.	63	58	57	58
12	Feed water, deg. fahr.	207.80	184.50	206.00	190.90
13	Escaping gases, deg. fahr.	506	513	557	640
	Fuel				
14	Moist coal consumed per hour, lb.	1032	1149.70	1443.70	1934
15	Moisture in coal per cent	1.49	2.15	1.54	1.91
16	Dry coal per hour, lb	1017	1125	1411.50	1897
17	Dry refuse per hour weight, lb.	111	168	143.40	166.90
18	Dry refuse per cent	10.95	13.45	10.15	8.80
19	Combustible per hour, lb.	906	957	1268.10	1730.50
	Quality of Steam				
20	Moisture in steam, per cent	0.61	1.07	0.62	0.54
	Heat Value of Coal and Efficiency				
21	Heat value of one pound of dry coal, B.t.u.	14292	13990	14215	14216
22	Efficiency of boiler based on dry coal, per cent	72.60	72.80	71.00	70.90
23	Efficiency of boiler based on combustible, per cent	74.30	76.60	72.30	71.00
	Water				
24	Water supplied to boiler per hour, lb.	10349	11074	14042	18399
25	Dry steam generated per hour, lb.	10286	10956	13858	18408
26	Factor of evaporation	1.053	1.077	1.065	1.078
27	Equivalent evaporation from and at 212 deg. per hour, lb.	10831	11800	14728	19758
	Evaporative Performance				
28	Water evap. per pound dry coal, lb.	10.10	9.78	9.80	9.75
29	Equivalent from and at 212 deg., lb.	10.70	10.50	10.40	10.48
30	Water evap. per pound combustible, lb.	11.30	11.46	11.00	10.62
31	Equivalent from and at 212 deg., lb.	11.95	12.30	11.61	11.48
	Rate of Combustion				
32	Dry coal burned per sq. ft. grate per hour, lb.	15.30	16.90	21.30	28.60
33	Dry coal burned per sq. ft. heating surface per hour, lb.	0.41	0.46	0.57	0.76
	Rate of Evaporation				
34	Water evap. from and at 212 deg. per sq. ft. h. s. per hr., lb.	4.30	4.72	5.90	7.90
35	Water evap. from and at 212 deg. per sq. ft. grate per hr., lb.	163	177.50	222	297
	Power of Boiler				
36	Commercial horse power for land use, hp.	314	342	427	573
37	Excess above commercial rating of 250 hp., per cent	26	37	71	129
38	Marine rating in water evap. from and at 212 deg. per hr., lb.	15000	15000	15000	15000
39	Equivalent commercial hp. marine rating, hp.	435	435	435	435
40	Excess above or below marine rating, per cent	28 below	22 below	2 below	31 above

TABLE 4 RESULTS OF EVAPORATIVE TESTS OF FOUR-PASS BOILER EQUIPPED WITH TWO TYPE "E" UNDERFEED STOKERS

1	Date, 1919	April 8	April 9	April 13	April 12
2	Duration, hr.	23.	24	22	24
	Dimensions and Proportions				
3	Grate area, sq. ft.	77.50	77.50	77.50	77.50
4	Heating surface, sq. ft.	2500	2500	2500	2500
5	Ratio grate area to heating surface	32.26	32.26	32.26	32.26
	Average Pressures				
6	Gage pressure, lb.	199.00	199.00	197.00	198.40
7	Atmospheric pressure, lb.	14.51	14.51	14.50	14.44
8	Absolute pressure, lb.	213.51	213.51	211.50	212.84
9	Draft between damper and boiler, in.	0.86	0.86	1.40	1.43
	Average Temperatures				
10	External air, deg. fahr.	47	58	50	44
11	Fire room, deg. fahr.	61	68	63	62
12	Feed water, deg. fahr.	66	67	66	63.40
13	Escaping gases, deg. fahr.	536	591	618	659
	Fuel				
14	Moist coal consumed per hour, lb.	1193	1614	2000	2392
15	Moisture in coal, per cent	3.44	2.81	3.36	4.15
16	Dry coal consumed per hour, lb.	1152	1569	1933	2293
17	Dry refuse per hour, lb.	127	161	184	215
18	Dry refuse in per cent	11.00	10.30	9.50	9.40
19	Combustible consumed per hour, lb.	1025	1408	1749	2078
	Quality of Steam				
20	Moisture in steam, per cent	0.75	0.75	0.85	1.02
	Heat Value of Coal and Efficiency				
21	Heat value of one pound of dry coal, B.t.u.	14517	14312	14234	14405
22	Efficiency of boiler based on dry coal, per cent	77.40	75.10	72.40	71.40
23	Efficiency of boiler based on combustible, per cent	79.70	76.60	73.10	72.30
	Water				
24	Water supplied to boiler per hour, lb.	11040	14588	17253	20390
25	Dry steam generated per hour, lb.	10967	14479	17106	20182
26	Factor of evaporation	1.201	1.200	1.200	1.204
27	Equivalent evaporation from and at 212 deg. per hour, lb.	13199	17374	20527	24299
	Evaporative Performance				
28	Water evaporated per pound of dry coal, lb.	9.51	9.23	8.86	8.80
29	Equivalent from and at 212 deg., lb.	11.51	11.07	10.62	10.60
30	Water evaporated per pound of combustible, lb.	10.69	10.21	9.78	9.71
31	Equivalent from and at 212 deg., lb.	12.84	12.34	11.74	11.69
	Rate of Combustion				
32	Dry coal burned per sq. ft. grate per hour, lb.	14.86	20.22	24.90	29.52
33	Dry coal burned per sq. ft. heating surface per hr., lb.	0.46	0.63	0.77	0.92
	Rate of Evaporation				
34	Water evap. from and at 212 deg. per sq. ft. h. s. per hr., lb.	5.26	6.94	8.21	9.72
35	Water evap. from and at 212 deg. per sq. ft. grate per hr., lb.	169.80	224.30	264.90	313.60
	Power of Boiler				
36	Commercial horse power for land use, hp.	381	504	595	704
37	Excess above commercial rating of 250 hp., per cent	52	101	138	182
38	Marine rating in water evap. per hr. from and at 212 deg., lb.	15000	15000	15000	15000
39	Equiv. commercial hp. of marine rating, hp.	435	435	436	436
40	Excess above or below marine rating, per cent	12 below	16 above	37 above	62 above

the great loss continued, and it was then restored to its former position.

31 Among the experiments tried with this boiler were those to determine whether the number of circulating tubes and the number of holes in the bottom of the drum connecting its water space to that of the front header were excessive. Half of them, together and separately, were closed, but no effect could be observed. The conclusion was that an unnecessary number of such tubes and openings are used in this type of boiler, and by reducing them the drum will be made a safer structure.

32 The quality of the steam was most satisfactory when the water was carried at a proper level, and it was necessary to carry it near the top of the glass before the limit of a throttling calorimeter was reached. The quality of the steam is given in all of the tables.

PART II SPECIAL INVESTIGATIONS

By Henry Kreisinger

THIS part of the paper presents some of the results of special investigations conducted by the Fuel Section of the Bureau of Mines in connection with the tests of the three-pass and four-pass marine boilers shown in Figs. 1 and 2. Outline diagrams illustrating the methods of testing these boilers are given in Figs. 3 and 4. The special investigation consisted of the study of combustion in the furnaces and the temperature of gases as they flow through the boiler.

34 The study of combustion was carried on by the analysis of samples of furnace gases collected at various points in the setting. The gases were sampled with water-cooled sampling tubes and collected into glass holders over mercury. A continuous stream of gas was drawn through the sampling tubes by a steam ejector, and only a small part of the stream was collected in the gas holder. Each group of samples was collected simultaneously over a period varying from 15 to 40 min. The samples collected at the base of the stack were analyzed with an Orsat for CO_2, O_2, and CO. The gas samples taken in the furnace were analyzed, by Hempel's method, for CO_2, O_2, CO, H_2, CH_4, and unsaturated hydrocarbons.

35 Fig. 3 is a diagram of the three-pass boiler as originally designed. After the first few tests the baffles were extended, making smaller the gas passages between the ends of the baffles and the

water legs, as indicated in Fig. 8. Other changes were also made as the investigation indicated the need for them. The points at which gas samples were collected are indicated by the small circles designated by the capital letters *A*, *A′*, *B*, *C*, *D*, *E*, *F*, *G*, *H*, and *I*.

36 During the first six tests a large amount of combustible gas passed out of the furnace and either burned at the base of the stack, causing high flue-gas temperature, or escaped unburned. It was apparent that not enough air was admitted over the fuel bed,

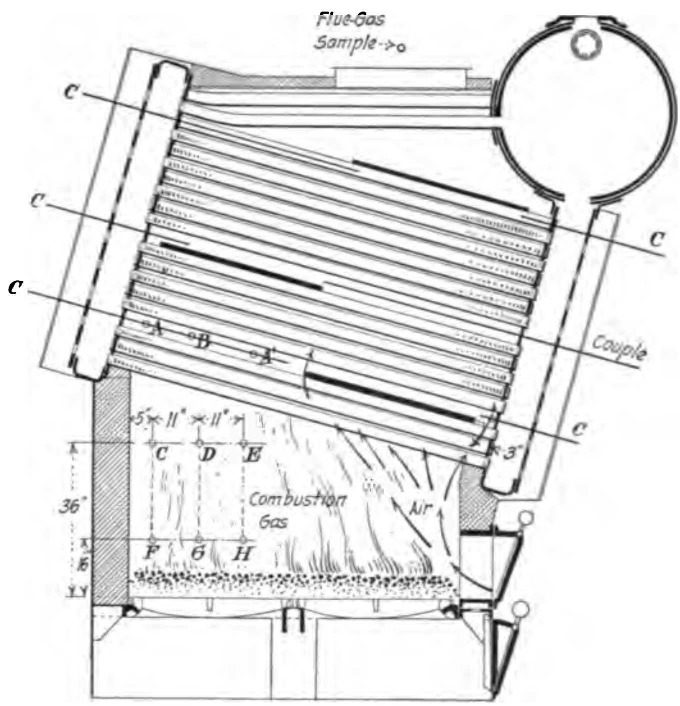


Fig. 3 Diagram Showing Method Used in Testing Three-Pass Type of Boiler

and that the air which did find its way into the furnace through the firing door was not sufficiently mixed with the combustible gases to cause them to burn completely in the furnace.

37 Some of the air admitted into the furnace through the firing door found its way out of the furnace through the opening at *I*, between the first baffle and the front water leg, as indicated by the arrow in Fig. 3. A large part of the air also passed around the end of the first baffle without mixing thoroughly with the combus-

tible gases rising from the fuel bed. As the gases and air passed through the boiler they formed a better mixture which tended to burn at the base of the stack. Evidently the combustion space was too small and the path of the gases and the air through the furnace too short for the two to form an intimate mixture.

ADMISSION OF AIR OVER FUEL BED

38 It was apparent that more air had to be admitted over the fuel bed, and better means provided for mixing it with the combustible. It is a proven fact that in a hand-fired furnace, when the fuel bed is level and 5 in. thick, no free oxygen can be forced through it, no matter what air pressure is used in the ashpit. The gases rising from the fuel bed contain practically no free oxygen and a large percentage of combustible gas. It is true that if the fuel bed contains holes or thin spots, air passes through them. However, admission of air through the holes in the fire is undesirable because the size of holes cannot be controlled. The holes are large before firing, when a small quantity of air is needed over the fuel bed, and are nearly absent immediately after firing, when a large quantity of air is needed to burn the volatile matter from the freshly fired coal. It is therefore much better to admit air over the fuel bed through special openings in the firing door or a bridge wall. The quantity of air admitted through such openings is nearly constant and can be controlled more easily. The air should be supplied in a large number of small jets and as close to the fuel bed as practicable, so that as much as possible of the combustion space above the fuel bed may be utilized for mixing and combustion.

39 Following these principles a number of ½-in. holes were made in each firing door; the small opening in the first baffle at *I* was closed and the baffle extended, making the gas passage between the end of baffle and the rear water leg 36 in. This had the effect of causing the air admitted through the firing doors to flow farther to the rear of the furnace and facilitate better mixing with the combustible gases. In addition to these changes, the Wager bridge wall was installed to supply additional air to the rear part of the furnace. The method of installing the Wager bridge wall is shown in Fig. 4. The bridge wall consists of a large number of cast-iron bars placed against the rear wall of the furnace, and forms a structure similar to [illegible] grate. The air passes into the furnace through narrow slots between the bars, and is regulated by the

thickness of the fuel bed near the bridge wall. The thicker the fuel bed the greater is the area of the air spaces covered with coal, and the smaller is the quantity of air flowing into the furnace. The air enters in a large number of thin streams.

40 With the air admitted through the firing door and through the bridge wall, the combustible gases rising from the fuel bed are squeezed between two streams of air coming in from two different

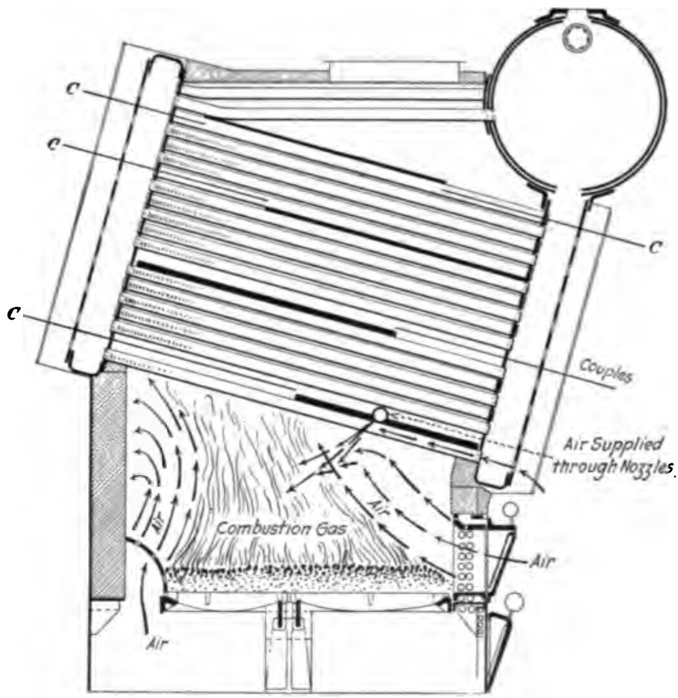


FIG. 4 DIAGRAM SHOWING FOUR-PASS TYPE OF BOILER WITH FACILITIES FOR INTRODUCING AIR OVER FUEL BED

directions and the mixing is greatly aided. The direction of the streams of air admitted over the fuel bed under these conditions is indicated in Fig. 4.

OPENINGS FOR ADMITTING AIR OVER FUEL BED

41 In the front of the furnace there were a total of 240 half-inch holes located in the fire doors, the cast-iron arches over the doors, and the cast-iron columns supporting the arches. The total area of these holes is about 48 sq. in. In addition to this there was

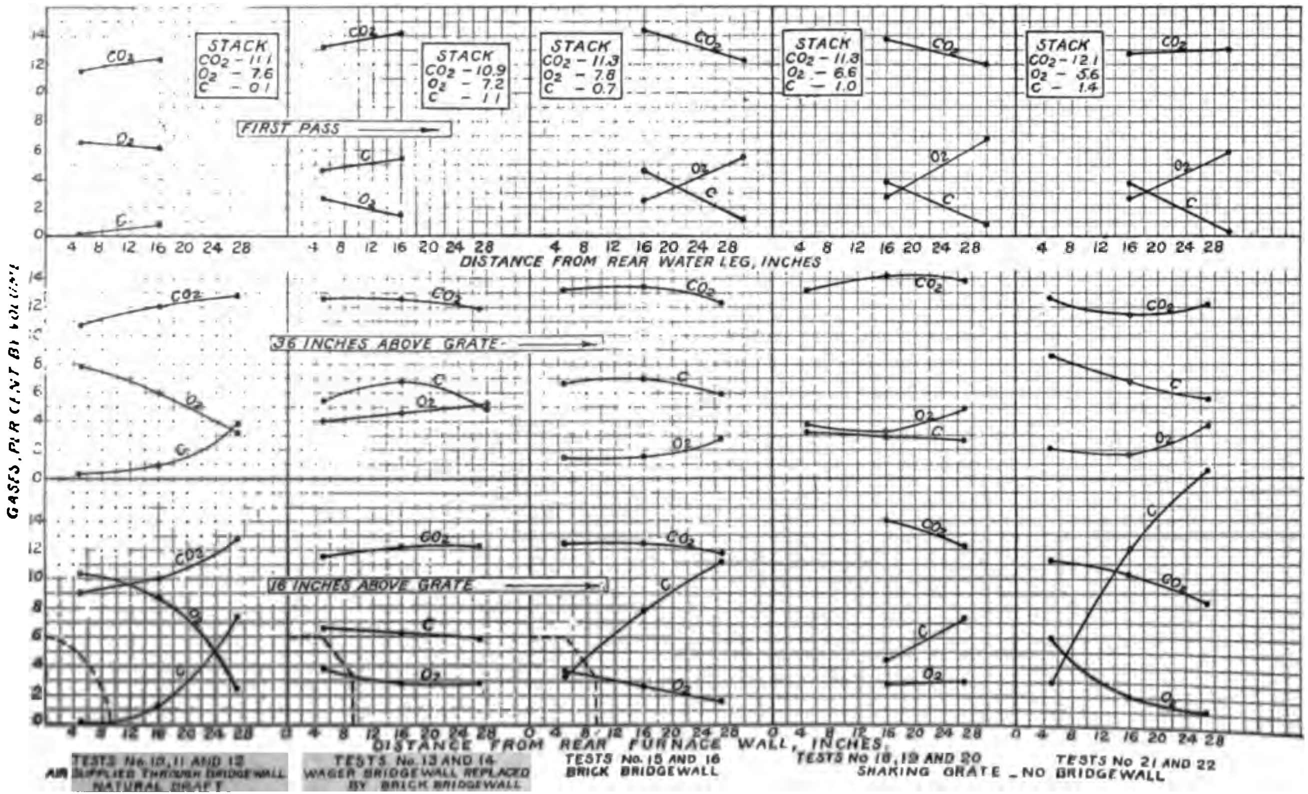


Fig. 5 Composition of Gases in Furnace with Various Methods of Introducing Air over Fuel Bed and the Effectiveness of Mixing. Three-Pass Water-Tube Boiler. In this Figure C Represents Combustible Gases Consisting of CO, H_2, CH_4 and Unsaturated Hydrocarbons.

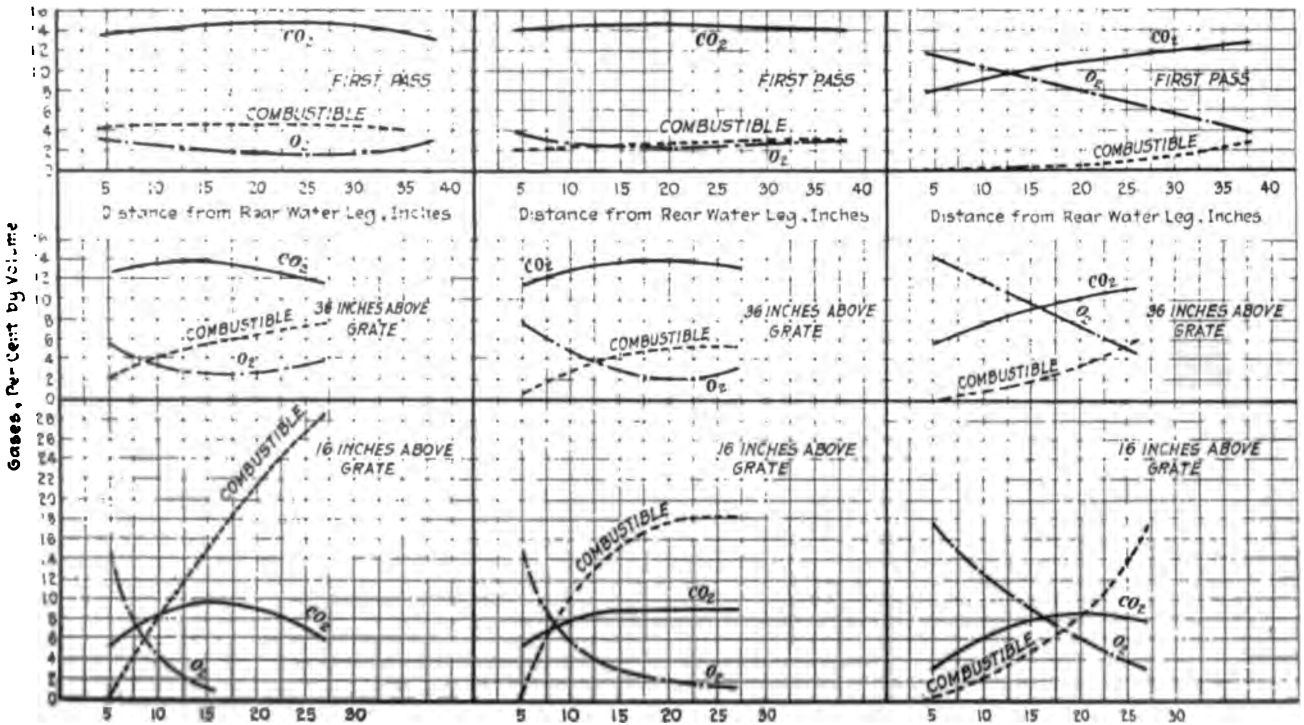
still containing about 5 per cent of combustible, and in the third group they contained 3 per cent of combustible gas. It should be noted, however, that in the third group of tests the samples collected in the first pass were taken farther away from the rear wall, and their analyses show the presence of the air admitted through the firing doors, the sample taken 30 inches from the rear water leg showing more free oxygen than the sample taken only 16 in. from it.

49 The tests of the fourth and fifth groups were made with a shaking grate and no bridge wall of any description. The samples taken 16 in. above the grate indicate that a small quantity of air passed into the furnace close to the rear wall, but it was not sufficient to have much effect on the combustion. The percentage of combustible remained high even when the gases reached the first pass. In these two groups of tests the samples taken in the first pass 30 in. from the rear water leg show the effect of the air admitted through the firing door. The stack samples show over one per cent of combustible gas and lower oxygen than either of the three previous groups. The lack of proper admission of air over the fuel bed is clearly indicated in these last two groups of tests. The advantage of air admission through the Wager bridge wall was clearly demonstrated and was therefore used on all subsequent tests with hand-fired furnaces.

EFFECT ON COMBUSTION OF DIFFERENT METHODS OF ADMITTING AIR OVER FUEL BED

50 Fig. 6 shows in a similar way the effect on combustion of various methods of admitting air over the fuel bed on tests made on the four-pass standard boiler illustrated in Fig. 4. The samples were taken similarly to those on the tests of the three-pass boiler, the points at which they were taken being also indicated by the abscissæ and label in each square.

51 The first group of tests was made with the air supplied over the fuel bed, through the firing door, and through the Wager bridge wall. Sixteen inches above the grate the admission of air through the bridge wall was apparent only in the sample taken 5 in. from the rear wall. Samples taken 36 in. above the grate and in the first pass indicated fairly uniform distribution of the air but an insufficient quantity of it. Apparently during these tests the fuel bed near the bridge wall was carried too thick. Had a thinner fuel bed been



Distance from Rear Furnace Wall, Inches.
Analysis of Furnace Gases.
Tests No. 6 and No. 7.
Jan 19 to 20.
Air Supplied by Draft through Bridge Wall

Distance from Rear Furnace Wall, Inches
Analysis of Furnace Gases
Tests No. 8 to No. 11 inclusive
Jan 22 to 25
Air injected into Furnace from Nozzles above First Row of Tubes

Distance from Rear Furnace Wall, Inches
Analysis of Furnace Gases
Tests No. 12 and No. 13
Jan 27 to 28
Air under Pressure Supplied through Bridge Wall

FIG. 6 COMPOSITION OF GASES IN FURNACE WITH VARIOUS METHODS OF INTRODUCING AIR OVER FUEL BED AND THE EFFECTIVENESS OF MIXING, FOUR PASS, WATER-TUBE BOILER carried the combustible in the first pass would have been much lower.

52 The second group of tests was made with the air over fuel bed being supplied with natural draft, through the Wager bridge wall, and through the firing doors, and in addition to this also under pressure of 2 in. of water through 17 half-inch nozzles from a 3½-in. pipe placed between the first and second rows of tubes, as indicated in Fig. 4. The object of the air forced in through these nozzles was to force the air supplied through the firing door into the center of the stream of combustible gas rising from the fuel bed and cause intimate mixing. The action of these nozzles is shown in Fig. 4. The samples collected in the first pass showed fairly uniform mixture and lower percentage of combustible than was obtained when the air was admitted only through the bridge wall and through the firing door. However, the amount of combustible was a little too high, indicating that not quite enough air was admitted.

53 The third group of tests was made with air admitted through the firing door and with natural draft, and air admitted through the Wager bridge wall under a pressure of ⅜ in. of water. In this case the air admitted through the bridge wall tended to flow in a separate stream close to the rear wall without penetrating very deep into the stream of combustible gas. The analyses of the samples taken in the first pass showed that the excess of air decreases as the distance from the rear water leg increases. On the whole, the admission of air through the bridge wall under pressure proved to be undesirable in this type of furnace. The tests represented by the second group showed conditions most favorable to complete combustion. However, the installation of the nozzles entails undesirable complications in boiler plants of ships. The conditions represented by the first group recommend themselves by their simplicity and effectiveness when attention is given to the proper thickness of fuel bed next to the bridge wall.

TEMPERATURE MEASUREMENTS

54 The temperatures were measured with thermocouples inserted into the setting through the hollow staybolts and moved across the gas passages, as indicated by the straight lines labeled "couples", or C, in Figs. 3 and 4. The temperatures were measured across the spaces between the ends of the baffles and the water legs. These spaces are called in this paper the first, second, third, and

fourth pass, respectively, counting from the lowest baffle. The couples were made of No. 22 gage and were mounted in $\frac{3}{8}$-in. iron pipe with the hot junction left exposed to the gases in order to reduce radiation error. All couples were connected to a central switchboard and the temperature was read with a portable potentiometer. On the first 17 tests two couples were used in each pass,

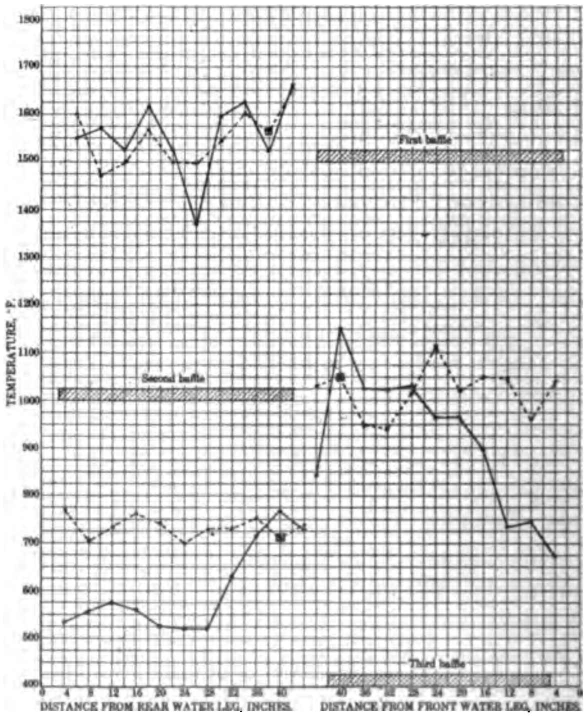


FIG. 7 TEMPERATURE OF GASES ACROSS THE SPACE BETWEEN THE END OF BAFFLES AND THE WATER LEGS. THREE-PASS BOILER, SHORT BAFFLES

both being placed approximately in the center of the boiler. The hot junction of one of those couples was placed 12 in. from the end of the baffle and was held in this position while the other was moved across the pass and readings taken every 6 in. Thus the stationary couple showed the temperature variation due to varying furnace conditions and the moving couple showed the variation in temperature due to the position of the couple.

RESULTS OF TEMPERATURE MEASUREMENTS ON THREE-PASS BOILER

55 Fig. 7 shows graphically the results of such temperature measurements made on Test No. 2, which was made on the three-

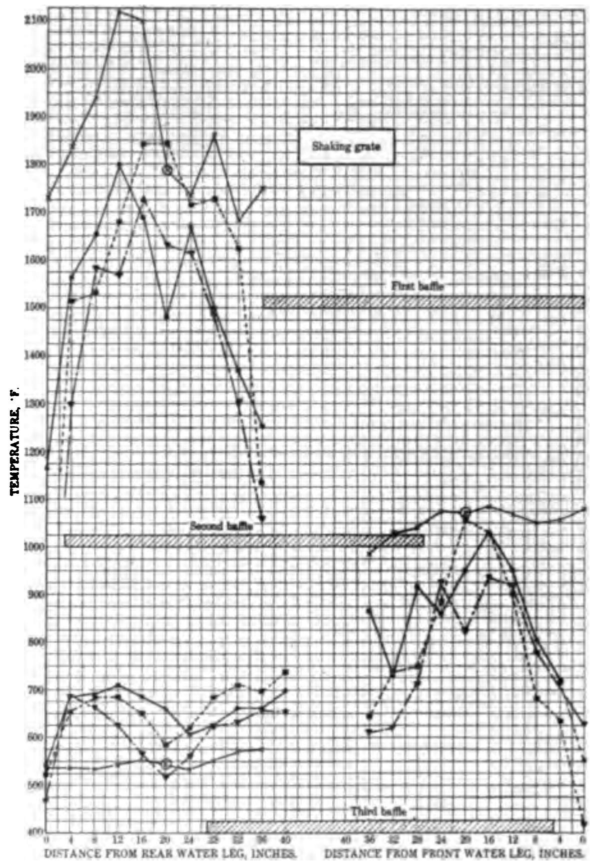


FIG. 8 TEMPERATURE OF GASES ACROSS THE SPACE BETWEEN THE ENDS OF BAFFLES AND THE WATER LEGS. THREE-PASS BOILER, LONG BAFFLES

pass boiler with the original arrangement of the baffle. The length and the position of the baffles with respect to the water legs are indicated in the figure. The solid line connects the points giving the reading of the moving couple, and the dotted line gives the reading of the stationary couple taken at the same time the moving couple was read in each position. Thus, the dotted line shows the variation of temperature due to varying furnace conditions and the solid line gives the temperature due to different position of the moving couples. In order to obtain approximately the correct reading of the moving couple in each position, allowance must be made for the varying temperature due to furnace conditions. Be-

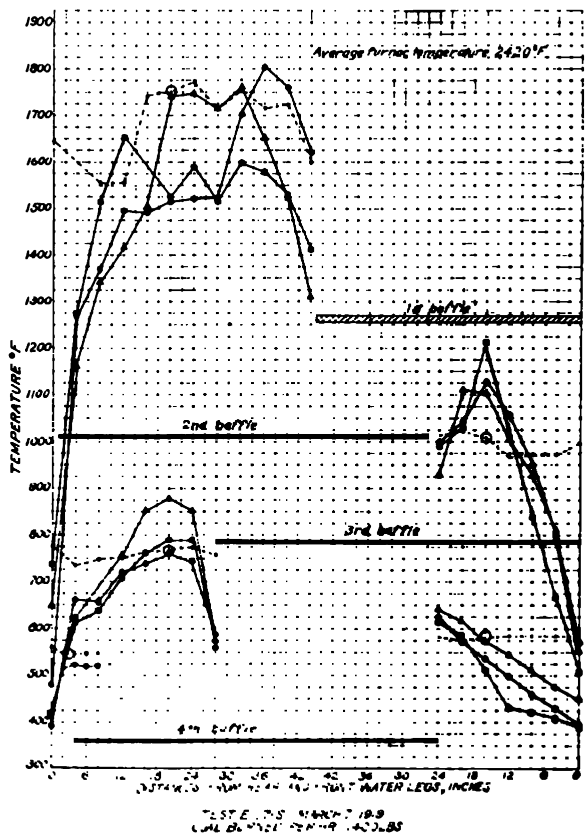


Fig. 9. Temperature of Gases Across the Space Between the Ends of Baffles and the Water Legs. Four-Pass Boiler. Four Long Baffles

cause the temperatures of the first pass are the highest they are placed at the top of the chart, and similarly, because the temperatures in the fourth pass are the lowest they are placed at the bottom of the chart. Therefore in studying these charts it should be kept in mind that the passes would be represented thereon from the top to the bottom.

56 The first pass shows no regular variation in temperature due to position of the moving couple. This indicates that the distribution of the hot gases through this pass is fairly uniform. The second pass, however, shows the highest temperature one foot from the end of the baffle and the temperature drops rapidly each side of this point. This seems to indicate that a large part of the hot gases pass through the space between 6 in. to 24 in. from the end of the baffle. The remaining space of about 24 in., next

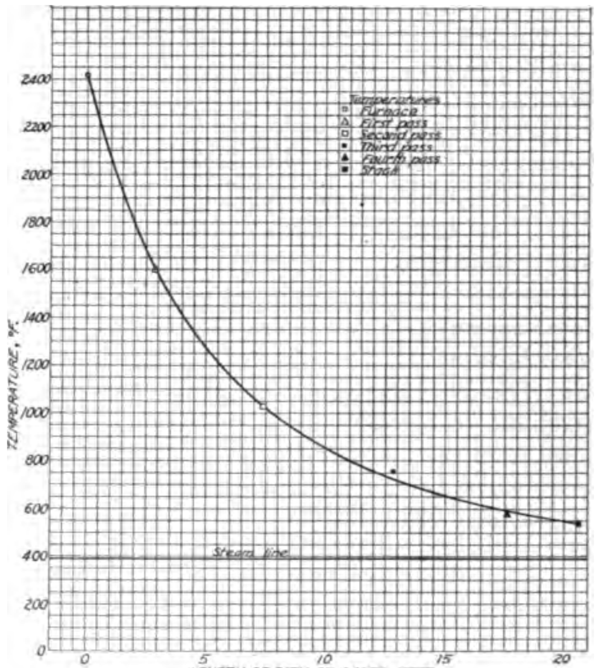


FIG. 10 TEMPERATURE DROP THROUGH FOUR-PASS BOILER

to the front water leg, contains only slowly moving and comparatively cool gases. The same condition exists in the third pass, most of the hot gases passing through the 18-in. space next to the end of the third baffle, the remaining 28 in. of the third pass being practically unused by the moving gases. The slight elevation in the temperature at a distance of 12 in. from the rear water leg is very likely due to hot gases passing through the narrow space left between the second baffle and the rear water leg. These temperature measurements show the desirability of increasing the length of the second and third baffles and thus cause the hot gases to pass over a greater part of the heating surface of the boiler. Although the temperature measurements do not show the desirability of increasing the length of the first baffle, nevertheless the baffle was made 10 in. longer after the first six tests. Lengthening the first baffle had the tendency to bring the air admitted over the fuel bed through the firing doors farther into the stream of combustible gases rising from the fuel bed and thus aid in their combustion.

57 Fig. 8 gives the temperature measurements on one of the tests made after the baffles of the three-pass boiler were lengthened. In this test three moving couples and one stationary couple were used in each pass. The temperatures obtained by the stationary couple are represented by the line connecting the points indicated by small crosses. One of the moving couples was placed in the center of the boiler and the other two about 18 in. from each side wall of the boiler. With these longer baffles the hot gases more nearly fill the passage left between the end of the baffle and the water legs. Considerable variation is also shown by the three moving couples, indicating that on one side of the boiler the gases may be hotter than on the other, which fact can be expected with such a wide and shallow furnace. The fuel bed on one side of the furnace might be in different condition than the fuel bed on the other. Owing to the short distance from the fuel bed to the boiler the gases have little chance to mix and equalize the temperature.

58 Fig. 8 also indicates that it is very difficult to tell what the temperature is in any one pass by making temperature measurements with only one stationary couple. Such measurements may be hundreds of degrees higher or lower than the average temperature in the pass.

TEMPERATURE MEASUREMENTS ON FOUR-PASS BOILER

59 Fig. 9 shows similar measurements on the four-pass boiler. In this case three moving couples and one stationary couple are used in each pass. The position of the stationary couple is indicated by the small circle, and the temperatures indicated by the couple are shown by the dotted line. These temperature measurements indicate that in the second, third, and fourth passes a large part of the stream of hot gases passes within the narrow space at the end of the baffle. The little hump in the curves in the third pass near the

rear water leg is very likely due to some of the hot gases passing through the small space between the second baffle and the rear water leg. These gases pass up close to the rear water leg and cause a somewhat higher temperature.

60 Fig. 10 shows the average temperature drop of the gases along their path of travel through the four-pass water-tube boiler. The points on these curves have been obtained by averaging the points in the stream of gas in Fig. 9. The abscissae of the chart are the approximate lengths of the paths of gases through the boiler, measured from the point where they enter among the lowest row of boiler tubes.

DISCUSSION

Albert A. Cary (written). This paper is of considerable interest to the writer who is not wholly unfamiliar with its details, as this boiler came to his attention in the early days of its development. It resulted from a study of a number of the older forms of boilers, followed by an incorporation of their various desirable features, which might apply to the existing requirements.

From the description given it will be seen that this boiler has 388 three-inch tubes in the steam generating surface which are attached front and rear to water legs slightly inclined to the vertical. The steam generated in these tubes passes back to the rear water leg from the top of which its entire volume is supposed to pass forward to the steam drum through 21 three-inch tubes. In other words, practically 18.5 of these steam generating tubes have but one three-inch outlet through which to discharge the steam they generate. This statement alone does not wholly express the insufficiency of steam outlet area as it will be realized that the water heated to the temperature of the steam, say 379 deg. fahr., occupies only 0.0184 cu. ft. per lb. while the resulting steam under 179 lb. gage pressure occupies 2.358 cu. ft., 128 times its former volume.

By applying the formulae used for computing the flow of steam in pipes, where the pipes deliver steam from the boiler and there is a material pressure drop, it could be shown that the 21 three-inch tubes delivering steam from the rear header to the front cross drum were not only ample in number and size, but that the steam flowing was traveling at an absurdly low velocity.

Such formulae do not, however, apply in this case.

By applying a steam gage to the rear of the header and another tendency in the merchant marine, he said, was toward higher pressures, superheat and the use of turbines.

L. P. BRECKENRIDGE congratulated the Society upon having presented to it a paper by men so well qualified to do so as Messrs. Dean and Kreisinger, and pointed out that the complete combustion of fuel depended upon three things — the proper amount of air, mixed thoroughly with the fuel at the proper temperature.

R. SANFORD RILEY thought that the water-tube boiler had made an effective entrance into merchant marine engineering practice and that this opened the way for the automatic stokers, which were formerly prohibited by the small furnace in the prevailing Scotch marine type boilers. If coal continued to be used at sea the impossibility of obtaining skilled firemen would force the adoption of automatic stokers under boilers that allowed space enough for their installation.

JOHN VAN BRUNT said that underfeed stokers had been used at sea and that there was no mechanical reason why they should not come into universal use.

In a closure, Henry Kreisinger said that the Bureau of Mines experiments showed that the circulation in boilers was as Mr. Cary had pointed out, very slight in the middle section of tubes, and in opposite directions in the sets of tubes above and below. He did not think that the reversed circulation in the tubes would cause steam to remain in them very long. If a rise in overall efficiency is to be credited to increased water circulation in the boiler, it must be accompanied by a corresponding drop in the flue gas temperature, other conditions, of course, remaining the same. A rise in overall boiler efficiency of 5.5 per cent would have to be accompanied by a drop in flue gas temperature of about 135 deg. fahr. Therefore experimenters who wish to demonstrate the effectiveness of any device for increased circulation should carefully measure the flue gas temperatures, analyze the gases, and see that the heating surfaces are in the same condition as to cleanliness.

Mr. Kreisinger announced that the investigations published in the paper would be completely reported in a publication of the Bureau of Mines.

No. 1714

FLOW OF WATER THROUGH CONDENSER TUBES

By William L. De Baufre, Lincoln, Neb.
and
Milton C. Stuart, Annapolis, Md.
Members of the Society

In this paper the authors give particulars regarding an extended series of tests recently conducted at the United States Naval Engineering Experiment Station at Annapolis, Md., to determine the friction loss of water flowing through ⅝*-in. No.* 18 *gage standard condenser tubes. The investigations covered variable velocities, water temperatures, and tube lengths, as well as the effect of both fresh and salt water. The results obtained are presented in tabular form and from them a general formula has been derived which gives the total drop in pressure due to entrance and exit losses and to frictional resistance within the tube.*

THERE was conducted recently at the U. S. Naval Engineering Experiment Station, Annapolis, Md., an investigation upon the friction loss of water flowing through ⅝-in. standard condenser tubes of No. 18 gage, 0.522 in. in internal diameter. The principal variables were velocity and temperature of water, and tube length. The investigation also covered the variation in friction loss with clean tubes and tubes as received, and the effect of fresh and salt water. The computations were made in such a manner that losses at the tube ends and along the tube could be separated, the results being expressed in a general formula.

2 A number of sections of 3-in. iron pipe were assembled with flanges as indicated in Fig. 1. A corresponding number of lengths of ⅝-in. condenser tubing were obtained, and two brass blank flanges prepared to serve as tube sheets. These tube sheets could be inserted between any two pairs of flanges, thus enabling tube lengths of approximately 5, 8, 11, 14, 17 and 20 ft. to be tested. The tubes were held in place by ordinary screwed glands in the tube sheets, and packed with cotton held between two fiber washers as

---

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

shown in Fig. 2. They were supported by sheet-iron disks between the intermediate pairs of flanges.

3 Referring to Fig. 1, water from the storage tank S was pumped by the motor-driven centrifugal pump P through the heater

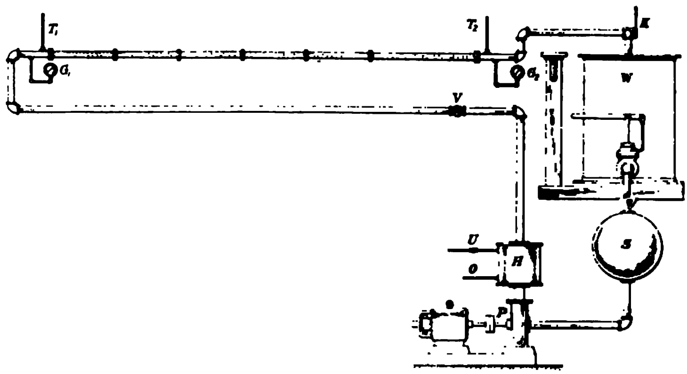


FIG. 1 DIAGRAM OF TEST APPARATUS

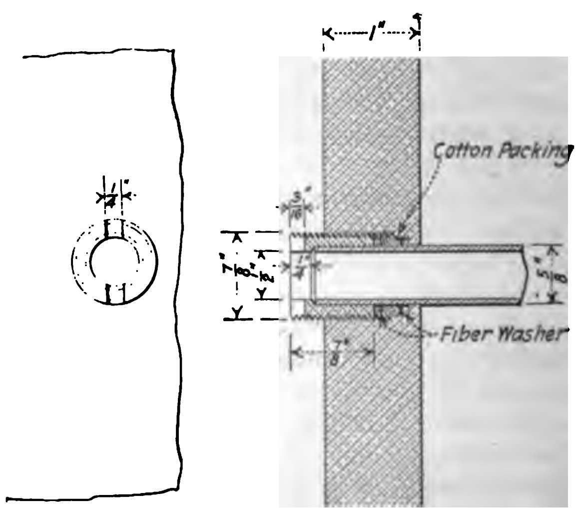


FIG. 2 METHOD OF SECURING CONDENSER TUBE IN TUBE SHEET

H to the condenser tube indicated by the dotted lines. From the condenser tube the water was discharged into the two tanks W, where it was weighed before being discharged into the storage tank S below. The centrifugal pump P ran at constant speed, the rate of flow of water being regulated by valve V. The temperature of the water as indicated on thermometers T_1 and T_2, was regulated by the valve U admitting steam to the heater H. The condensed

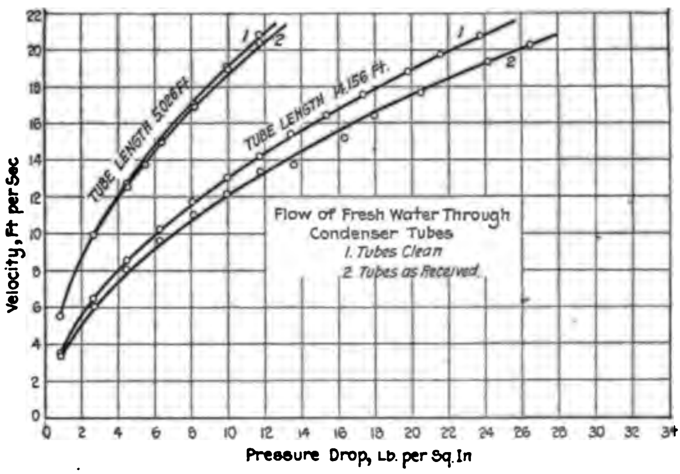


FIG. 3 CURVES SHOWING THE EFFECT OF CLEANING TUBES

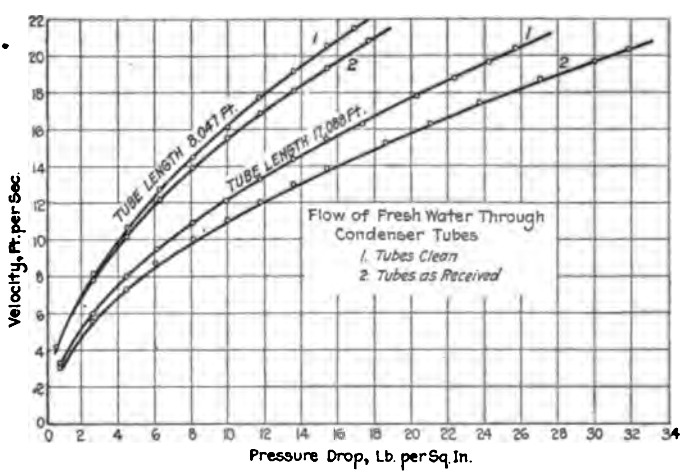


FIG. 4 CURVES SHOWING THE EFFECT OF CLEANING TUBES

steam was discharged through O into a trap not shown. Gages G_1 and G_2 served to measure the pressures before and after the tube.

4 With each tube length a number of runs were made with

distilled water at temperatures of 85, 100, 130, 160, and 190 deg. fahr. and with rates of flow up to about 7500 lb. per hour, corresponding to a velocity of about 22 ft. per sec. A few runs were first made

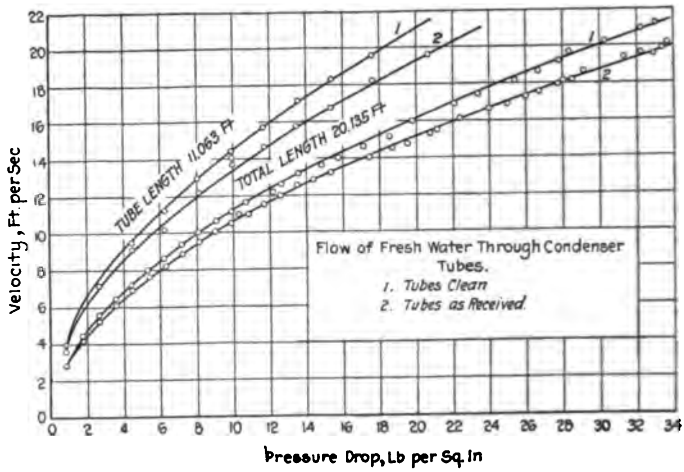


FIG. 5 CURVES SHOWING THE EFFECT OF CLEANING TUBES

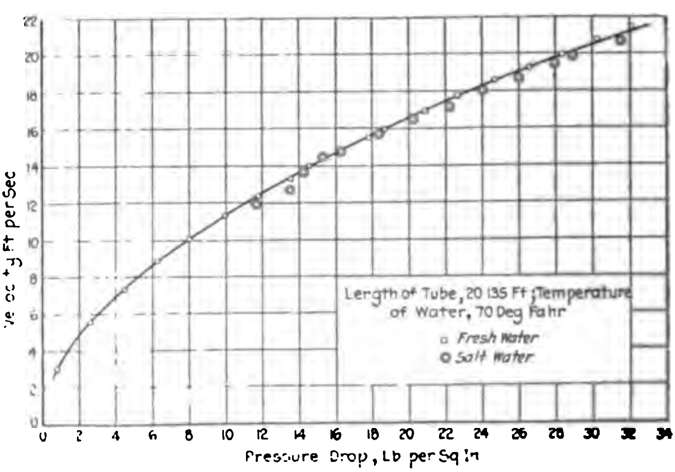


FIG. 6 CURVES SHOWING A COMPARISON OF THE EFFECTS OF SALT AND FRESH WATER FLOWING THROUGH CONDENSER TUBES

with the tube in the condition as received and at 100 deg. fahr. The tube was then cleaned by pushing through it a small rag which had been soaked in kerosene. After cleaning, the complete set of runs was made as tabulated in Table 1. In order to check the constancy of the results from day to day, a certain pressure drop and rate of flow were selected as a standard at each temperature,

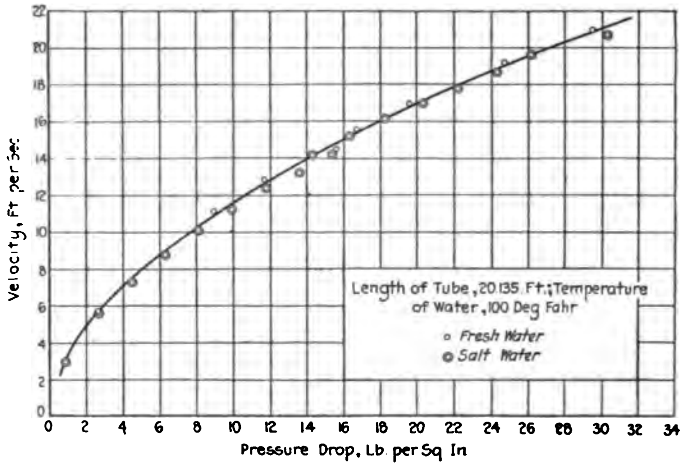


FIG. 7 CURVES SHOWING A COMPARISON OF THE EFFECTS OF SALT AND FRESH WATER FLOWING THROUGH CONDENSER TUBES

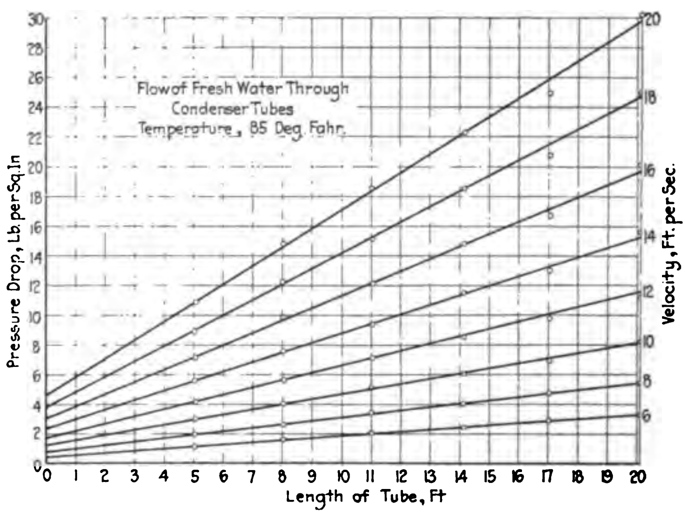


FIG. 8 CURVES SHOWING THE RELATION OF PRESSURE DROP TO TUBE LENGTH WITH WATER AT 85 DEG. FAHR.

and the run under these conditions was repeated when starting up in the morning and just before shutting down at night.

5 After completing the runs with distilled water, which began with the longest tube and ended with the shortest one, the

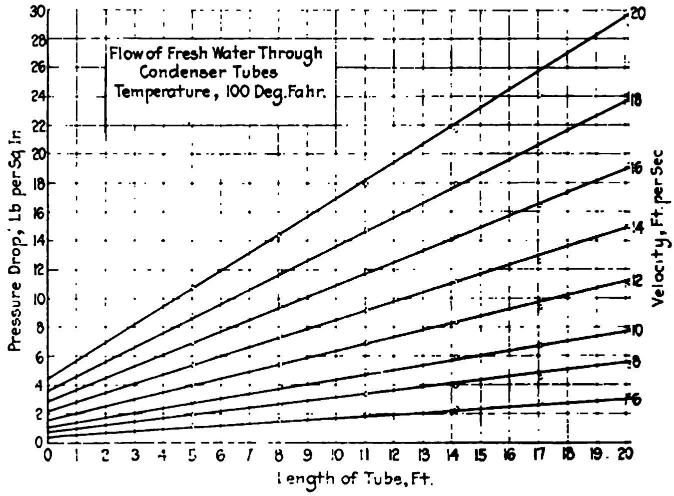


FIG. 9 CURVES SHOWING THE RELATION OF PRESSURE DROP TO TUBE LENGTH WITH WATER AT 100 DEG. FAHR.

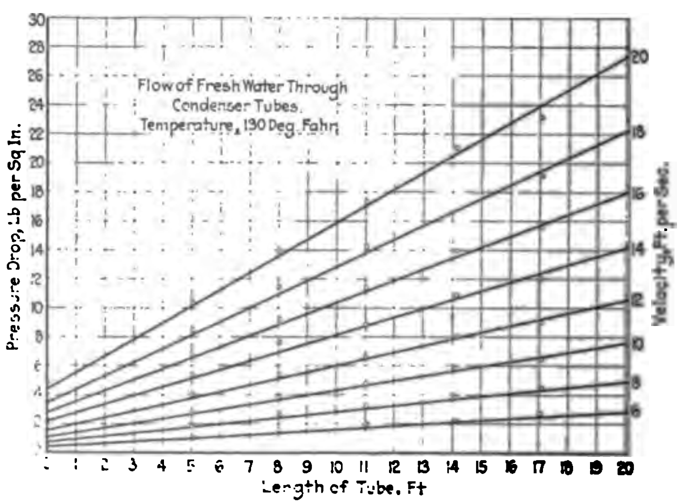


FIG. 10 CURVES SHOWING THE RELATION OF PRESSURE DROP TO TUBE LENGTH WITH WATER AT 130 DEG. FAHR.

longest tube was again installed in order to make runs with salt water and compare the results with the data obtained for fresh water. A few runs were repeated with fresh water and then duplicated with salt water at both 70 and 100 deg. fahr. The salt water was prepared by adding to the fresh water one thirty-second of its

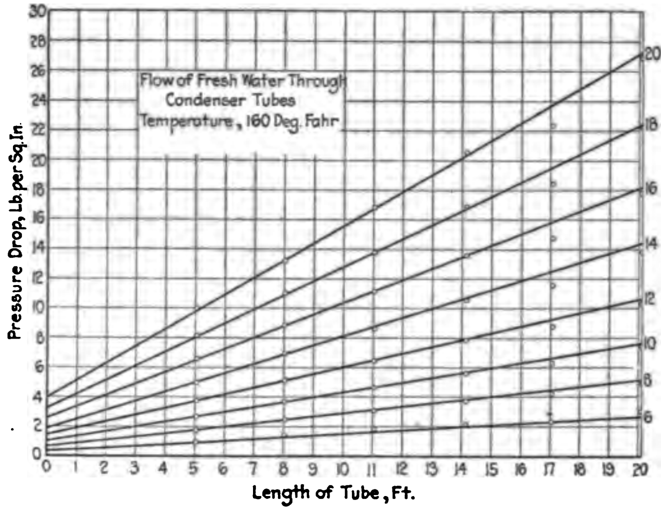


FIG. 11 CURVES SHOWING THE RELATION OF PRESSURE DROP TO TUBE LENGTH WITH WATER AT 160 DEG. FAHR.

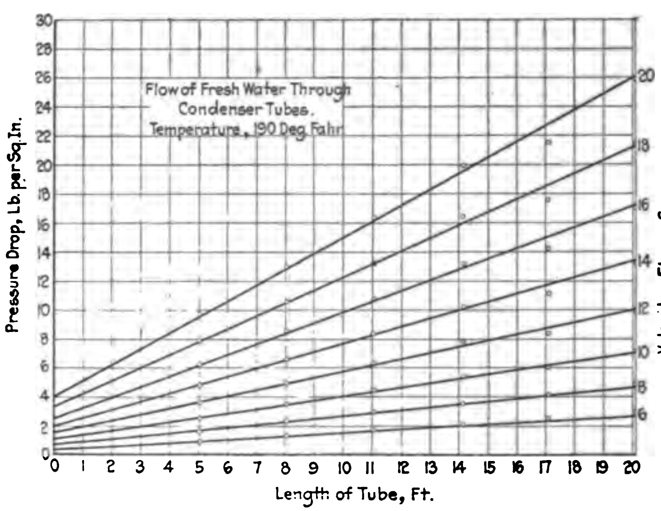


FIG. 12 CURVES SHOWING THE RELATION OF PRESSURE DROP TO TUBE LENGTH WITH WATER AT 190 DEG. FAHR.

weight of salt, in order to approximate the normal density of sea water.

6 For the several tube lengths it was decided to use entirely different tubes rather than to take one long tube and cut off parts to obtain the shorter tubes. The results therefore include the variations that are liable to occur with commercial tubes of this size and gage.

7 The data observed and the results calculated therefrom are given in Table 1. The source of each item in the table is given in the Appendix.

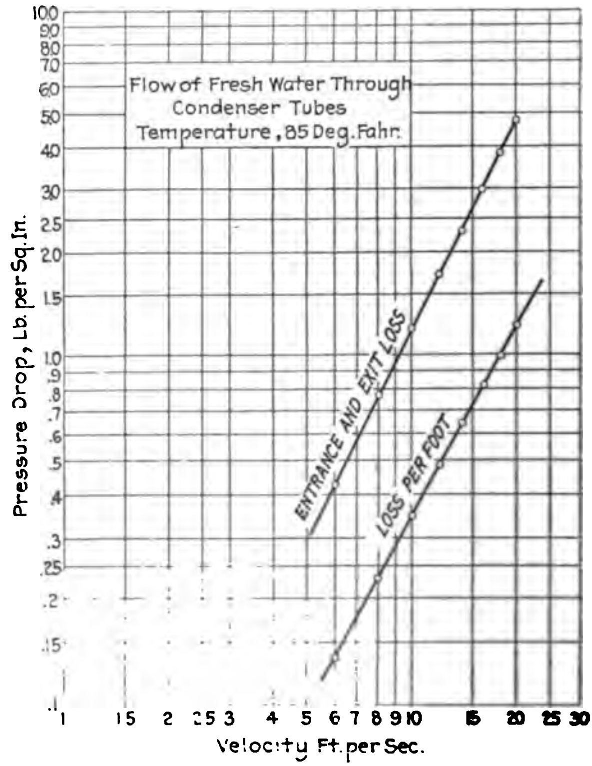


FIG. 13 LOGARITHMIC PLOTTING OF PRESSURE DROP VERSUS VELOCITY FOR A WATER TEMPERATURE OF 85 DEG. FAHR.

8 The effect of cleaning the tubes is shown in the curves of Figs. 3, 4 and 5. As received from the manufacturers, condenser tubes apparently offer a resistance to the flow of water from 10 to 20 per cent greater than after cleaning them. A more thorough cleaning would probably have still further reduced the frictional resistance. To approximate actual condenser conditions it was deemed advisable

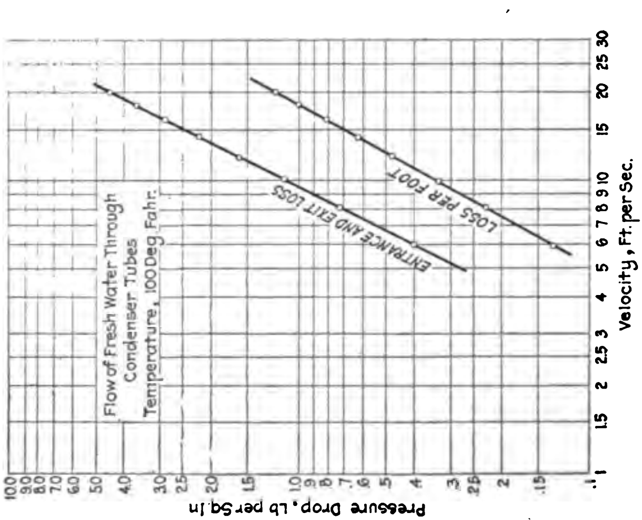


FIG. 14 LOGARITHMIC PLOTTING OF PRESSURE DROP VERSUS VELOCITY FOR A WATER TEMPERATURE OF 100 DEG. FAHR.

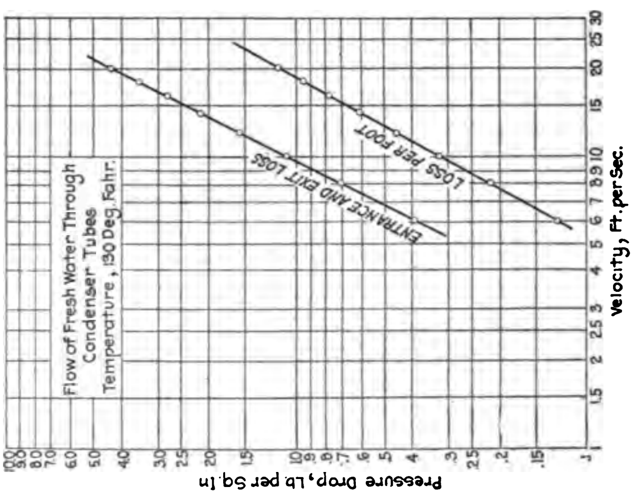


FIG. 15 LOGARITHMIC PLOTTING OF PRESSURE DROP VERSUS VELOCITY FOR A WATER TEMPERATURE OF 130 DEG. FAHR.

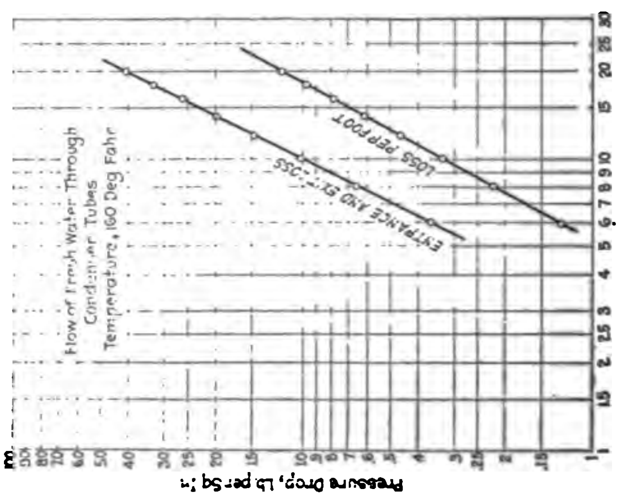


Fig. 16 Logarithmic Plotting of Pressure Drop versus Velocity for a Water Temperature of 160 Deg. Fahr.

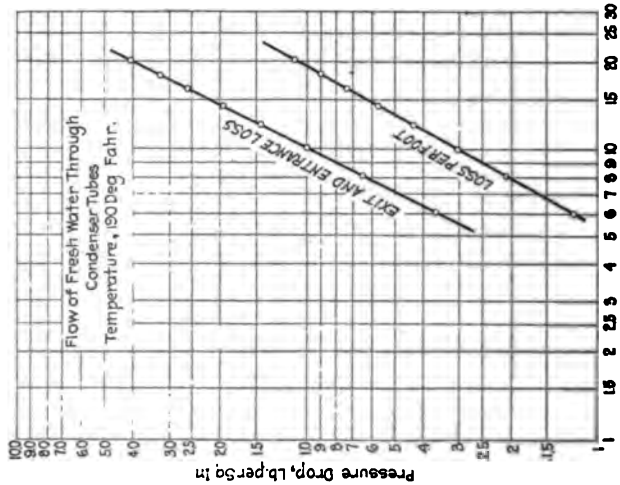


Fig. 17 Logarithmic Plotting of Pressure Drop versus Velocity for a Water Temperature of 190 Deg. Fahr.

to add about 20 per cent to the tabulated results obtained with cleaned tubes.

9 The curves of Figs. 6 and 7 show that the resistance with salt water having a salinity of one thirty-second, equivalent to that of sea water, was practically the same as with fresh water. Consequently no further corrections are necessary to apply the results obtained in this investigation to condenser conditions on board ship.

10 In order to separate the loss of head at entrance and exit

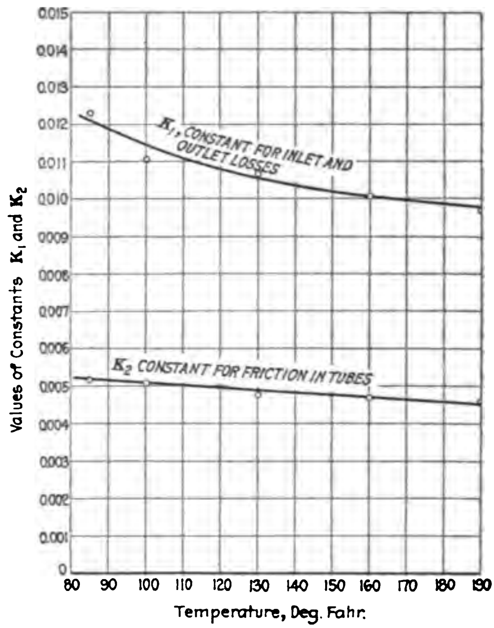


FIG. 18 CURVES SHOWING THE RELATION OF CONSTANTS TO THE TEMPERATURE OF WATER

from the frictional resistance through the tubes, there were plotted in Figs. 8 to 12, inclusive, the pressure drop versus the tube length for various velocities. The points plotted in these figures were not taken directly from Table 1, but from faired curves (not included in this paper) of all runs made. By prolonging the straight lines for each velocity back to zero tube length, there was obtained the loss of head, or pressure drop, at entrance and exit.

11 The total drop in pressure of water flowing through a condenser tube may be expressed by the formula —

$$\begin{aligned} P &= P_1 + P_2 \\ &= K_1 V^{n_1} + K_2 L V^{n_2} \end{aligned}$$

where P = total drop in lb. per sq. in.

P_1 = drop in lb. per sq. in. due to entrance and exit losses

P_2 = drop in lb. per sq. in. due to frictional resistance within the tube

n_1 = velocity exponent for entrance and exit losses

n_2 = velocity exponent for frictional resistance within the tube

L = tube length in ft.

K_1 = factor for entrance and exit losses, and

K_2 = factor for frictional resistance.

12 Theoretically, the entrance and exit losses should vary as the square of the velocity. This is confirmed experimentally by the curves plotted in Figs. 13 to 17, inclusive, from the pressure drops in Figs. 8 to 12, corresponding to zero tube length. The velocity exponent for the frictional resistance within the tube should lie between 1 and 2. The curves in Figs. 13 to 17 for the loss per foot of tube length were obtained by plotting the slopes of the straight lines drawn in Figs. 8 to 17. The exponent in all cases was found to be 1.83.

13 The following values of K_1 and K_2 were taken from the curves of Figs. 12 to 16, inclusive, and are plotted in Fig. 18.

Temperature, deg. fahr.	K_1	K_2
85	0.0123	0.0052
100	0.0111	0.0051
130	0.0107	0.0048
160	0.0101	0.0047
190	0.0097	0.0046

14 For clean condenser tubes, standard ⅝-in. outside diameter, No. 18 gage, 0.0345 in. thick, and for an average water temperature of 90 deg. fahr., we may write

$$P = 0.0118\ V^2 + 0.0051\ LV^{1.83}$$

where P is the total loss in pounds per square inch; V is the velocity in feet per second; and L is the tube length in feet. With very dirty tubes the entrance and exit losses would undoubtedly be unchanged, but the loss per foot would be greater. Assuming the latter to be increased 20 per cent for condenser tubes in the ordinary condition, for standard ⅝-in. tubes No. 18 gage we may write

$$P = 0.0118\ V^2 + 0.0061\ LV^{1.83}$$

Expressing the resistance in feet head of water, we have

$$H = 0.0274\ V^2 + 0.0141\ LV^{1.83}.$$

TABLE 1 RESULTS OF INVESTIGATION ON FRICTION LOSS OF WATER FLOWING THROUGH CONDENSER TUBES

Reference number	Date, 1917	Duration of run, minutes	Room temperature, degrees Fahr.	Tube length, feet	Tube diameter, inches	Temperature of water at inlet, degrees Fahr.	Temperature of water at outlet, degrees Fahr.	Average temperature	Pressure of water at inlet, lb. per sq. inch	Pressure of water at outlet, lb. per sq. inch	Pressure drop, lb. per sq. inch	Weight of water during run	Velocity, feet per second
1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	July 18	15	773	20135	03214	843	850	848	3631	315	3316	17672	21134
2	18	15	779	-	-	842	847	845	3354	315	3239	17467	20909
3	18	15	781	-	-	844	848	846	3344	305	3039	16683	20181
4	18	15	785	-	-	844	848	846	3149	298	2851	16327	19545
5	19	15	862	-	-	837	845	841	3046	250	2796	15949	19085
6	19	15	871	-	-	844	849	847	2929	240	2684	15569	18632
7	19	15	874	-	-	846	853	850	2786	240	2546	15150	18140
8	19	15	880	-	-	844	849	847	2578	230	2348	14508	17406
9	19	15	885	-	-	844	849	847	2451	230	2221	14126	16927
10	19	15	889	-	-	844	849	847	2356	365	1991	13282	15906
11	19	15	793	-	-	844	848	846	2208	350	1858	12756	15272
12	19	15	797	-	-	844	849	847	2054	340	1714	12185	14590
13	19	15	801	-	-	843	850	846	1911	330	1581	11630	13820
14	19	15	805	-	-	843	849	847	1870	330	1540	11600	13922
15	19	15	801	-	-	843	849	847	1818	320	1498	11258	13565
16	19	15	808	-	-	846	850	848	1679	320	1359	10879	13115
17	19	15	817	-	-	846	851	849	1568	300	1268	10420	12555
18	19	15	820	-	-	843	849	847	1497	270	1227	10062	12126
19	19	15	819	-	-	843	848	846	1337	260	1077	9679	11598
20	19	15	811	-	-	843	848	846	1237	240	997	9192	11081
21	19	15	808	-	-	844	849	847	1146	240	906	8731	10519
22	20	15	786	-	-	839	845	842	1175	360	815	8241	9923
23	20	15	803	-	-	848	856	852	1055	330	725	7697	9272
24	20	15	804	-	-	845	850	848	944	310	634	7117	8575
25	20	15	822	-	-	847	853	850	802	260	542	6588	7938
26	20	15	828	-	-	847	852	850	689	240	449	5971	7194
27	20	15	832	-	-	845	848	847	581	220	361	5277	6358
28	23	15	765	-	-	846	852	849	1861	320	1541	11627	14008
29	23	15	785	-	-	863	869	866	432	160	272	4510	5434
30	23	15	793	-	-	860	866	863	338	150	188	3601	4338
31	26	15	804	-	-	849	857	853	1940	400	1540	11631	14013
32	26	15	842	-	-	826	834	830	250	160	090	2434	2932
33	23	15	791	-	-	990	1002	1001	1860	320	1540	11741	14184
34	23	15	809	-	-	1000	1003	1002	1760	310	1450	11356	13728
35	23	15	821	-	-	999	1002	1001	1659	300	1359	10974	13262
36	23	15	828	-	-	999	1003	1001	1558	290	1268	10577	12779
37	23	15	824	-	-	996	997	997	1770	230	1540	11747	14194
38	24	15	830	-	-	997	999	998	1840	300	1540	11767	14196
39	24	15	839	-	-	1001	1003	1002	1447	220	1227	10198	12317
40	24	15	826	-	-	1000	1003	1002	1357	220	1137	9739	11767
41	24	15	823	-	-	994	996	995	1237	240	997	9282	11216
42	24	15	848	-	-	1003	1008	1005	1146	240	906	8851	10744
43	24	15	828	-	-	997	999	998	1045	230	815	8314	10047
44	24	15	839	-	-	998	1002	1000	945	220	725	7830	9463
45	24	15	851	-	-	997	1001	999	844	210	634	7288	8811
46	24	15	875	-	-	1002	1004	1003	753	210	543	6700	8103
47	24	15	883	-	-	996	998	997	653	200	453	6054	7316
48	24	15	882	-	-	993	995	994	542	180	362	5353	6470
49	24	15	898	-	-	1008	1008	1008	442	170	272	4568	5522
50	24	15	891	-	-	1003	1006	1004	1859	320	1539	11724	14172
51	24	15	844	-	-	993	997	995	1859	320	1539	11749	14194
52	25	15	839	-	-	1001	1005	1003	341	160	181	3653	4417
53	25	15	837	-	-	1006	1001	1004	250	160	090	2490	3010
54	25	15	840	-	-	997	1000	999	2256	345	1911	13103	15839
55	25	15	843	-	-	1004	1007	1006	2161	340	1821	12801	15476
56	25	15	847	-	-	999	1004	1002	2354	365	1989	13406	16206
57	25	15	850	-	-	999	1005	1002	2473	370	2103	13749	16622
58	25	15	854	-	-	999	1005	1002	2573	373	2200	14076	16946
59	25	15	858	-	-	999	1004	1002	2654	392	2262	14287	17275
60	25	15	860	-	-	998	1007	1002	2747	397	2350	14770	17732
61	25	15	868	-	-	1005	1008	1007	1869	330	1539	11784	14246
62	26	15	769	-	-	996	999	998	1855	320	1535	11747	14194
63	26	15	774	-	-	1005	1008	1007	1770	320	1450	11436	13825
64	26	15	786	-	-	1010	1014	1012	1964	360	1604	12206	14757
65	26	15	795	-	-	995	1001	998	2834	437	2397	14940	17936
66	26	15	804	-	-	988	992	990	2976	325	2651	15739	18988
67	26	15	816	-	-	995	999	997	3145	330	2816	16225	19536
68	26	15	828	-	-	1001	1005	1003	3456	360	3096	17138	20734
69	26	15	840	-	-	984	987	986	3468	360	3108	17237	20674
70	26	15	852	-	-	994	998	996	3661	370	3291	17661	21189
71	26	15	868	-	-	997	1001	999	3808	377	3431	18088	21712
72	26	15	880	-	-	998	1002	1000	2476	305	2231	14832	17880
73	26	15	891	-	-	988	992	990	1794	250	1544	11768	14217
74	Aug. 7	15	850	-	-	997	998	998	1880	340	1540	11738	14166
75	8	15	831	-	-	1007	1008	1008	1790	340	1450	11374	13653
76	July 27	15	849	-	-	1293	1293	1293	1780	240	1540	12094	14664
77	27	15	833	-	-	1293	1293	1293	1659	210	1449	11681	14217
78	27	15	880	-	-	1316	1313	1313	1568	210	1358	11305	13767
79	27	15	863	-	-	1316	1310	1313	1468	220	1248	10899	13279
80	27	15	853	-	-	1300	1300	1300	1387	210	1177	10485	12767
81	27	15	833	-	-	1305	1303	1304	1238	200	1038	10010	12195
82	27	15	919	-	-	1295	1296	1296	1147	200	947	9545	11621
83	27	15	928	-	-	1304	1302	1303	1105	200	905	9048	11013
84	27	15	932	-	-	1308	1305	1307	904	180	724	8011	9755
85	27	15	936	-	-	1302	1298	1301	833	290	543	6874	8373

TABLE 1 (*Cont.*)

Reference number	Date, 1917	Duration of run, minutes	Room temperature, degrees fahr	Tube length, feet	Tube diameter, inches	Temperature of water at inlet, degrees fahr	Temperature of water at outlet, degrees fahr	Average temperature	Pressure of water at inlet, lb per sq inch	Pressure of water at outlet, lb per sq inch	Pressure drop, lb per sq inch	Weight of water during run	Velocity, feet per second
1	2	3	4	5	6	7	8	9	10	11	12	13	14
86	July 27	15	938	20 35	0.5214	1305	1304	1305	2078	540	1538	12054	14678
87	28	15	868			1300	1298	1299	553	190	363	5516	6606
88	28	15	880			1304	1297	1301	2148	450	1698	7702	9310
89	30	15	890			1292	1292	1292	1879	413	1466	11705	14155
90	30	15	896			1310	1310	1310	2243	470	1773	13060	15795
91	30	15	902			1284	1283	1284	2131	220	1871	13697	16555
92	30	15	910			1294	1292	1293	2439	220	2219	14639	17694
93	30	15	923			1288	1298	1293	2619	233	2386	15174	18341
94	30	15	930			1297	1297	1297	2879	238	2581	15851	19161
95	30	15	938			1290	1292	1291	3056	267	2789	16501	19945
96	30	15	[illegible]			1288	1288	1293	3174	270	2904	16859	20384
97	31	15	[illegible]			1300	1299	1300	1759	220	1539	7704	9310
98	31	15	[illegible]			1300	1301	1301	3321	245	3076	17732	[illegible]
99	31	15	[illegible]			1320	1317	1319	3548	270	3278	18062	[illegible]
100	31	15	[illegible]			1322	1300	1311	2703	260	2443	15454	18687
101	31	15	948			1304	1298	[illegible]	432	160	272	4680	5660
102	Aug 2	15	942			[illegible]	1286	[illegible]	341	160	181	3676	4442
103	3	15	[illegible]			1609	1637	[illegible]	3601	370	3231	18811	[illegible]
104	3	15	[illegible]			1603	1587	[illegible]	3431	433	2998	17471	21311
105	3	15	827			[illegible]	1607	[illegible]	[illegible]	440	2881	17139	20908
106	3	15	824			[illegible]	[illegible]	[illegible]	[illegible]	430	2751	16716	20438
107	3	15	830			[illegible]	1594	[illegible]	[illegible]	410	2611	16165	19785
108	3	15	835			[illegible]	1592	[illegible]	[illegible]	390	2336	15600	[illegible]
109	3	15	840			[illegible]	1590	1601	[illegible]	380	2281	15050	18341
110	3	15	847			[illegible]	1600	1604	2243	350	2081	14529	17842
[illegible]	3	15	847			[illegible]	[illegible]	[illegible]	2225	340	1885	13357	16510
[illegible]	3	15	848			[illegible]	1603	[illegible]	[illegible]	[illegible]	1706	12806	15617
[illegible]	6	15	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1635	12132	[illegible]
[illegible]	6	15	884			[illegible]	[illegible]	[illegible]	[illegible]	280	858	11449	[illegible]
[illegible]	6	15	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1177	10605	[illegible]
[illegible]	6	15	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	6	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	814	8710	[illegible]
[illegible]	6	[illegible]	[illegible]			[illegible]	586	[illegible]	[illegible]	280	634	7653	9277
[illegible]	6	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	200	453	6346	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	180	272	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	160	181	3805	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	60	090	[illegible]	[illegible]
123	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	70	[illegible]	3132	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	272	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	632	80	452	6163	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	220	634	7778	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	240	908	9161	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	280	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	310	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	340	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1636	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	13764	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	450	2389	15322	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	3096	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	3203
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

TABLE 1 (*Cont.*)

Reference number	Date, 1917	Duration of run, minutes	Room temperature, degrees Fahr.	Tube length, feet	Tube diameter, inch	Temperature of water at inlet, degrees Fahr.	Temperature of water at outlet, degrees Fahr.	Average temperature	Pressure of water at inlet, lb. per sq. inch	Pressure of water at outlet, lb. per sq. inch	Pressure drop, lb. per sq. in.	Weight of water during run	Velocity, feet per second
1	2	3	4	5	6	7	8	9	10	11	12	13	14
171	Aug 21	15	911	7088	05204	1538	1536	1537	626	174	452	6906	8475
172	21	15	912	-	-	1905	1904	1904	3089	350	2737	[illegible]	[illegible]
173	21	15	913	-	-	1905	1892	1899	2431	343	2088	[illegible]	[illegible]
174	21	15	913	-	-	1913	1907	1910	2624	328	2296	[illegible]	[illegible]
175	22	15	890	-	-	1904	1906	1905	2173	280	2021	[illegible]	[illegible]
176	22	15	894	-	-	1930	1930	1930	2123	284	1838	[illegible]	[illegible]
177	22	15	897	-	-	1904	1902	1903	1792	253	1534	[illegible]	[illegible]
178	22	15	904	-	-	1893	1891	1892	1516	249	1267	[illegible]	[illegible]
179	22	15	911	-	-	1909	1907	1908	1228	233	996	[illegible]	[illegible]
180	22	15	923	-	-	1904	1899	1902	930	206	724	[illegible]	[illegible]
181	22	15	923	-	-	1905	1899	1902	531	174	357	6189	7450
182	22	15	927	-	-	1904	1901	1903	331	150	181	[illegible]	[illegible]
183	Sept 6	15	774	4156	05201	858	856	857	270	150	120	[illegible]	[illegible]
184	6	15	772	-	-	839	840	840	452	180	272	[illegible]	[illegible]
185	6	15	788	-	-	857	859	858	633	180	453	7162	[illegible]
186	6	15	773	-	-	846	848	847	824	190	634	[illegible]	[illegible]
187	6	15	782	-	-	846	847	847	1056	240	816	[illegible]	[illegible]
188	6	15	792	-	-	845	847	846	1247	250	997	[illegible]	[illegible]
189	6	15	795	-	-	846	847	846	1448	270	1178	[illegible]	[illegible]
190	7	15	751	-	-	834	836	835	1438	260	1178	[illegible]	[illegible]
191	7	15	752	-	-	857	859	858	1639	280	1359	[illegible]	[illegible]
192	7	15	753	-	-	848	849	849	1841	300	1541	[illegible]	[illegible]
193	7	15	755	-	-	848	849	848	2030	330	1740	[illegible]	[illegible]
194	7	15	756	-	-	846	847	847	2330	350	1980	[illegible]	[illegible]
195	7	15	758	-	-	846	847	847	2520	370	2160	[illegible]	[illegible]
196	7	15	760	-	-	846	847	847	2710	390	2330	[illegible]	[illegible]
197	7	15	763	-	-	996	998	997	2750	370	2360	[illegible]	[illegible]
198	7	15	770	-	-	1000	1001	1001	2100	340	1760	[illegible]	[illegible]
199	7	15	787	-	-	996	998	997	1435	260	1178	[illegible]	[illegible]
200	8	15	764	-	-	981	966	976	470	200	270	3394	[illegible]
201	11	15	733	-	-	1287	1286	1287	449	176	272	[illegible]	[illegible]
202	11	15	742	-	-	1234	1295	1295	643	190	453	5464	[illegible]
203	11	15	748	-	-	1303	1304	1304	906	240	776	[illegible]	[illegible]
204	11	15	747	-	-	1297	1300	1299	1256	260	998	[illegible]	[illegible]
205	11	15	764	-	-	1294	1297	1296	1549	280	1269	[illegible]	[illegible]
206	11	15	774	-	-	1288	1289	1289	[illegible]	320	1520	[illegible]	[illegible]
207	11	15	778	-	-	1302	1297	1300	2225	340	1825	[illegible]	[illegible]
208	11	15	785	-	-	1236	1297	1236	2510	360	2150	[illegible]	[illegible]
209	11	15	786	-	-	1288	1295	1297	2770	390	2376	[illegible]	[illegible]
210	11	15	783	-	-	1300	1295	1298	2040	330	1710	[illegible]	[illegible]
211	19	15	828	-	-	1573	1581	1585	424	180	234	[illegible]	[illegible]
212	19	15	793	-	-	1633	1636	1634	1347	170	1177	[illegible]	[illegible]
213	20	15	780	-	-	1621	1622	1622	1930	150	1780	[illegible]	[illegible]
214	20	15	784	-	-	1597	1606	1606	2430	150	2280	[illegible]	[illegible]
215	20	15	788	-	-	1932	1932	1932	2560	170	2390	[illegible]	[illegible]
216	20	15	792	-	-	1932	1931	1932	2240	150	2030	[illegible]	[illegible]
217	20	15	796	-	-	1945	1945	1945	1990	150	1840	[illegible]	[illegible]
218	21	15	904	-	-	1918	1919	1919	1690	150	1540	[illegible]	[illegible]
219	21	15	906	-	-	1927	1928	1928	1410	150	1260	[illegible]	[illegible]
220	21	15	907	-	-	1914	1913	1914	[illegible]	150	996	[illegible]	[illegible]
221	21	15	907	-	-	1893	1904	1904	876	170	726	[illegible]	[illegible]
222	21	15	908	-	-	1922	1926	1924	599	150	449	7212	[illegible]
223	21	15	911	-	-	1948	1943	1946	424	150	274	5556	[illegible]
224	21	15	905	-	-	1912	1915	1916	238	150	088	2794	[illegible]
225	21	15	900	-	-	1905	1904	1905	1980	150	1830	[illegible]	[illegible]
226	26	15	780	1063	02596	857	857	857	241	150	091	3295	[illegible]
227	26	15	778	-	-	847	[illegible]	847	425	150	275	[illegible]	[illegible]
228	26	15	778	-	-	860	[illegible]	860	604	150	454	[illegible]	[illegible]
229	26	15	778	-	-	851	851	851	785	150	635	[illegible]	[illegible]
230	26	15	778	-	-	847	848	848	964	150	814	[illegible]	[illegible]
231	26	15	779	-	-	848	849	849	1147	150	997	[illegible]	[illegible]
232	26	15	777	-	-	851	850	851	1326	150	1134	[illegible]	[illegible]
233	26	15	779	-	-	856	851	851	1509	150	1359	[illegible]	[illegible]
234	26	15	780	-	-	853	[illegible]	853	1697	150	1547	[illegible]	[illegible]
235	27	15	742	-	-	848	848	848	1148	150	998	[illegible]	[illegible]
236	27	15	755	-	-	859	861	862	1915	150	1765	[illegible]	[illegible]
237	29	15	777	-	-	841	846	843	1691	150	1541	[illegible]	[illegible]
238	29	15	776	-	-	853	854	854	1836	150	1540	[illegible]	[illegible]
239	Oct 6	15	721	-	-	852	853	853	1741	200	1541	[illegible]	[illegible]
240	9	15	712	-	-	848	852	850	1731	190	1541	[illegible]	[illegible]
241	Sept 27	15	761	-	-	1001	1004	1001	2294	150	2044	[illegible]	[illegible]
242	27	15	767	-	-	998	1000	999	1691	150	1541	[illegible]	[illegible]
243	27	15	774	-	-	980	982	981	421	150	287	[illegible]	[illegible]
244	28	15	796	-	-	1299	1299	1299	2300	170	2130	[illegible]	[illegible]
245	28	15	804	-	-	1304	1305	1305	1853	160	1693	[illegible]	[illegible]
246	28	15	812	-	-	1306	1303	1305	1690	150	1540	[illegible]	[illegible]
247	28	15	807	-	-	1303	1303	1303	1479	150	1329	[illegible]	[illegible]
248	28	15	837	-	-	1305	1304	1305	1328	150	1178	[illegible]	[illegible]
249	28	15	831	-	-	1301	1300	1301	1147	150	997	[illegible]	[illegible]
250	29	15	798	-	-	1301	1290	1300	965	150	815	[illegible]	[illegible]
251	29	15	802	-	-	1302	1310	1310	784	150	634	[illegible]	[illegible]
252	29	15	803	-	-	1294	1292	1293	603	150	453	[illegible]	[illegible]
253	29	15	817	-	-	1299	1294	1296	422	150	272	6095	[illegible]
254	29	15	822	-	-	1286	1281	1284	240	150	090	[illegible]	[illegible]
255	29	15	827	-	-	1606	1339	1605	2230	160	2070	[illegible]	[illegible]
256	Oct 9	15	716	-	-	1605	1606	1606	2270	210	2060	[illegible]	[illegible]
257	9	15	730	-	-	1608	1609	1609	442	170	272	6080	[illegible]
258	9	15	765	-	-	1914	1915	1916	1731	190	1541	[illegible]	[illegible]
259	9	15	764	-	-	1907	1903	1905	1530	170	1360	[illegible]	[illegible]
260	9	15	780	-	-	1907	1909	1908	1358	180	1178	[illegible]	[illegible]

TABLE 1 *(Cont.)*

Reference number	Date, 1917	Duration of run, minutes	Room temperature, degrees fahr	Tube length, feet	Tube diameter, inches	Temperature of water inlet, deg fahr	Temperature of water outlet, degrees fahr	Average temperature, degrees fahr	Pressure of water at inlet, lb per sq inch	Pressure of water at outlet, lb per sq inch	Pressure drop, lb per sq inch	Weight of water during run	Velocity, feet per second
1	2	3	4	5	6	7	8	9	10	11	12	13	14
261	Oct 9	15	776	11063	05326	1907	1910	1909	986	170	816	11085	13738
262	9	15	776	"	"	1918	1911	1910	804	170	634	9720	12062
263	9	15	770	"	"	1913	1913	1913	623	170	453	8100	10058
264	10	15	705	"	"	1920	1915	1918	2272	220	2052	18168	22552
265	10	15	723	"	"	1896	1897	1897	1930	220	1710	16635	20631
266	10	15	768	"	"	1908	1908	1909	432	160	272	6192	7682
267	10	15	768	"	"	1910	1911	1911	260	170	090	3420	4243
268	13	15	803	"	"	848	851	850	228	140	088	3710	4253
269	13	15	807	"	"	852	855	854	420	150	270	6795	8098
270	15	15	806	"	"	844	846	843	601	150	451	8965	10671
271	15	15	804	"	"	842	843	843	782	150	632	10340	12756
272	15	15	806	"	"	842	843	843	976	160	816	12250	14597
273	15	15	800	"	"	842	843	844	1157	160	997	13595	16189
274	16	15	773	8047	05228	842	846	844	1368	190	1178	14913	17753
275	16	15	783	"	"	857	860	859	1553	200	1359	16123	19209
276	16	15	731	"	"	857	860	853	1750	210	1540	17215	20494
277	16	15	732	"	"	854	856	855	1903	210	1693	18086	21546
278	17	15	757	"	"	849	854	857	1731	210	1541	17235	20520
279	16	15	734	"	"	841	843	842	1761	220	1541	17270	20560
280	16	15	793	"	"	998	998	998	1750	210	1540	17065	20330
281	16	15	798	"	"	993	995	994	1056	190	906	13124	15636
282	16	15	803	"	"	994	995	995	442	170	272	6871	8210
283	16	15	803	"	"	1303	1304	1304	1750	210	1540	17578	21170
284	16	15	811	"	"	1290	1292	1291	1563	210	1333	16454	21640
285	16	15	805	"	"	1302	1301	1302	1376	200	1178	15266	18350
286	17	15	783	"	"	1301	1304	1303	1191	200	990	14021	16870
287	17	15	792	"	"	1311	1311	1311	995	180	813	12570	15120
288	17	15	806	"	"	1309	1311	1300	804	170	634	11621	13260
289	17	15	810	"	"	1300	1303	1302	623	170	453	9208	11060
290	17	15	814	"	"	1298	1300	1299	432	160	272	6597	6420
291	17	15	821	"	"	1285	1288	1287	251	160	091	3864	4650
292	17	15	831	"	"	1606	1607	1607	1750	210	1540	17671	21440
293	18	15	771	"	"	1597	1598	1598	1005	190	815	12340	15430
294	18	15	770	"	"	1600	1601	1601	432	160	272	7185	8634
295	18	15	794	"	"	1907	1908	1908	1760	220	1540	17860	21910
296	18	15	795	"	"	1913	1913	1913	1569	210	1359	16675	20460
297	18	15	800	"	"	1907	1908	1908	1388	210	1178	15468	18970
298	18	15	798	"	"	1898	1899	1899	1197	200	991	14147	17360
299	18	15	802	"	"	1898	1900	1899	1006	190	816	12735	15420
300	18	15	786	"	"	1908	1911	1910	814	180	634	11171	13310
301	18	15	806	"	"	1906	1908	1907	623	170	453	9316	11490
302	18	15	802	"	"	1910	1910	1910	433	160	272	7200	8652
303	18	15	802	"	"	1907	1909	1908	246	150	096	3860	4740
304	23	16	734	5026	05238	853	855	854	1600	240	1360	18654	22371
305	23	15	762	"	"	859	852	851	1398	220	1178	17302	20717
306	23	15	769	"	"	847	850	849	1217	220	997	16011	19002
307	23	15	775	"	"	842	845	844	1016	200	816	14412	17169
308	23	15	784	"	"	848	852	850	824	190	634	12648	15007
309	23	15	786	"	"	846	851	849	623	170	453	10584	12567
310	23	15	793	"	"	839	842	841	432	160	272	8053	9552
311	23	14	804	"	"	838	841	840	250	160	090	4483	5321
312	24	15	675	"	"	833	842	841	1211	210	1001	16182	19201
313	25	15	748	"	"	848	852	850	1418	240	1178	17565	20834
314	24	15	696	"	"	997	1000	999	1406	230	1176	17787	21097
315	24	15	700	"	"	1000	1002	1001	1026	210	816	14593	17314
316	24	15	680	"	"	997	1000	999	432	160	272	8181	9738
317	23	15	821	"	"	1300	1299	1300	1391	220	1171	17793	21124
318	23	15	818	"	"	1294	1297	1295	1217	220	997	16322	19366
319	23	15	809	"	"	1300	1300	1300	1012	200	813	14376	17067
320	24	16	678	"	"	1293	1294	1293	834	200	634	12921	15337
321	24	15	675	"	"	1309	1312	1310	624	170	454	10680	12671
322	24	15	690	"	"	1303	1307	1305	422	150	272	8305	9862
323	24	15	693	"	"	1298	1301	1300	239	160	096	4424	5253
324	24	15	756	"	"	1615	1619	1617	1007	150	817	14920	17328
325	24	15	760	"	"	1597	1601	1599	808	150	633	[illegible]	[illegible]
326	24	15	797	"	"	1606	1609	1608	614	160	454	10540	[illegible]
327	24	15	789	"	"	1622	1625	1624	424	150	272	[illegible]	[illegible]
328	24	15	799	"	"	1624	1626	1625	229	140	089	[illegible]	[illegible]
329	24	15	807	"	"	1633	1637	1635	1399	220	1179	[illegible]	[illegible]
330	24	15	817	"	"	1632	1635	1634	1194	[illegible]	[illegible]	[illegible]	[illegible]
331	24	15	787	"	"	1893	1897	1895	1386	[illegible]	[illegible]	[illegible]	[illegible]
332	24	15	783	"	"	1928	1931	1930	1196	200	[illegible]	[illegible]	[illegible]
333	24	15	782	"	"	1917	1918	1918	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
334	24	15	803	"	"	1944	1941	1944	732	160	634	[illegible]	[illegible]
335	24	15	807	"	"	1936	1938	1937	624	[illegible]	[illegible]	[illegible]	[illegible]
336	24	16	793	"	"	1893	1901	1900	424	[illegible]	[illegible]	[illegible]	[illegible]
337	24	16	791	"	"	1911	[illegible]	1910	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

APPENDIX

SOURCES OF ITEMS IN TABLE 1

ITEMS

1 Reference number.
For reference.

2 Date.
From records taken during run.

3 Duration of run, minutes.
From records taken during run.

4 Room temperature, degrees fahrenheit.
From thermometer readings taken during run.

5 Tube length, feet.
From measurement.

6 Tube diameter, inches.
From measurement with micrometers, checked by weight of water contained.

7 Temperature of water inlet, degrees fahrenheit.
From thermometer readings taken during run corrected for calibration error.

8 Temperature of water outlet, degrees fahrenheit.
From thermometer readings taken during run corrected for calibration error.

9 Average temperature, degrees fahrenheit.
Average of inlet and outlet temperatures, Items 7 and 8.

10 Pressure of water at inlet, pounds per square inch.
From gage and mercury-column readings corrected for calibration error.

11 Pressure of water at outlet, pounds per square inch.
From gage and mercury-column readings corrected for calibration error.

12 Pressure drop, pounds per square inch.
Item 10 minus Item 11.

13 Weight of water during run.
From readings taken on calibrated scales.

14 Velocity of water, feet per second.
Calculated from formula, $V = Q/A$, where Q is the quantity in cu. ft. per sec. calculated from Item 13 and a density corresponding to the average temperature (Item 9), and A is the area in sq. ft. corresponding to the diameter (Item 6).

DISCUSSION

William R. Eckart (written). The question arises whether it can be assumed that the method of "cleaning" the tube as described by the authors was proper procedure in a research of this nature. The tendency of the kerosene would be to cut and remove from the metal surface any possible accumulation of grease, etc., but would leave a coating of kerosene over the entire surface which in turn would have the effect of preventing perfect contact between the water and the surface of the metal. This film of kerosene must be considered as a contamination and we can not, therefore, assume that the surface has been cleaned. It must be considered whether a large part of the reduction of resistance, said to amount to from 10 to 20 per cent and attributed to the tubes being "cleaned by kerosene," is not due to the probability that the resistance between the surface of the kerosene, adhering to the metal surface, and the water is considerably less than the resistance between a film of water adhering to metal surface and the moving water would be.

Again it would seem that the amount of this resistance between the surfaces of the kerosene and the water would tend to change with an increase in temperature, due to the decreased surface tension of the fluids which would tend to cause the gradual displacement of the kerosene film. The action would be greater if the temperature rise was produced by a source of heat on the outside of the tubes, which would be the case in actual condenser practice, so that the heat first passed through the tube, then through the kerosene film to the water. In this case it is conceivable that the entire kerosene film might eventually be removed.

In condenser practice, unless such an influence is removed, the kerosene film on the tubes so "cleaned" we could expect to cause a reduction in heat transfer due to the increased thermal resistance of the film of kerosene as compared to one of water.

The use of distilled water in these experiments, considering that it is not commonly used for condensing purposes in practice and in view of the contaminating influence of the kerosene, would seem to have been an unnecessary refinement.

As a matter of comparison, it would hardly seem desirable to make a practice of cleaning a viscosimeter with kerosene, when attempting to make accurate determinations of the viscosity of other fluids, especially non-miscible fluids such as water.

The loss of head in pipes or tubes of any diameter of the same order of smoothness as brass condenser tubes, for fluids of any viscosity, density, etc., whether water, oil, air, etc., and flowing at any velocity, either in the region of stream line or turbulent flow has been thoroughly analyzed, based upon the results of a large number of experiments by various investigators.

The results of these studies have been set forth by Stanton and Pannell,[1] dealing with water and air; and by Glazebrook, Higgins and Pannell,[2] dealing with oils; and again has appeared in our own TRANSACTIONS in the paper by E. Buckingham on Model Experiments and Empirical Equations.[3] In the discussion of Mr. Buckingham's paper, Mr. A. R. Dodge states that he has investigated the drop in pressure of superheated steam under similar conditions and found that the results coincided exactly with the curve, Fig. 1 of Mr. Buckingham's paper.

Apparently the great usefulness of these studies, in their general application to the flow of all fluids in smooth brass pipes or tubes, has not been generally recognized by the engineering fraternity.

The main point brought out in these studies was that the coefficient of friction f in the familiar hydraulic formula for loss of head, when used in connection with pipe of uniform smoothness;

$$h = f\frac{4l}{d}\cdot\frac{V^2}{2g} = \text{loss of head in feet}$$

or

$$p_1 - p_2 = f\frac{4l}{d}\cdot\frac{V^2}{2g}\cdot\frac{w}{144} = \text{pressure drop in lb. per sq. in.}$$

is a function of Vdw/u or of the equivalent expression $4W/Pu$ which may be called the turbulence factor.

Where V = velocity in ft. per sec.
W = weight of flow in lb. per sec.
d = diameter in feet
P = wetted perimeter in feet
w = density in lb. per cu. ft.
u = absolute viscosity in poundals-sec. per sq. ft.
l = length of pipe in feet.

That is, instead of f being simply a function of the velocity and diameter as it was formerly so considered, it was found to be also

[1] Phil. Trans. Royal Soc. London, vol. A 214, 1914, p. 199.

[2] Journal of the Inst. of Petroleum Technologists, vol. 2, part 5, Dec. 1915, p. 45.

[3] Trans. Am. Soc. M. E., vol. 37, 1915, p. 266.

dependent upon two additional variables which were a function of the temperature; that is, the density and absolute viscosity.

The index formula used by the authors for the pressure drop in the tubes, namely $P_2 = K_2 L V^n_2$ takes into consideration the fact that the resistance or pressure drop does not vary as the square of the velocity, but as some exponent between 1 and 2 or for the range of their present experiments — 1.83. In general this exponent increases with the turbulence in and the roughness of the pipe, approaching 2 as a limit.

The range of Stanton and Pannell's experiments on 1.255 cm. = 0.4841 in. inside diameter tubing was such that they were able to investigate the limits of the index law of resistance, due to Osborne Reynolds and William Froude, from velocities of 22 cm. per sec. or 0.72 ft. per sec. to 3150 cm. per sec. or 103.32 ft. per sec. and found the value of n for four different stages by the Reynold's (index) method to be as follows:

Velocity in cm. per sec........	58	258	900	2250
" " ft. " "........	1.9	8.46	29.5	73.8
Value of n from plotting......	1.72	1.77	1.82	1.92

and they state as a result that "it may therefore be taken as fully demonstrated that an index law for surface friction cannot be devised which will express the facts with any accuracy, except over a comparatively small range in the value $\frac{vd}{\nu}$." ($\frac{vd}{\nu}$ is the turbulence factor Vdw/u previously given, $\nu = u/w$ = kinematic viscosity.)

It is interesting to note that the exponent $n = 1.83$ obtained in the paper under discussion is for a range of velocities of from 6 to 20 ft. per sec. which compares with $n = 1.82$ at a velocity of 29.5 ft. per sec. as given by Stanton and Pannell.

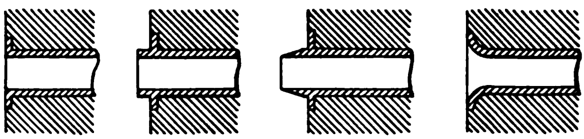
The general method of attacking the problem of loss of head at entrance and exit and the results obtained are to my mind the most valuable part of the present paper, as it takes us into a field in which very little satisfactory work has been done and where, consequently, very little working data is available.

In the past the general method of handling this part of the problem has been to assume that the case is analagous to two reservoirs with a connecting pipe. There is a loss at entrance depending upon the form of the inlet to the pipe or tube and that this loss is equal to a constant K, for the particular shape of entrance, multiplied into the velocity head $V^2/2g$ and that at the exit of the reservoir or chamber the section is large compared with the area of the tube, so that the entire velocity head $V^2/2g$ will be dissipated in eddies, giving a total loss of $(1 + K)\ V^2/2g$.

The general forms of entrance, for which constants are available are illustrated in Fig. 19.

For these forms of entrance the values of K and $1 + K$ are as follows:

Form of Inlet	K	$1 + K$
A	0.5	1.5
B	0.56	1.56
C	1.30	2.30
D	0.02 to 0.05	1.02 to 1.05



A B C D

FIG. 19 FORMS OF CONDENSER TUBE ENTRANCES

It is interesting to compare the results as obtained in the present paper with these earlier determinations and which we have had to depend upon so long.

$P_1 = K_1 V^2$ = drop in lb. per sq. in. present paper.

$P_1 = (1 + K)\dfrac{V^2}{2g} \cdot \dfrac{w}{144}$ = drop in lb. per sq. in., in the usual velocity head formula.

Or equating

$$1 + K = \frac{K_1\, 2g + 144}{w}$$

These values are tabulated below.

Temp. deg. fahr.	K_1	$1 + K$	Absolute viscosity poises = dynes-sec. per sq. cm.	Kinematic viscosity poises per gram per cu. cm.
85	0.0123	1.83	0.0081	0.00813
100	0.0111	1.655	0.00684	0.00689
130	0.0107	1.61	0.0051	0.00517
160	0.0101	1.535	0.004	0.004095
190	0.0097	1.49	0.00325	0.00336

As the re-entrant type of tube *B* is the nearest comparable with the screwed glands used in the present tests, $1 + K$ for that case equals 1.56 and would agree exactly with the present tests for a water temperature of approximately 140 deg. Fahr. Considering that the influence of temperature upon the friction losses of flowing water either in the case of surface friction or of entrance has only comparatively recently been recognized the earlier results were fairly reliable for all general work.

The influence of temperature upon the values of *K* are plotted

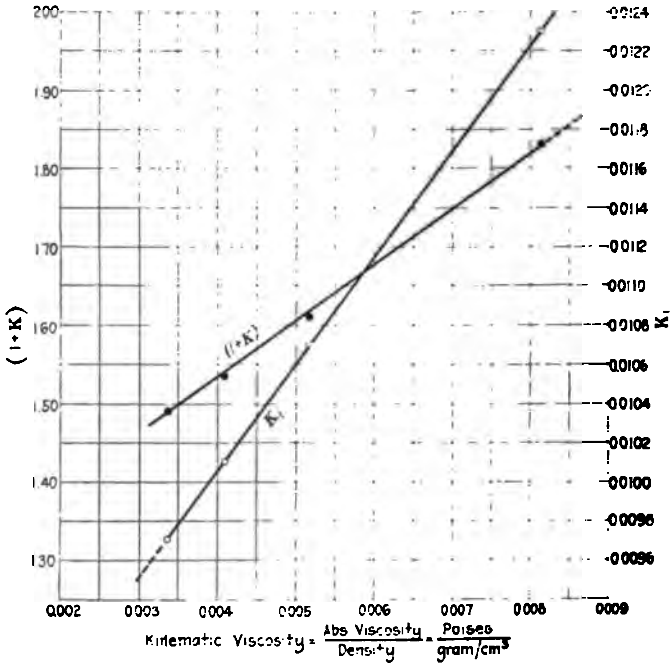


FIG. 20 RELATION BETWEEN *K* AND $1 + K$

in Fig. 18, resulting in a curve passing through the points (showing the value at 100 deg. Fahr. a little off from the general curve). As far as the friction losses are concerned, the two physical properties of the water which vary with the temperature are the viscosity and the density. The ratio of the absolute viscosity to density is the kinematic viscosity, values of which are given in the above table.

If the values of the factor *K*, or that of $(1 + K)$ are plotted against kinematic viscosity as abscissae, (Fig. 20) it is found that, for the range of the present experiments, at least, and within the limits of experimental error, there exists a straight line relation; that is, the factor K is increasing with the increase of the kinematic viscosity. The great value of this lies in its possible application beyond the range of the present experiments to fluids of greater viscosities, and it is to be hoped that further experiments with such fluids will demonstrate this to be the case, as it would completely solve in a rational and workable manner the whole problem of fluid flow in tubes or pipes, including entrance and exit losses, of roughness equivalent to that of standard brass tubing. The factors for the various forms of tube entrance, other than the screwed gland type, should likewise be determined, so as to cover the expanded tube and also a bell-mouthed tube.

One criticism of the present form of the apparatus, in so far as the application of the results to general condenser practice, would seem to be that tests carried out on a single $\frac{5}{8}$-in. tube in a 3-in. pipe would not give exactly the same entrance losses as would be expected with a nest of similar tubes pitched on about $1\frac{5}{8}$-in. centers, as in usual practices for screwed gland packing, and in which case there might be expected more or less interference as the water entered the tubes, which interference would be absent in the case of a single tube.

JOSEPH J. NELIS said that the copper packing of condenser tubes in marine practice had given trouble in the past and asked how leaks in the tubes could be prevented. William de Baufre answered that the present Navy practice was the rolled joint with bent tubes to take up contraction strains. EDWIN B. RICKETTS thought that the corset-lace packing of standard stationary condenser practice should prove equally as serviceable at sea as on land, and JOHN F. GRACE said that he had inspected condensers on a recent destroyer where corset-lace packing had been used.

PHILLIP E. REYNOLDS and G. L. KOTHNY asked about the friction losses at the entrance of the tubes, and R. J. S. PIGOTT wanted to know if any condenser manufacturer had attempted to eliminate much of this entrance and exit friction by chamfering the ferrules. AUGUST H. KRUESI said that he had always specified chamfered ferrules and JOHN F. GRACE drew on the blackboard a diagram of the chamfered ferrule made by his company.

In closing, Professor de Baufre pointed out that the experiments showed a loss of about three feet of tube due to the entrance and exit friction and that further experiments should be performed to see if it would be possible to reduce this to, say, one foot. He had no data upon the effect of bent tubes.

EDGAR BUCKINGHAM (written). It is interesting to compare the authors' results, as embodied in their equations for end loss and tube resistance, with previous results.

The end loss having been separated out, it is to be presumed that the remaining resistance is the same as it would be for an equal length L in the middle of a much longer tube. In this case, where there are no end effects to be considered, the pressure drop per unit length, or p/L, depends on the diameter D, the speed V, the density of the liquid d, and its kinematic viscosity n, *i.e.*, the quotient of its viscosity by its density. If the tube is smooth so that the diameter suffices to specify it, there does not appear to be anything further for the pressure gradient to depend on; and if there is nothing else, there must be some sort of equation

$$\frac{p}{L} = F\,(D, V, d, n) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [1]$$

containing these five variables and no others.

But if there is such an equation, it is a mathematical certainty that it can be put into the form

$$p = \frac{dLV^2}{D} f\left(\frac{n}{DV}\right) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [2]$$

f being some unknown function of the single argument (n/DV) which remains to be determined by reference to experiment. This is a purely algebraic conclusion, not involving any physical theory of flow, and it may be obtained by dimensional reasoning such as was described in the writer's paper on Model Experiments and the Forms of Empirical Equations, TRANSACTIONS, Vol. 37, 1915.

Now the authors find that their results on tube resistance may be represented by the equation

$$P_2 = K_2LV^{1.83} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [3]$$

lb. per sq. in.) or if we let p be pressure in lb. per sq. ft., to be consistent with L in feet and V in ft. per sec.

$$p = 144\,K_2LV^{1.83} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad [4]$$

This is for tubes with a mean diameter D = 0.04347 ft., and K_2 depends on the temperature but is otherwise constant.

Eq. [4] shows that, other things being equal, $p \propto V^{1.83}$; and by comparison with [2] we see that in order to make [2] consistent with this experimental fact, the unknown function f must be proportional to $(1/V)^{0.17}$, so that it must have the form

$$f\left(\frac{n}{DV}\right) = C\left(\frac{n}{DV}\right)^{0.17} \quad . . . . [5]$$

in which C is a constant and does not depend on D, V, or n.

Combining [5] and [2] we have

$$p = C\,\frac{dn^{0.17}}{D^{1.17}}\,LV^{1.83} \quad . . . . [6]$$

and by comparison of [6] with [4]

$$144\,K_2 = C\,\frac{dn^{0.17}}{D^{1.17}} \quad . . . . [7]$$

We ought therefore to have

$$\frac{144\,K_2D^{1.17}}{dn^{0.17}} = C = \text{constant} \quad . . . . [8]$$

subject to the unavoidable experimental errors. And since we know the value of D and can look up the values of d and n for the five temperatures for which values of K_2 are given, we can check the conclusion represented by Equation [8] by applying it to the authors' values of K_2. If we then take the mean value thus obtained for C we can use it in [7] to compute back values of K_2 for comparison with those found by the authors directly from their observations.

For this purpose it evidently makes no difference what units we use, but for further comparison with other equations, it is well to measure everything in normal units, *i.e.*, in terms of a system in which the definitions of the derived units are as simple as possible. I have used normal English engineering units based on the foot, second, and pound-force as fundamental units. The derived unit of speed is then 1 ft. per sec.; that of mass is 1 slug = g lb.; that of density is 1 slug per cu. ft.; and that of pressure is 1 lb. per sq. ft. The unit of kinematic viscosity has not received any name, so far as I know, but the value of n in the present system may be obtained from the C.G.S. value by multiplying by 0.001076.

Table 1 contains the results of the comparison. Line 1 gives the approximate and line 2 the actual mean temperature of experiment. Line 3 gives the density d, and line 4 the value of the kinematic viscosity n. Line 5 gives the authors' values of K_2 and line 6 the values of C found from them by means of [8].

These values of C are nearly constant and their mean is $C = 0.0732$. If we use this value of C in [7] to compute values of K_2 from those of D, d, and n already given, we obtain the values of K_2 (calculated) in line 7. Line 8 shows the differences from the authors' values, and it appears that they are probably within the experimental uncertainties, though this cannot be said positively without re-examining all the original data.

TABLE 1 COMPARISON OF OBSERVED AND COMPUTED VALUES OF K_2

1	Approximate temp., deg. fahr.	85	100	130	160	190
2	Actual mean temp., deg. fahr.	84.92	99.88	129.91	160.67	191.21
3	Density, d	1.933	1.927	1.914	1.896	1.875
4	Kinematic viscosity, n	$10^{-6} \times 8.756$	7.450	5.583	4.385	3.593
5	K_2 (observed)	0.0052	0.0051	0.0048	0.0047	0.0046
6	C	0.0716	0.0724	0.0720	0.0742	0.0759
7	K_2 (calculated)	0.00532	0.00516	0.00488	0.00464	0.00444
8	K_2 (calculated) $-$ K_2 (observed)	0.00012	0.00006	0.00008	$-$0.00006	$-$0.00016

Assuming that these differences are not significant, the authors' equation with its separate value of K_2 for each temperature may be regarded as a simplified form, applicable to a particular diameter and to water at a particular temperature, of the more general equation

$$p = 0.0732 \frac{dn^{0.17}}{D^{1.17}} LV^{1.83} \quad . \; . \; . \; . \; . \; . \; . \quad [9]$$

applicable to any diameter and to any liquid. All the quantities in Equation [9] are supposed to be measured by some sort of normal units, and the constant, being dimensionless, is the same for any other normal system as for the normal English engineering units used above. In reality, the writer first went through the computations with C.G.S. units because he had density and viscosity tabulated in those units; but the resulting value of C was the same as that given here and obtained by an independent computation.

Equation [9] shows how the value of

$$K_2 = \frac{0.0732}{144} \frac{dn^{0.17}}{D^{1.17}}$$

would change if we changed to a different temperature or to another liquid. It also shows that the resistance is inversely proportional to the 1.17-power of the tube diameter — a conclusion which is not immediately obvious from the experiments. For example, if the tubes were three times as large inside, the values of K_2 in Equation [3] would all have to be multiplied by $(\frac{1}{3})^{1.17} = 0.28$.

We already have good measurements on smooth drawn brass tubes by Saph and Schoder (Am. Soc. Civil Eng., 1903, vol. 29, p. 419) and by Stanton and Pannell (Phil. Trans. Roy. Soc. London, 1914, vol. A214, p. 199) and they are well represented by the equation (in normal units)

$$p = \frac{dLV^2}{D}\left[0.0036 + 0.306\left(\frac{n}{DV}\right)^{0.35}\right] \quad . \; . \; . \; . \; . \quad [10]$$

deduced for the purpose by C. H. Lees (Proc. Roy. Soc. London, 1914, vol. A91, p. 46). This may be regarded as satisfactory for all smooth, straight, round pipes; for any liquid or gas; and for any value of (DV/n) greater than 4000. Equation [9] representing the authors' observations may be put into the form

$$p = 0.0732\,\frac{dLV^2}{D}\left(\frac{n}{DV}\right)^{0.17} \quad . \; . \; . \; . \; . \quad [11]$$

and it seems worth while to compare this with Equation [10] which represents the best previous work on smooth tubes.

The range of (DV/n) covered by the present experiments was $\log (DV/n) = 4.15$ to 5.42; and if we compare the values of p from [9] and [10] for equal values of dLV^2/D but for various values of (DV/n) within the range just given, we have the following results:

TABLE 2 COMPARISON OF VALUES OF p FROM EQUATIONS [9] AND [10]

$\log DV/n$	4.1	4.4	4.7	5.0	5.3
$\frac{p(11)}{p(10)}$	0.99	1.05	1.12	1.14	1.17

Over any short range of (DV/n) Equation [10] may be represented approximately by an equation of the form

$$p = N\,\frac{dLV^2}{D}\left(\frac{n}{DV}\right)^{x} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [12]$$

of which [11] is a particular case with $N = 0.0732$ and $x = 0.17$. The values of N and x required to make [12] represent [10] over certain particular ranges are given in Table 3.

Tables 2 and 3 seem to indicate that the authors' tubes acted like perfectly smooth tubes at somewhat higher values of (DV/n), *i.e.*, as if the flow were somewhat more turbulent than it would be in perfectly smooth tubes at the same values of (DV/n). Very possibly the tubes were not quite so smooth as those used by Saph and Schoder and Standon and Pannell.

Turning now to the end losses, the authors find that they may be represented for each temperature by $P_1 = K_1V^2$ lb. per sq. in., or

$$p_1 = 144\,K_1V^2 \quad \ldots\ldots\ldots\ldots \quad [13]$$

lb. per sq. ft.

It seems reasonable to suppose that the pressure in the downstream end of the tube is sensibly the same as outside and that the pressure drop P_1 is all due to what happens at the entrance end. The flow does not settle down to its permanent state till it has reached some section A at a distance l from the entrance, and in this initial

TABLE 3 VALUES OF N AND x

Values of log DV/n	N	x
4.1 to 4.4	0.160	0.258
4.4 to 4.7	0.140	0.239
4.7 to 5.0	0.115	0.221
5.0 to 5.3	0.0911	0.201
4.1 to 5.4 (Authors' experiments)	0.0732	0.17

length the turbulence is not the same as later on but doubtless greater, in view of the sharp entrance. Hence the tube resistance in the initial length is also different from and probably somewhat greater than that in an equal length l farther along the tube.

In addition to this, a certain amount of power is required to accelerate the liquid and maintain the flow of kinetic energy which passes A and all later sections, and this requires a certain pressure drop p_a. The observed end loss is made up of this acceleration drop and the extra tube resistance e in the initial length l, so that we have

$$p_1 = p_a + e \quad \ldots\ldots\ldots\ldots \quad [14]$$

Without measurements of static pressure in the tube right along up to the entrance, there is no way to separate the observed p_1 into its two parts; but it is interesting, nevertheless, to ignore e and go on a little farther as if p_1 were due to acceleration alone, and identical with p_a.

The flow of kinetic energy past the section A is proportional to the density, and at any given average speed V it obviously depends on how the speed is distributed over the section. The acceleration drop depends on these same things, and in the simplest imaginable case, namely that of uniform speed, we have $p_a = 0.5dV^2$. The speed is, of course, never completely uniform, but the uniformity increases with the turbulence, *i.e.*, with the value of DV/n, so that $p_a = 0.5dV^2$ represents a limiting value which would be approached if DV/n were increased indefinitely.

For stream line flow, which is the permanent state in a smooth straight tube when $(DV/n) < 2000$, we know what the speed distribution is, and it is easily shown that $p_a = 1.0dV^2$, at least approximately, a value which agrees with experimental determinations.

Ordinary hydraulic flow represents something between these two extremes and we must therefore have

$$p_a = B\,dV^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [15]$$

where B is somewhere between 1.0 and 0.5, decreasing as DV/n increases, *i.e.*, as the flow becomes more turbulent and the speed more uniform.

If we now set $p_1 = p_a$, Equations [13] and [15] give us

$$B = \frac{144\,K_1}{d}$$

and the resulting values of B obtained from the authors' values of K are exhibited in Table 4.

TABLE 4 VALUES OF *B*

Temperature, deg. fahr.	85	100	130	160	190
Value of B	0.92	0.83	0.80	0.77	0.74

These values all lie between 1.0 and 0.5, and they change with temperature in the expected direction. For as the temperature rises, the kinematic viscocity decreases; DV/n increases; the turbulence increases; and the speed distribution approaches perfect uniformity, for which $B = 0.5$.

As regards the amount of the dependence of K_1 on the temperature, it may be fairly well represented by the purely empirical equation.

$$K_1 = 0.136\,dn^{0.21}$$

as is shown by Table 5.

TABLE 5 DEPENDENCE OF K_1 ON THE TEMPERATURE

Temperature, deg. fahr.	85	100	130	160	190
$0.136dn$. (calc.)	0.0118	0.0114	0.0107	0.0102	0.0098
K_1 (obs.)	0.0123	0.0111	0.0107	0.0101	0.0097

It is very satisfactory to find that when these valuable experimental results are examined by the dimensional method they fall well in line with what is already known and do not have to stand isolated by themselves.

No. 1715

SLOW-SPEED AND OTHER TESTS OF KINGSBURY THRUST BEARINGS

By H. A. S. Howarth, Pittsburgh, Pa.
Member of the Society

This paper presents operating and experimental data which show the wide range of application of Kingsbury thrust bearings, and the author gives particulars of typical installations which have been in successful operation since 1911. *Wide application of these bearings has been made to vertical and horizontal hydroelectric units, horizontal steam turbines, centrifugal pumps, and ship-propelling shafts. To determine some of the service conditions a series of slow-speed tests was recently made on a bearing carrying a load of* 10,000 *lb. The area of the supporting thrust-bearing segments was varied from* 71.5 *sq. in. to* 10 *sq. in. in order to test unit pressures ranging from* 140 *lb. to* 1000 *lb. per sq. in. The speed was varied from* 0.1 *r.p.m. to* 16 *r.p.m., long runs being made at the lowest speeds.*

THE Kingsbury thrust bearing in its simplest form consists of one or more pivoted segments or shoes against which the thrust collar presses as it rotates. The bearing faces are copiously supplied with oil so that perfect film lubrication takes place with its resulting low friction coefficient. This bearing was invented many years ago by Albert Kingsbury, of Pittsburgh, but it met with so much conservatism on the part of engineers and manufacturers that it was slow to be taken up. The author of this paper believes there are many engineers wrestling with thrust problems who will welcome data on the wide range of usefulness of this pivoted-segment type of thrust bearing. Whether the load be great or small, or the speed be high or low, it can be applied successfully. It is the object of this paper to present operating and experimental data to show the wide range of its application to date.

2 A Kingsbury thrust bearing was applied to a billet mill in 1911 to take the bevel-pinion thrust on the horizontal drive shaft. Its characteristics are as follows: load 66,000 lb.; 70 r.p.m.; unit

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

thrust load, 100 lb. per sq. in.; mean surface speed, 8.4 ft. per sec. The collar is of air-furnace iron and the pivoted shoes are babbitt-faced. It runs in machine oil. The heat of friction is dissipated by its connection with an oil-circulating system, but the bearing will lubricate itself automatically from an oil reservoir in its housing if the central-station system gets out of order. This bearing has been in successful operation ever since 1911, none of its parts has ever been replaced and today it is in first-class condition.

3 The next application of the bearing was in a plate-glass grinding and polishing machine in 1911. This bearing, of the vertical type, was placed below the lower end of the spindle. It was designed to carry a load of 160,000 lb. at 35 r.p.m. with a unit thrust of 980 lb. per sq. in. Its mean surface speed is 2.1 ft. per sec. On account of the high unit pressure and slow speed the collar was made of chilled iron and the shoes of bronze, and the oil used was very heavy. Its service requires alternate runs of two to three hours and stops of about one hour. A conservative estimate of the starts and stops of this machine to date is 15,000. This bearing is still in service and none of its parts has ever been replaced.

4 Following the above applications the Kingsbury thrust bearing began to be applied to vertical and horizontal hydroelectric units, to horizontal steam turbines and centrifugal pumps. During the war its use in these fields was greatly extended and it began to be employed to take propeller thrust. When our country entered the war the demand for turbine-driven ships advanced with a bound, which caused a similar increase in the demand for Kingsbury thrust bearings both for the turbines and for taking the propeller thrust.

5 Extensive application of these bearings was made in naval vessels and a notable demonstration of their capabilities was made at Hoboken by the Navy Department and witnessed by the naval engineering ensigns who were being trained at Stevens Institute. The interned twin-screw German ship, *Prince Eitel Friedrich*, driven by reciprocating engines, had thrust bearings with 21 collars, 14 of which were used to take the ahead thrust and the other 7 for the astern thrust. One of these was taken out and replaced by a Kingsbury bearing, the single collar of which took the full propeller thrust in either direction. Both engines were run with the same mean effective pressure and speed, the ship being moored to the dock.

6 The development of the Kingsbury thrust bearing has now reached such a stage that it is being used in steam turbines to take the whole steam thrust, no dummies whatever being required. Such

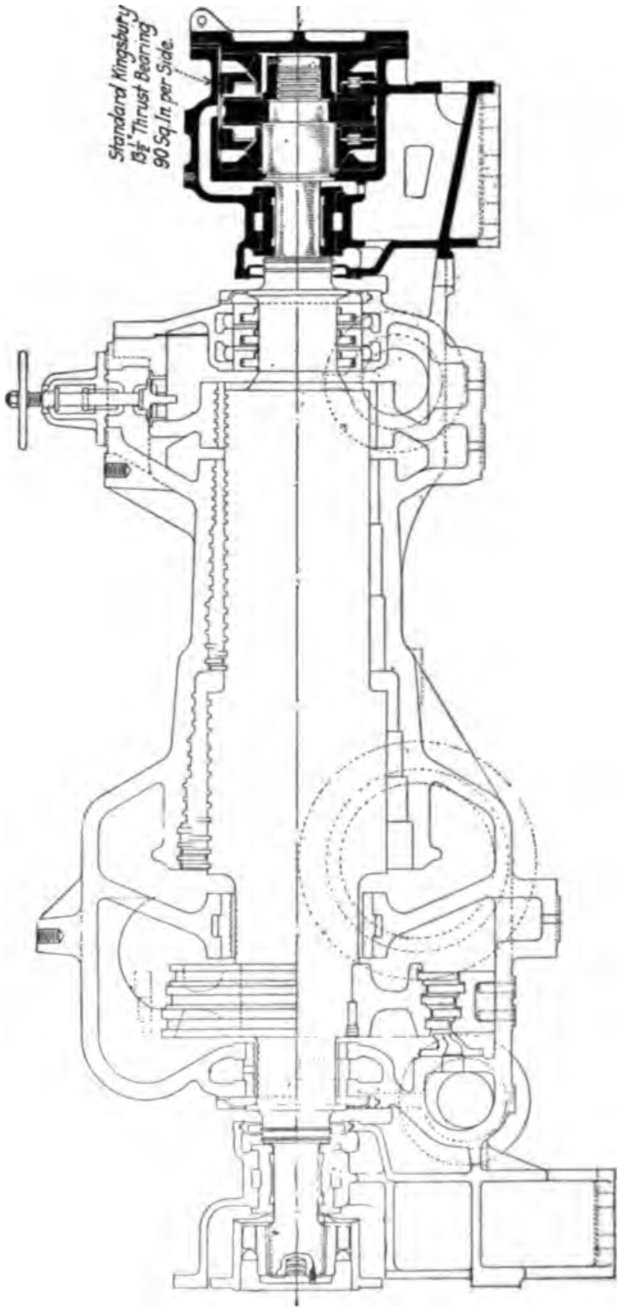


FIG. 1 2800-HP. PARSONS TURBINE WITH KINGSBURY BEARING FOR TAKING THE WHOLE STEAM THRUST a turbine is illustrated in Fig. 1. It runs at 3600 r.p.m., the mean surface speed in the thrust bearing being 159 ft. per sec. This is the highest speed attained thus far, but there is no reason to believe it cannot be greatly exceeded. High surface speed is not always

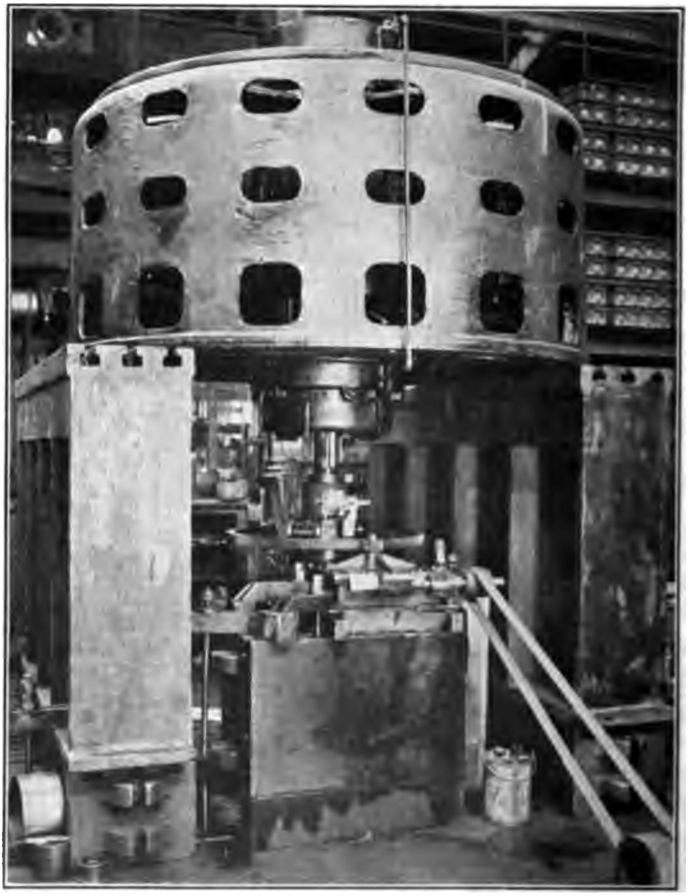


Fig. 2 View of Machine Employed in Testing Thrust Bearings at Slow Speeds

accompanied by high angular velocity. Tests are now being made that show the bearings will operate successfully at 15,000 r.p.m.

7 For [illegible] low-speed operation the best operating example is the plate glass-machine bearing described in Par. 3, whose mean surface speed is 2 [illegible] ft. per sec. Much lower speeds occur when the

machine slows down to a stop. A mean operating surface speed of 2.1 ft. per sec. is not low for a heavy oil, however, even with a unit load of 1000 lb. per sq. in.

8 Hydroelectric units ordinarily run on light engine oils and the film thicknesses are therefore much less at any speed and load than would be the case if heavy oils were used. They have to start

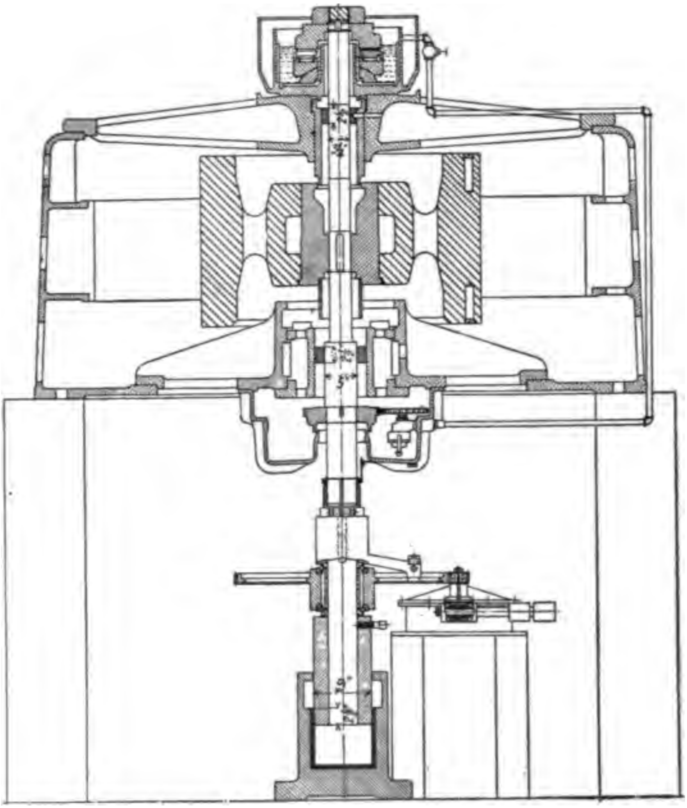


FIG. 3 VERTICAL SECTION OF THRUST-BEARING TESTING MACHINE SHOWN IN FIG. 2

and stop under practically full thrust load; consequently that service may be considered severe for thrust bearings of this type.

9 When one of these units slows down to a stop it passes gradually through the whole range of speeds from normal to a very low speed from which it is quickly brought to rest by a rapid increase of the turning resistance due largely to friction in the thrust bearing.

10 Consideration of the above service conditions suggests the following questions.

a At what surface speed does the friction force begin to increase rapidly when a bearing is slowing down under load?

b What happens to the bearing if it is forced to continue running at that low speed?

c Is there a sudden increase in friction when a definite low speed is reached?

d Does the oil film actually break?

11 The above questions are important. The slow speed tests to be described later were undertaken for the purpose of answering them. An effort was made to cover the field between film lubrication and so-called metallic friction.

SLOW-SPEED TESTS

12 The results of the slow-speed tests may be briefly summarized as follows:

a Unit pressures ranging from 140 lb. to 1000 lb. per sq. in. were employed using machine oil as a lubricant.

b Unit pressures of 600 lb. and 1000 lb. per sq. in. were employed using extra heavy turbine oil and also cylinder oil as lubricants.

c A low speed of 0.34 r.p.m., corresponding with a mean surface speed of 0.175 in. per sec., was maintained continuously for hours, at pressures of 140 lb. to 1000 lb. per sq. in., with machine oil as a lubricant. The bearing surfaces were in every case improved by the treatment.

d A low speed of 0.10 r.p.m., corresponding with a mean surface speed of 0.0515 in. per sec., was maintained all day on two successive days with a unit pressure of 140 lb. per sq. in., using machine oil as a lubricant. The bearing surfaces were improved by the treatment.

e Improvement of the bearing surfaces during the low speed runs was accompanied by a marked reduction of the friction.

f No sharp line of demarcation was found between fluid friction and so-called solid friction.

g At very low speeds lower friction coefficients can be obtained with heavy oils than with light ones. So far as the range of these experiments extended, the lower the speed the heavier the oil that will give the minimum friction.

h The friction coefficients obtained in these tests include the friction of the three guide bearings which it was impossible to measure or estimate accurately. This friction may have been a large part of the total friction measured for speeds from 2 to 16 r.p.m.

i Measurements of starting friction showed it to vary from 0.15 to 0.20.

13 The machine used for the slow-speed tests to be described is shown in Figs. 2 to 7, inclusive. Power from a variable-speed motor is belted and geared down so as to produce constantly any desired low speed at the driving gear *A* which is mounted on ball bearings around the rotor shaft *B*, Fig. 4. A hub *C* is keyed to the shaft as shown in Fig. 5. It has arms *D* and *E* while the gear arm

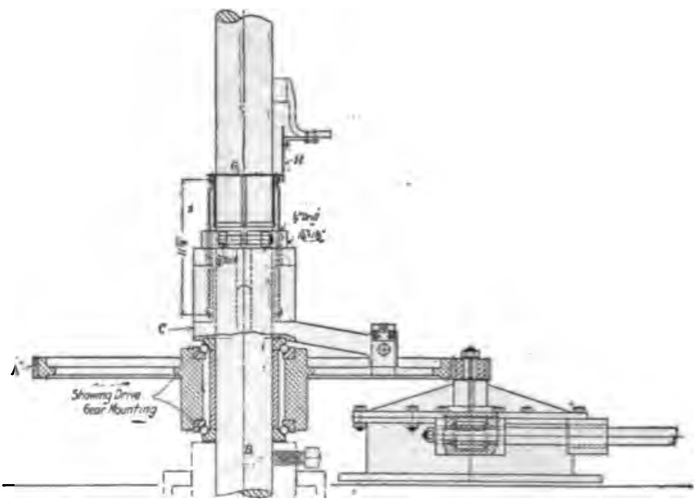


FIG. 4 DETAILS OF DRIVE-GEAR MOUNTING OF TESTING MACHINE SHOWN IN FIG. 3

has a lug *F*. Between lug *F* and arm *D* are mounted two compression springs in a holder that is so arranged as not to bind when the springs are compressed. Angular movement between the hub and gear, by means of a suitable linkage, causes a corresponding axial motion of the ring-shaped indicator *G* that surrounds the shaft as in Figs. 4 and 6. A fixed scale *H* is fastened to the machine frame. This can be graduated so the torque can be read directly. Since the load is constant the friction coefficient can be read directly on the scale if desired.

14 It will be noted in the section shown in Fig. 7 that two springs are used, a light one and a heavy one. When the driving gear is started its arm lug *F* moves to the right and begins to com-

press the springs. The light one compresses first and then the heavy one also comes into action. When the compression of both becomes great enough, rotation begins and the springs expand as the coefficient of friction reduces from its starting values to its running value. The coefficient of running friction is so very low for a continuous film that the light spring will be compressed but little.

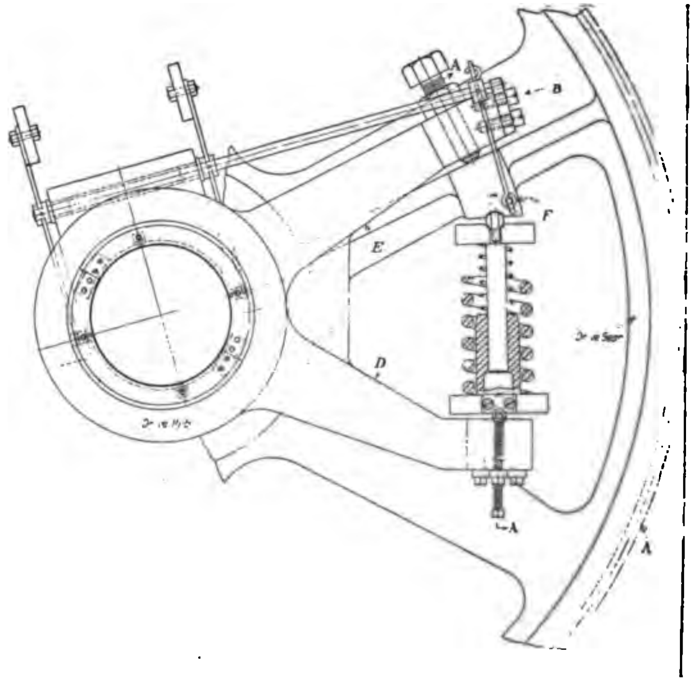


FIG. 5 PORTION OF DRIVE GEAR, SHOWING METHOD OF MOUNTING THE COMPRESSION SPRINGS

15 The following action was noted during certain tests at extremely low speeds. As soon as the starting resistance was overcome the shaft would be sufficiently accelerated by the expansion of the stiff spring to be driven by the light one. As the driving gear would not be turning fast enough to keep up with it the shaft would slow down till the friction became high enough to bring it to rest. Then the driving mechanism would catch up and compress the springs sufficiently to start the shaft turning again. This cycle would be repeated rapidly or slowly according to the speed and stiffness of the springs used.

16 Three grades of oil were used.

a No. 14 machine oil, referred to as "light oil" and "machine oil"; Saybolt viscosity, 534.5 sec. at 71 deg. fahr.

b A mixture of machine and cylinder oils, referred to as

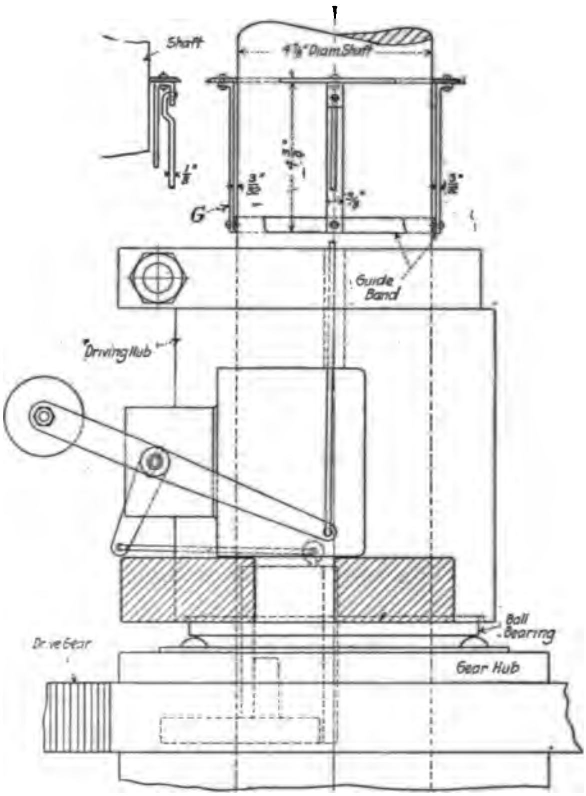


FIG. 6 DETAIL OF INDICATOR, SHOWING ANGULAR MOVEMENT BETWEEN HUB AND GEAR OF THRUST-BEARING TESTING MACHINE

"medium oil," and "extra heavy turbine oil"; Saybolt viscosity, 937 sec. at 71 deg. fahr.

c Acme cylinder oil, referred to as "heavy oil" and "cylinder oil"; Saybolt viscosity, about 9000 sec. at 71 deg. fahr.

17 The temperature of the oil bath surrounding the thrust bearing did not vary more than 2 deg. fahr. from 71 deg. fahr. during the tests.

18 The rotating weight was about 10,000 lb. The thrust

bearing had four shoes of the form shown by Fig. 8, their total area being 71.5 sq. in. The initial tests were run with all four shoes in place, the unit pressure amounting to 140 lb. per sq. in. Subsequently two opposite shoes were removed so as to increase the pressure to 280 lb. per sq. in. In order to further increase the pressure, the babbitt faces were reduced first to bring the unit loading up to 600 lb. and finally to 1000 lb. per sq. in.

19 Before making the tests the thrust bearing, which is shown in section by Fig. 9, was carefully inspected. The face of the cast-iron collar was in excellent condition, i.e. smooth and flat. The babbitt-faced shoes were then fitted to the collar face under load. This fitting process developed some interesting features. Not knowing at how low a speed the machine would run and have a perfect film, it was assumed that the high spots would rub at 2.8

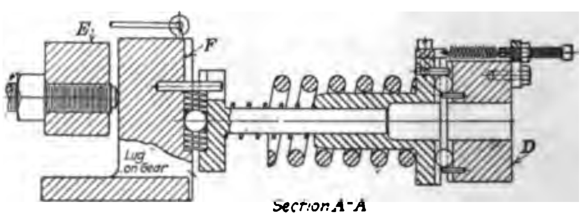


FIG. 7 MOUNTING OF COMPRESSION SPRING IN THRUST-BEARING TESTING MACHINE

(Section *A–A* in Fig. 5)

r.p.m. The machine was run at this speed for 30 min. The friction was so high that the load came on the heavy spring. The shoes were then removed and the high spots scraped down slightly. The shoes were then put back and the bearing run at 2.9 r.p.m. for 15 min. They were then removed and scraped again, and run at 3 r.p.m. After the starting friction was overcome the load was carried on the light spring and the speed kept constant for 45 min. The spring compression gradually reduced from $\frac{9}{16}$ in. to $\frac{3}{8}$ in., showing that the friction was decreasing rapidly. This indicated that such high spots as interfered with each other at this speed were gradually being rubbed down. The results of this first test are plotted in Fig. 10. The shoes were inspected after this run and found to be in excellent condition.

20 The next test was run on Saturday, October 25, beginning with 2.86 r.p.m. This speed was maintained for an hour. See Fig.

11. The speed was then reduced to 2 r.p.m. and continued there for an hour. The spring compression decreased to $\frac{5}{16}$ in. It was expected that the compression would decrease with the speed so long as a good film was maintained. The machine was then run

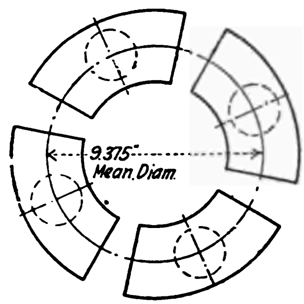


FIG. 8 ARRANGEMENT OF SHOES IN THE THRUST BEARING TESTED

for an hour at 1.9 r.p.m., and the light-spring compression was $\frac{15}{64}$ in., showing that the friction coefficient still continued to reduce. This run had to be stopped on account of quitting time.

21 Tests were continued on Monday, October 27, starting

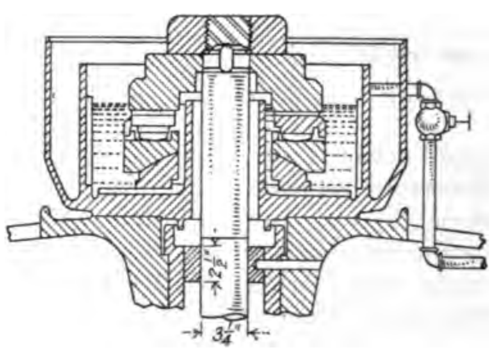


FIG. 9 SECTION THROUGH THRUST BEARING EMPLOYED IN THE TESTS

with 2.72 r.p.m. See Fig. 12. The r.p.m. was decreased at hourly intervals at first. The first run showed that the friction decreased until the speed was 2 r.p.m. When an attempt was made to run at 1.81 r.p.m., the friction increased considerably — about 100 per cent. The friction fell off with the speed down to 1.93 r.p.m. Then at 1.87 r.p.m. the friction increased about 100 per cent. At 1.66 r.p.m. it was 20 per cent higher still. It will be noted that with a speed as low as 1.66 r.p.m. the light spring still carried the load easily.

22 The results of Tuesday's tests correspond very closely with the previous ones. See Fig. 13. The speed was lowered at intervals until it reached 1.9 r.p.m., the friction coefficient likewise decreasing. When an attempt was made to run at 1.71 r.p.m., the spring compression increased more than 100 per cent. At 1.5 r.p.m. it increased to almost double its value at 1.71 r.p.m.

23 After the run on Tuesday the thrust bearing was inspected. It was found to be in excellent condition, the shoe faces having apparently worked in so as to be a much better fit to the collar. In

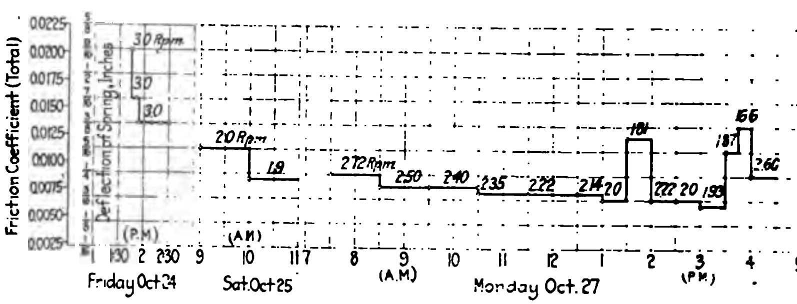


Fig. 10 Fig. 11 Fig. 12

Figs. 10 to 12 Results of Tests of Thrust Bearing

view of this improvement it was decided to see what would happen if the machine were run constantly at 1 r.p.m. It was thought that the friction would be very high. Whether it would increase or decrease as the run was continued was the problem. The machine was run at 1 r.p.m. from 7.20 a.m. to 4.30 p.m. See Fig. 14. At the beginning of the run the heavy spring was slightly compressed. At the end of four hours the load was beginning to be carried by the light spring alone. At the end of another four hours the compression of the light spring had reduced considerably, finally reaching $\frac{1}{16}$ in. This compression remained constant for 30 min. It was decided that from this point on the lowering of the coefficient would be slow. Hence this test was discontinued at quitting time. As there was an evident improvement in the bearing surfaces, shown by the gradual reduction of the friction coefficient, it was decided to run the next day at various speeds and see whether the coefficient would reduce with the speed to a lower point than before.

24 On Thursday the machine was started up and run for 15-min. intervals at each speed beginning with 2.25 r.p.m. See Fig. 15. The friction coefficient was the same as shown at this speed on Tuesday. The minimum friction was obtained at 2 r.p.m. The

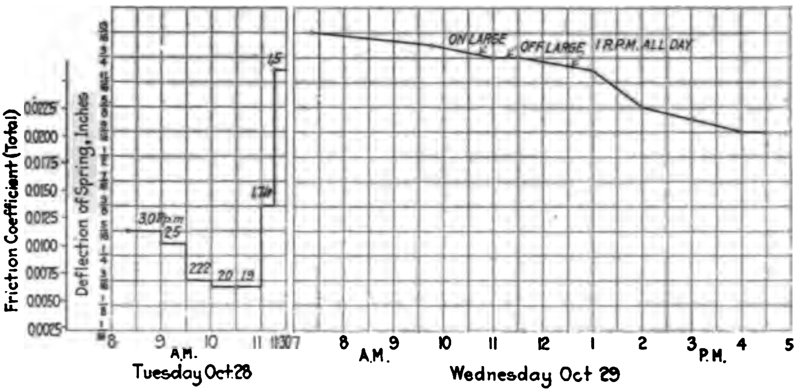


FIG. 13 FIG. 14

FIGS. 13 AND 14 RESULTS OF TESTS OF THRUST BEARING

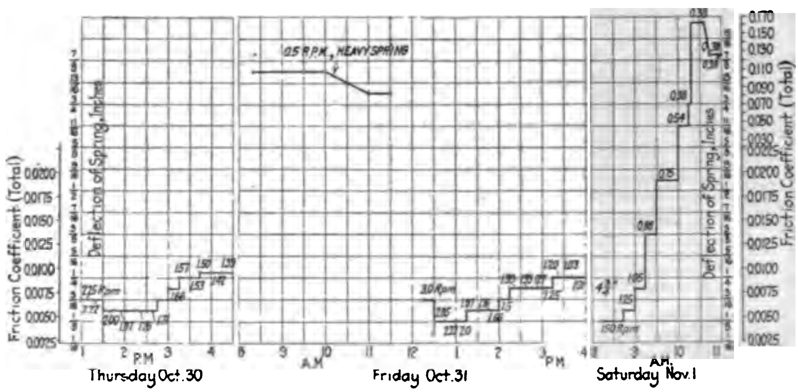


FIG. 15 FIG. 16 FIG. 17

FIGS. 15 TO 17 RESULTS OF TESTS OF THRUST BEARING

coefficient remained constant, however, until the speed reached 1.71 r.p.m. It increased appreciably when the speed was reduced to 1.66 r.p.m. and continued to increase, though slowly, until the speed reached 1.5 r.p.m. The friction then remained constant until the speed was reduced to 1.39 r.p.m. At this point the machine was shut down

25 It is evident from the foregoing that by running continuously at 1 r.p.m. the bearing surfaces improved so that the friction coefficients at speeds slightly above 1 r.p.m. were less than they otherwise would have been.

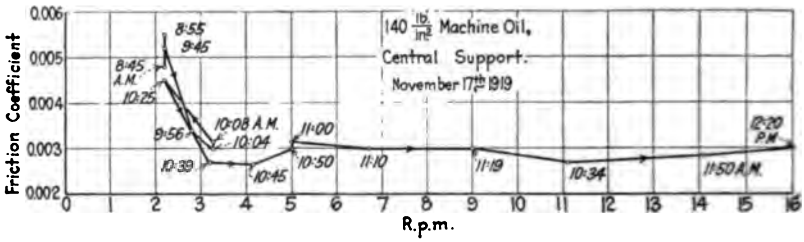


FIG. 18 RESULTS OF TEST RUN OF 3½ HOURS; SPEEDS, 2.2 TO 16 R.P.M.

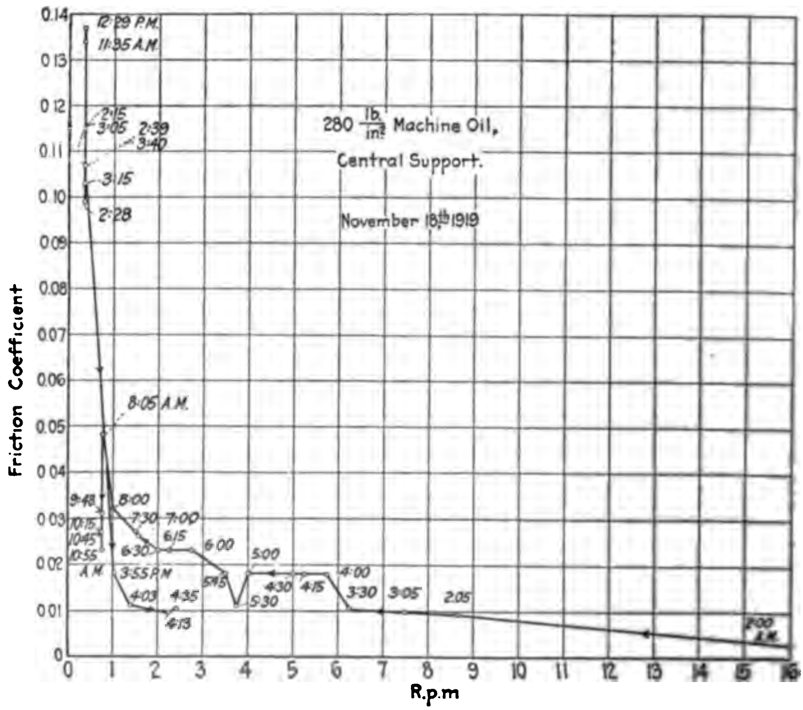


FIG. 19 RESULTS OF TESTS AT 280 LB. PER SQ. IN. WITH MACHINE OIL

26 On Friday, October 31, the machine was run at 0.5 r.p.m. for 4½ hours. See Fig. 16. The load was carried by the heavy spring throughout the run, but its compression gradually decreased.

The machine was then run at various speeds, beginning with 3 r.p.m., to see if the bearing had been improved by its long run at 0.5 r.p.m. The bearing was to be maintained at each speed for 15 min. The friction force at low speeds was less than it had been in previous tests, but it remained constant from 2.85 to 2 r.p.m. It then slightly increased as the speed was further lowered, and remained constant from 1.87 to 1.66 r.p.m. From here on it gradu-

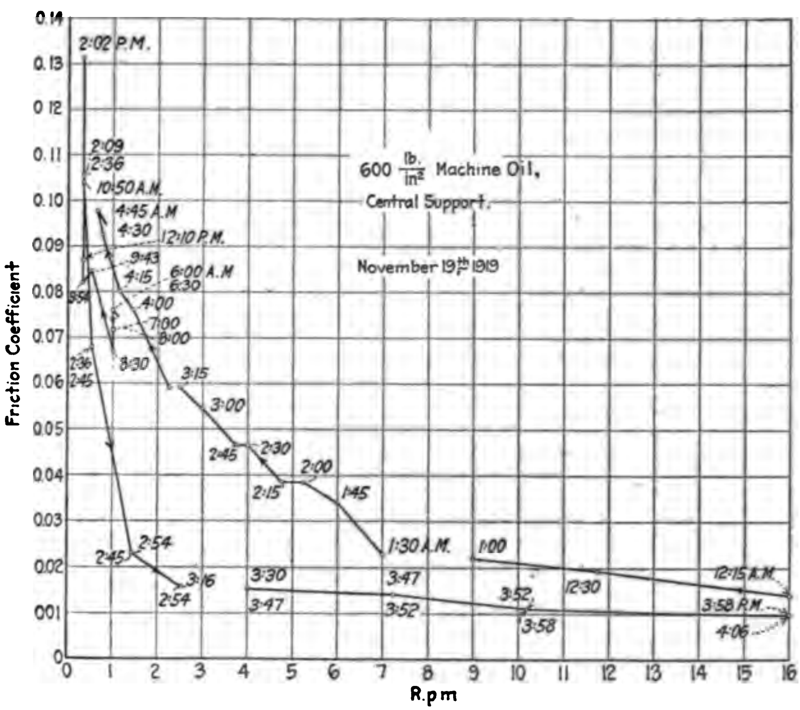


FIG. 20 RESULTS OF TESTS AT 600 LB. PER SQ. IN. WITH MACHINE OIL

ally increased, but remained very low until the speed had been reduced to 1.01 r.p.m. At this low speed the friction coefficient was about one-third as great as the minimum value that had been obtained when the bearing had been run steadily for a whole day at 1 r.p.m. The unit was then shut down.

27 On Saturday the test was begun at 1.5 r.p.m. and continued for 30 min. See Fig. 17. The coefficient was less than at same speed the previous day. This is explained by the assumption that the surfaces improved during the lower speed runs following that at

1.5 r.p.m. on Friday. At 1.25 r.p.m. the friction was still considerably less than on the previous day. At 1.05 r.p.m. it was nearly as great as on Friday. At 0.88 r.p.m. it increased about 70 per cent and continued to increase rapidly as the speed was lowered until 0.38 r.p.m. was reached. The motor was not belted to run slower than this, so the run at this speed was continued. The friction coefficient at 0.38 r.p.m. gradually decreased as it had done at

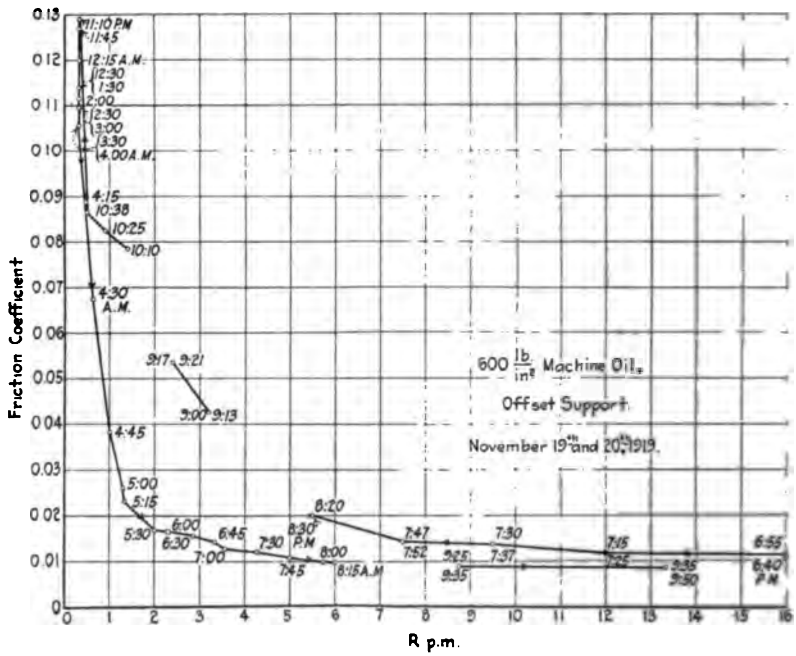


FIG. 21 RESULTS OF TESTS AT 600 LB. PER SQ. IN. WITH MACHINE OIL

other speeds, indicating continued improvement in the bearing surfaces.

28 On Tuesday, November 4, a continuous run of 5½ hr. was made at 0.1 r.p.m. The friction coefficient gradually reduced from 0.178 to 0.130. The run was continued the next day for 6½ hr. at the same speed. The friction coefficient remained steady at 0.130.

29 After the long runs at 0.1 r.p.m., the machine was put through the various higher speeds from 1.0 to 8.5 r.p.m. The resulting friction coefficients were considerably lower than previously obtained for the same speeds. Hence the long run at 0.1 r.p.m. resulted in marked improvement in the bearing.

30 The test results thus far given, shown by Figs. 10 to 17, were plotted with the vertical scales giving spring deflection and corresponding friction coefficient, and with the horizontal scale showing time of day. The figures that follow show the friction coefficient, to scale, plotted against the r.p.m. The time of day is marked along the curve, at each reading, to indicate the sequence. Arrow heads are also used.

31 In Fig. 18 are plotted the results of a test run of 3½ hours, through a speed range from 2.2 r.p.m. to 16 r.p.m. The lowest coefficient obtained was 0.00265 at 4 r.p.m. This includes, as do

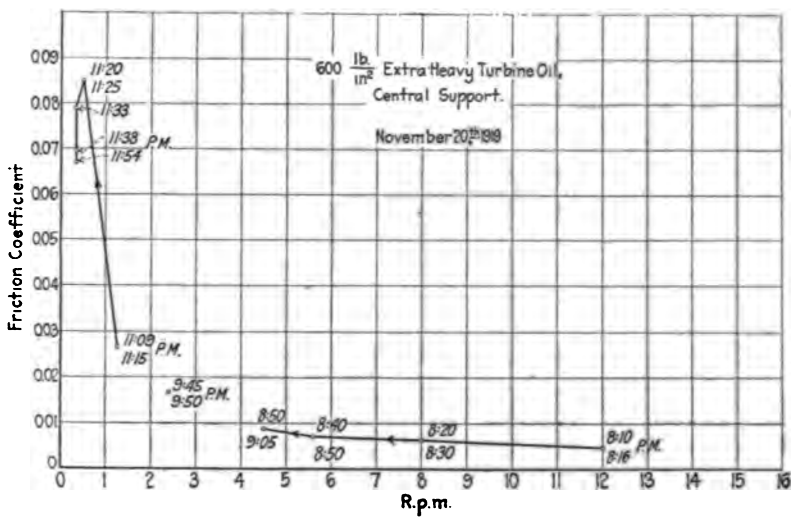


FIG. 22 RESULTS OF TESTS AT 600 LB. PER SQ. IN. WITH EXTRA TURBINE OIL

all the others given in this paper, the guide bearing friction, which it was impossible to estimate accurately. It will be noted in this figure that the friction remained fairly constant from 3 r.p.m. to 16 r.p.m. Similar phenomena will be noted in other figures.

32 In Fig. 19 are shown the first results of tests at 280 lb. per sq. in. with bath of machine oil. This test continued 14½ hours. All the coefficients are higher than for a unit pressure of 140 lb. This is to be expected at the low speeds of these tests, using a light oil. The order of the changes in speed should be noted carefully. Starting at 16 r.p.m. it was reduced at intervals till it reached 0.76 r.p.m., the friction being higher at each lower speed. The speed

was maintained at 0.76 for nearly three hours, the coefficient of friction gradually reducing till it was 50 per cent of its first value. The speed was then reduced to 0.34 r.p.m. and kept there for four hours. The friction while much higher than for 0.74 r.p.m. reduced

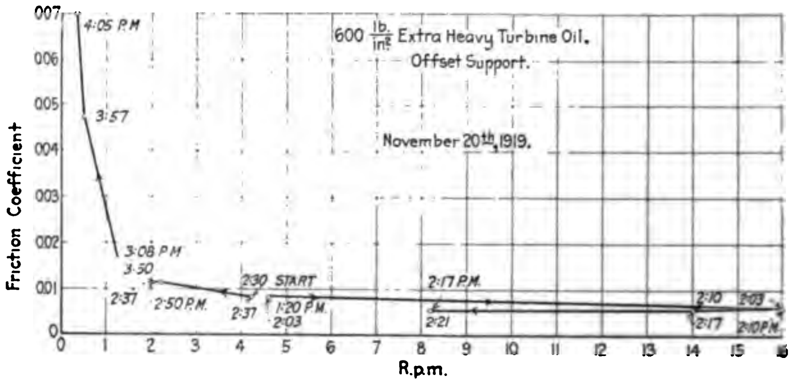


FIG. 23 RESULTS OF TESTS AT 600 LB. PER SQ. IN. WITH EXTRA TURBINE OIL

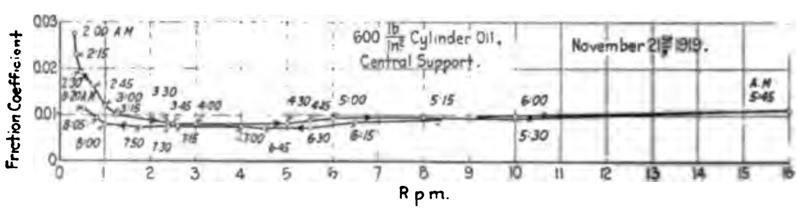


FIG. 24 RESULTS OF TESTS AT 600 LB. PER SQ. IN. WITH CYLINDER OIL

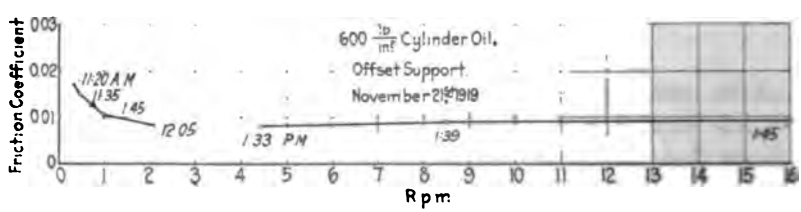


FIG. 25 RESULTS OF TESTS AT 600 LB. PER SQ. IN. WITH CYLINDER OIL

about twenty per cent in four hours. The speeds were then increased from 0.34 up to 2.4 at intervals. For each speed the friction was about half what it was prior to the long runs at 0.74 and 0.34 r.p.m., thus indicating improvement during the continuous slow speed runs.

33 In Fig. 20 are shown the results of a test with machine oil and a unit pressure of 600 lb. per sq. in. The results in general were the same as from the previous test, all the coefficients being higher however on account of the increased unit pressure. Considerable difficulty was experienced with surging at the low speeds during this

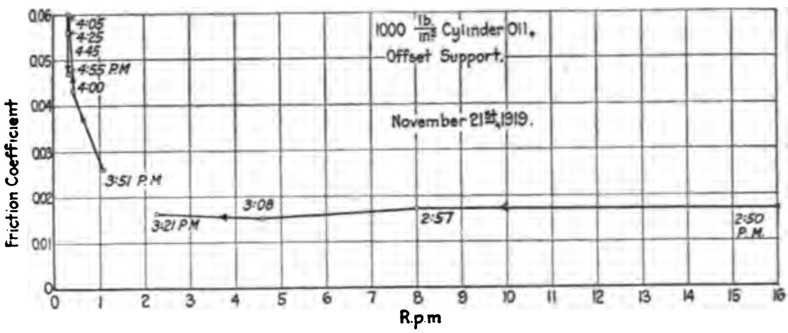


Fig. 26. Results of Tests at 1000 Lb. per Sq. In. with Cylinder Oil

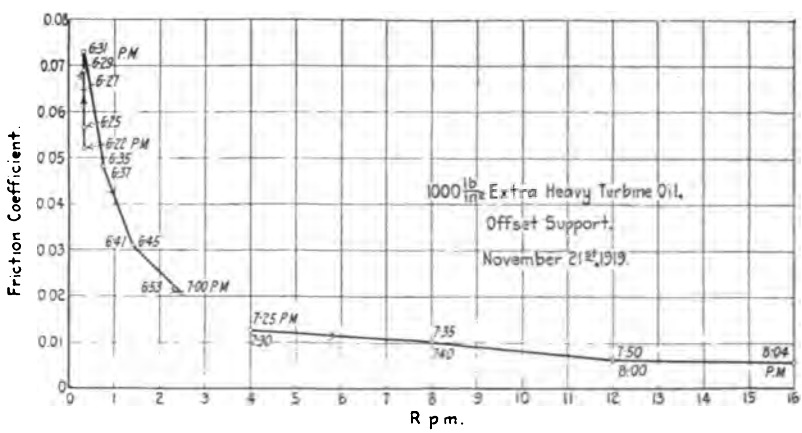


Fig. 27 Results of Tests at 1000 Lb. per Sq. In. with Extra Heavy Machine Oil

test accounting for the variety of friction coefficients at 0.34 r.p.m. The bearing was run all told about eight and a half hours at speeds ranging from 1 r.p.m. to 0.34 r.p.m. After that run the speed was increased at intervals up to 16 r.p.m., these new friction coefficients being in all cases very much less than before the low speed run. It should be noted that the improvement is much more marked than for the light pressures. In Fig. 21 the results are quite similar to those shown in Fig. 20 except that they are a bit more consistent. In this case the improvement is about as marked as before but the coefficients are slightly lower as a whole.

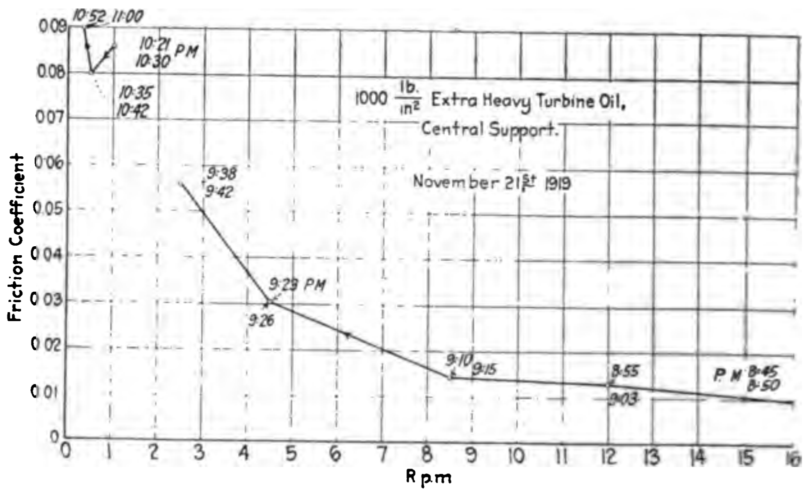


FIG. 28 RESULTS OF TESTS AT 1000 LB. PER SQ. IN. WITH EXTRA HEAVY TURBINE OIL (SHOES CHANGED)

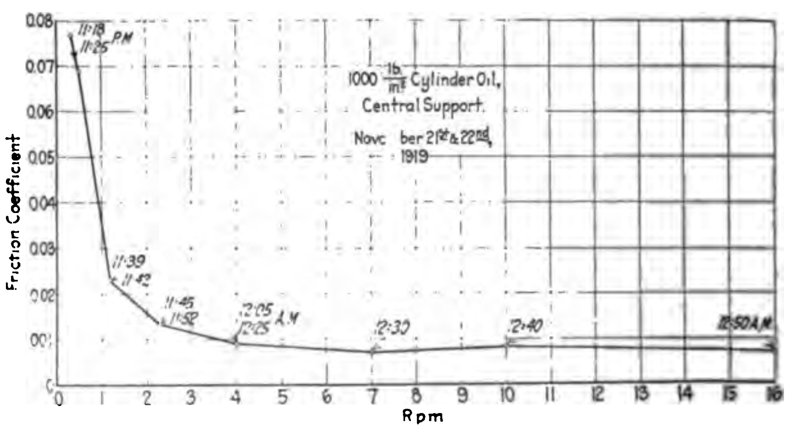


FIG. 29 RESULTS OF TESTS AT 1000 LB. PER SQ. IN. WITH CYLINDER OIL (SAME SHOES AS IN TESTS OF FIG. 28)

34 Figs. 22 and 23 show two tests run at 600 lb. per sq. in. with extra heavy turbine oil. The first noticeable feature is the reduction of friction, this being appreciably lower than for the lighter oil with

the same unit pressure. Unfortunately these tests were run in only one direction, i.e. from high to low speeds. Hence we cannot tell how much the bearing surfaces were improved by the low speed run. In Fig. 22 the friction fell off during the thirty-four minutes of running at 0.50 and 0.34 r.p.m.

35 Tests were then made with cylinder oil with a load of 600 lb. per sq. in. In Figs. 24 and 25 the results are plotted. They

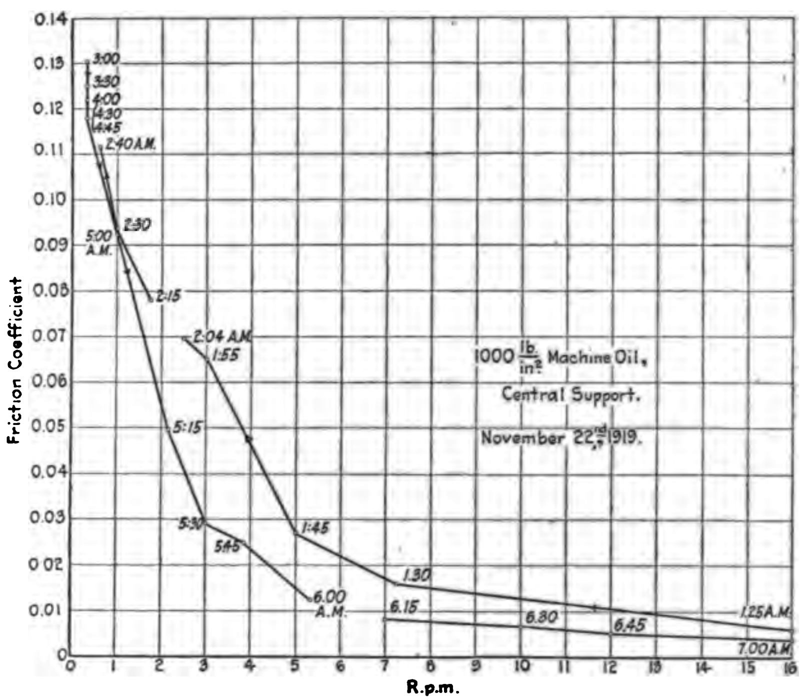


FIG. 30 RESULTS OF TESTS AT 1000 LB. PER SQ. IN. WITH MACHINE OIL (SAME SHOES AS IN TESTS OF FIG. 28)

show that for speeds below 3 or 4 r.p.m. the friction is less than for the same pressure using extra heavy turbine oil. In Fig. 24 the test was begun at slow speed, the speed increased up to 16 r.p.m. at intervals and then brought down again. The return readings show lower than the initial ones, the difference being most marked at very low speeds.

36 The unit pressure was next increased to 1000 lb. per sq. in. The first test, Fig. 26, was run with cylinder oil and begun at 16 r.p.m.

Burrs or high spots produced when the area of the babbitt was reduced may explain the high coefficient obtained at the start. The surface improved as the speed was lowered though not so much as expected.

37 The oil was then changed to extra heavy turbine oil. In Fig. 27 this test is shown. During the first few minutes the friction

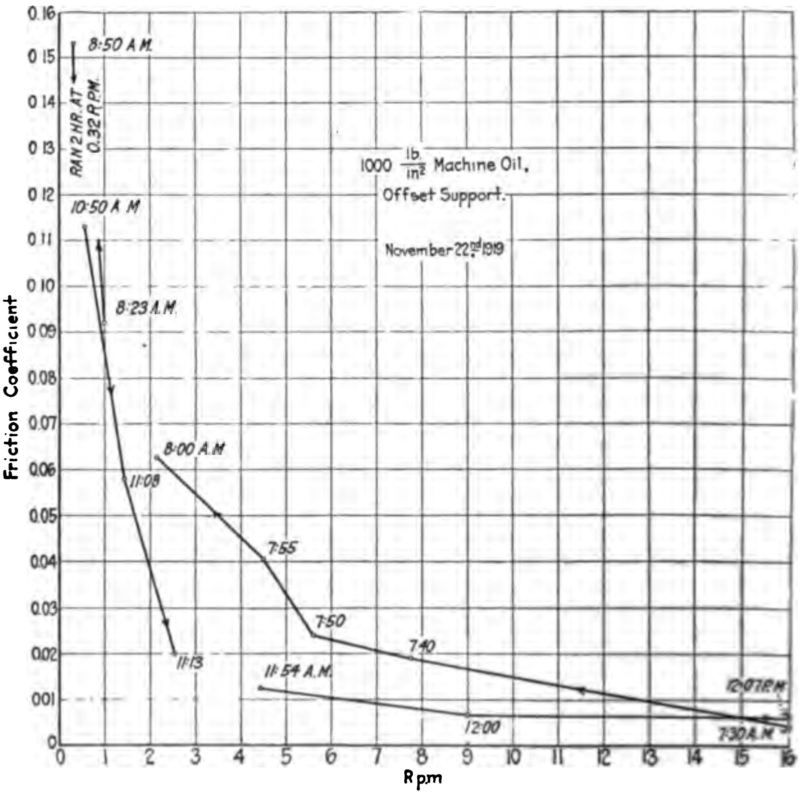


FIG. 31 RESULTS OF TESTS AT 1000 LB. PER SQ. IN. WITH MACHINE OIL (SHOES CHANGED)

at low speed increased, no doubt, while the cylinder oil was being washed off the bearing faces by the lighter oil. The speed was then increased and the friction gradually fell off as shown.

38 The shoes were then changed with same pressure and oil with the result indicated in Fig. 28. These shoes also probably had burrs or high spots produced when the babbitt area was cut down.

They would explain the high friction coefficients at medium speeds and the peculiar result at very low speeds.

39 The cylinder oil was then used without changing the shoes. They having been improved by the previous run showed lower coefficients at all speeds (see Fig. 29) than were obtained during the first run at 1000 lb. per sq. in. with cylinder oil as shown by Fig. 26.

40 The oil was next changed to machine oil and the same shoes used. See Fig. 30. Beginning at 16 r.p.m. the speed was reduced at intervals. A run was made of one and three quarters hours at 0.34 r.p.m., the friction dropping off gradually but noticeably. Going up through the speeds again gave the same general results as for lighter pressures, i.e. lower coefficients than before the low speed run.

41 The shoes were then changed and the same test repeated with the similar result shown by Fig. 31. This test completed the experiments.

42 A summary of the results of these slow-speed tests is given on page 8.

No. 1716

A DREDGING PUMP OF NOVEL CONSTRUCTION

BY WALTER J. WHITE,[1] NEW ORLEANS, LA.
Associate Member of the Society

THE port of New Orleans has long been denied its natural growth by restrictions imposed by the civil code of the state of Louisiana, which declares that the banks of all navigable rivers are public property and as such cannot be sold nor used for factory sites, shipyards, or other industrial enterprises. Attempts have been made to overcome this industrial handicap by proposals to build a navigable canal which would not be subject to the restrictions just enumerated. Some fifteen years ago Mr. Walter Parker proposed that such a waterway be built connecting the Mississippi River with Lake Pontchartrain, but the task was not undertaken until shortly before the war.

2 It is not the purpose of this paper, however, to deal with the history of this canal nor to present the many industrial advantages which its creation will bring to the port of New Orleans, but rather to present some of the outstanding engineering features incident to the completion of the work.

3 It may be of interest to note in passing, that in spite of the prevalent belief that the civil code has been a handicap to the general industrial development of New Orleans, even if there were no law standing in the way, manufacturing plants located upon the Mississippi would be exposed to many dangers occasioned by the encroachment of the river. Furthermore, any manufacturing plant due to the great range of level of the river would be burdened with the cost and inconveniences of loading and unloading of water craft and such fact would serve to make impracticable the operation of the plant. It is also generally conceded by experts in shipbuilding

[1] Supt. of Dredging, Board of Commissioners, Port of New Orleans.

Presented at the Annual Meeting, New York, December 1919, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

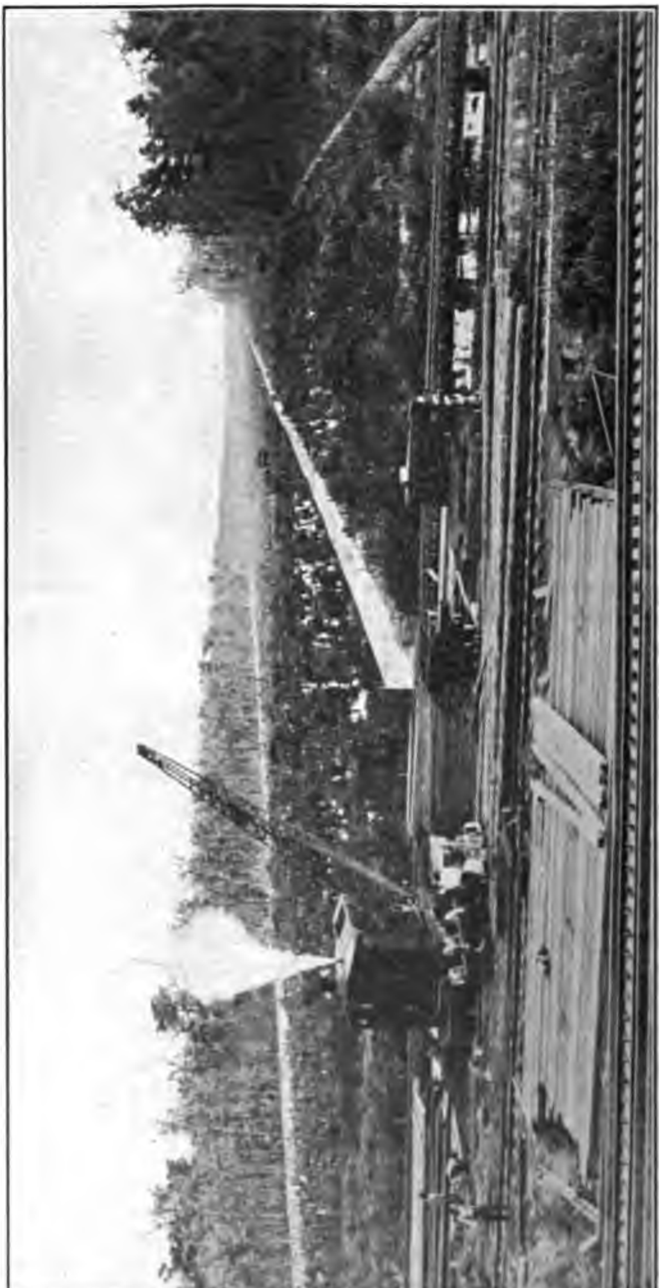


FIG. 1 VIEW OF CANAL SITE LOOKING SOUTH FROM THE RAILROAD

that operations of that kind could not be maintained on the banks of the river with any hope of success. It is of course obvious that an industrial canal (controlled by locks) would be entirely suitable for all such purposes.

4 During the construction of the canal the very unusual character of the dredged material necessitated a complete change in the construction of the centrifugal dredging pumps, and because the engineering features are particularly interesting, it is to this phase of the work that the writer wishes to direct attention; but before proceeding to examine the methods employed, a brief description of the canal itself will undoubtedly be of interest.

5 The surface elevation of that part of the city of New Orleans in which the canal is located is the same as mean gulf level. Occasional high tides, due to storms, occur when the water rises to an elevation of from three to five feet higher than the ground surface. It was therefore desirable that the dredge excavation be handled in such a way that the greatest benefit would be secured from the disposal of the material. This was accomplished by depositing the dredged material so as to form as nearly as practicable a continuous fill, which thus serves as a levee along both banks of the canal and which prevents any possibility of overflow during storm tides. The dredge excavation furthermore provides high, well-drained sites for shipyards and other industries. The length of the Inner Harbor-Navigation Canal from deep water in the Mississippi River to the 14-ft. contour in Lake Pontchartrain is 31,800 ft. The completed project is to have a clear channel width of 150 ft. on the bottom with a depth of 30 ft. below low water. In the development of shipyards and other industrial sites no encroachment on the channel width will be allowed, such locations being provided with launching basins, slips, piers, etc.

6 While it is true that the Mississippi River and Lake Pontchartrain will be connected, it is believed that the use of the word canal is misleading and very likely to be misunderstood. It should be remembered that Lake Pontchartrain is a shallow body of water when compared with the depth of 30 ft. in the inner harbor. The present plan contemplates the construction of an inner harbor, or basin having a constant level and connected with the deep-water channel of the Mississippi River, entrance from the river channel to the inner harbor being by means of one lock. After the harbor feature has been considerably developed, it is quite probable that extensive dredging operation will be undertaken to establish a deep-

FIG. 2 VIEW OF CANAL AFTER EXCAVATION

water channel from the north end through Lake Pontchartrain and Mississippi Sound to the Gulf, thereby justifying, in the fullest sense, the name Inner Harbor-Navigation Canal — an *Inner Harbor* having a constant level and a *Canal* connecting deep water in the Mississippi River with deep water through Lake Pontchartrain to the Gulf.

7 During the construction of the canal, traffic had to be maintained over railroads, electric-car lines, and highways which crossed the line of the Inner-Harbor. This feature and the necessity of utilizing the excavated material in levees and fills along the banks, and also the requirement that all excavated material be deposited well back from the edge of the bank, prohibited the use of dipper dredges or other types of dredges which dispose of material by means of scows or by depositing along edge of cut.

8 While planning methods by which this dredging could be done economically, and at the same time rapidly, several test pits were excavated which confirmed the belief that about 80 per cent of the area to be excavated was underlaid with large cypress stumps which at some time in the past had either subsided or had been covered by deposit from overflow. In addition to the submerged stumps there was a heavy surface growth of cypress and other timber from Station No. 88 to Station No. 221. The topography of this country is best shown by Figs. 1 and 2. After clearing the standing timber from the area to be dredged tentative efforts were made to remove all stumps by shooting and grubbing, but it was found that this would greatly increase the cost of the work. During the progress of the work very little manual labor was expended on stumps and roots, although it was found to be advantageous to blast some of the larger surface stumps.

9 Notwithstanding the great number and character of stumps and roots, as evidenced by Fig. 3, it was decided for the reasons mentioned that the work could best be done by hydraulic pipe-line dredges of the cutter type and contractors were asked to submit bids on the work. Due to the difficulties mentioned, however, contract prices could not be secured and arrangements were therefore made to charter dredges, the Board of Port Commissioners accepting all responsibility for output. Four 20-in. dredges, and one 22-in. dredge, the *Texas,* were accordingly secured and work was commenced on May 15, 1918. Fig. 4 shows one of these dredges, the *Captain Huston.* The dredges were powerful machines of good design in all respects, but it was fully realized that much trouble was to be expected on account of stumps and roots.

10 By utilizing Bayou Bienvenue four of the dredges were brought in to a point on the canal near Station No. 88, where one was placed in operation headed towards Lake Ponchartrain and three were started toward the Mississippi River; one dredge was also started south from Lake Ponchartrain. On account of larger areas in suction and discharge pipes and pump, the 22-in. dredge *Texas* was placed in the most difficult section. After allowing a reasonable time for dredge and crew to become adjusted to conditions, it became evident that progress would be slow and the work would be expen-

Fig. 3 Character of Excavation Encountered

sive, due entirely to delays and decreased capacity occasioned by stumps and roots clogging pump throat and suction. The soil itself was easily handled by a hydraulic dredge and the pipe-line conditions were not difficult, but the amount and character of roots encountered and later successfully handled is beyond description. Fig. 5 shows the character of the material actually pumped through the dredge *Texas*. It was thought that the solution of this difficulty would be found in a pump impeller designed with large and easy passage areas and having no projecting webs on vanes or hub. The writer learned that Mr. A. B. Wood, mechanical engineer for the

Sewerage and Water Board of New Orleans, had designed and patented a centrifugal-pump impeller for handling sewage containing trash. An impeller for a 12-in. pump was inspected, and while none had heretofore been used on dredges of the cutter type it was believed that the root problem was solved and Mr. Wood was engaged to design an impeller for the 22-in. dredge *Texas*. All costs in connection with the manufacture of the impeller were to be borne by the Board of Port Commissioners regardless of whether or not it proved a success.

11 From the first the results obtained with the new impeller

FIG. 4 THE DREDGE *Captain Huston*, USED IN THE EXCAVATION OF THE NEW ORLEANS INDUSTRIAL CANAL

were remarkable, the increase in output being between two and three hundred per cent. The increased yardage excavated by dredge *Texas* is shown by the following comparative statement, covering thirty days' operation immediately before and immediately after change of impellers:

	Old Impeller	New Impeller
Excavating, hours	482.43	394.00
Clogged suction delay, hours	130.75	71.50
All other delay,[1] hours	106.82	254.50

[1] The large number of delay hours after installation of new impeller was due to time lost cleaning boilers and raising a sunken fuel barge. For the period

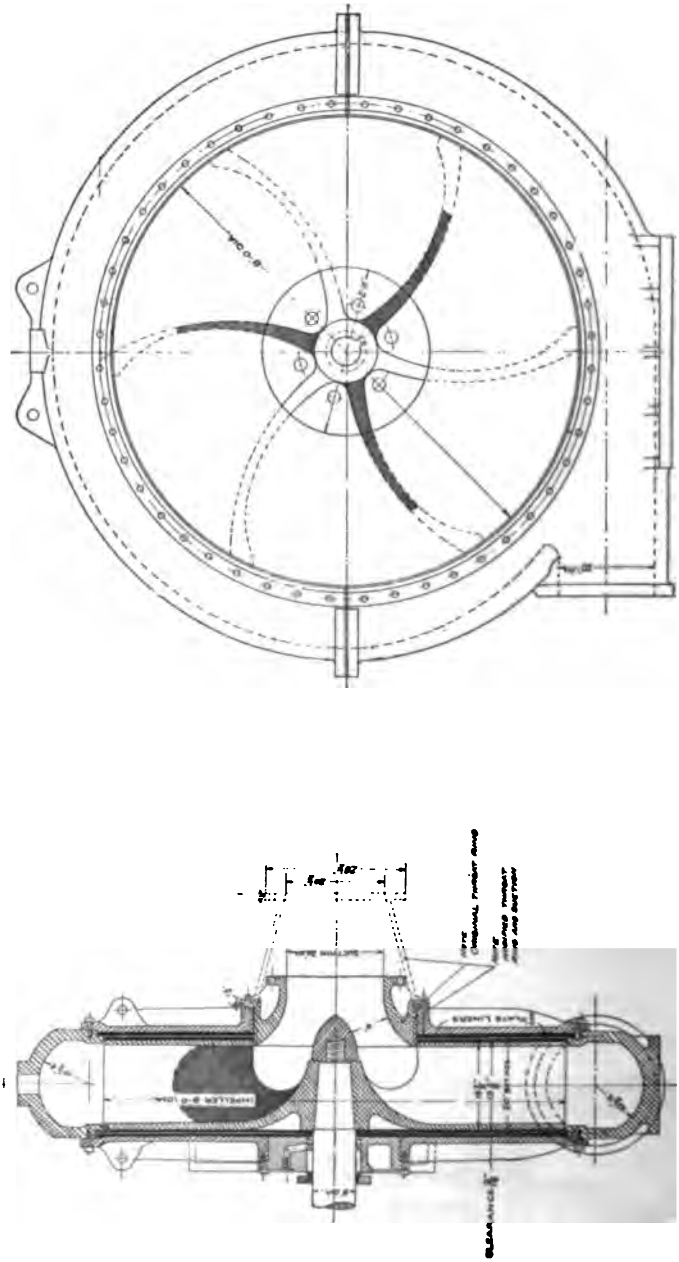


Fig. 7 Details of Pump on Dredge *Captain Huston* Showing the Modified Throat and Impeller

tioned, it was recognized that in order to secure maximum output in material of this character, it would be necessary to provide as large passages through pump and impeller as possible, and it was, therefore, determined that two-thirds of each alternate vane should be cut out of the impellers in these dredges, as shown in Figs. 6 and 7. The outer one-third of the vanes was left to preserve structural strength, but after observing the performance of the pumps it is the opinion of the writer that every alternate vane could have been entirely removed.

13 During the first three days that the dredge *Captain Huston* was in operation and prior to making alterations to the pump the yardage output of the dredge was practically nothing. This was due to the fact that the pump suction throat was continuously clogged with stringy roots and stumps, which it was impossible for the pump to handle. Conditions could not have been worse and it was felt that any method whereby the pump throat and impeller could be opened up would result in an increased output. It is admitted that some uneasiness was felt as to how these alterations might effect the performance of the pumps, but results have more than justified what at first seemed to be a drastic and hazardous measure. Pump and engine tests, typical curves of which are shown in Fig. 8, have shown that these alterations did not effect the capacity or efficiency of the pump, other than by greatly facilitating the passage of roots, etc., through the pump. While the writer understands that radical alterations should not be made without careful consideration, it is undoubtedly true that the performance of a dredge is to be judged largely, if not solely, by the unit cost of output, and where for any reason output is not being secured any modifications of plant are justified, provided that the unit cost of output is controlled. In this particular instance there was involved the difference between failure and success. Assuming the same general character of excavation, a further modification along the lines indicated would it is believed, result in a further increase in output.

14 The impeller designed for the 22-in. dredge *Texas* was of same diameter (80 in.) as that of the old impeller, but it had two instead of four vanes. An examination of Figs. 9 and 10 will show that, due to the peculiar shape of vane, the lodging or collection of roots or other material is made difficult, if not impossible, every encouragement being given by shape of vane and throat of impeller to the continuous flow of water containing irregularly shaped roots or other objects. The writer does not wish to convey the impression,

however, that all of the large cypress stumps and roots were cut up by the cutters and handled through a 20- or 22-in. dredging pump. While suction ladders were built for heavy service and cutters were of good design and powerful, a great many of the stumps were undercut and allowed to sink to the bottom, where they were deposited below grade.

15 In general the operation and maintenance of the dredges

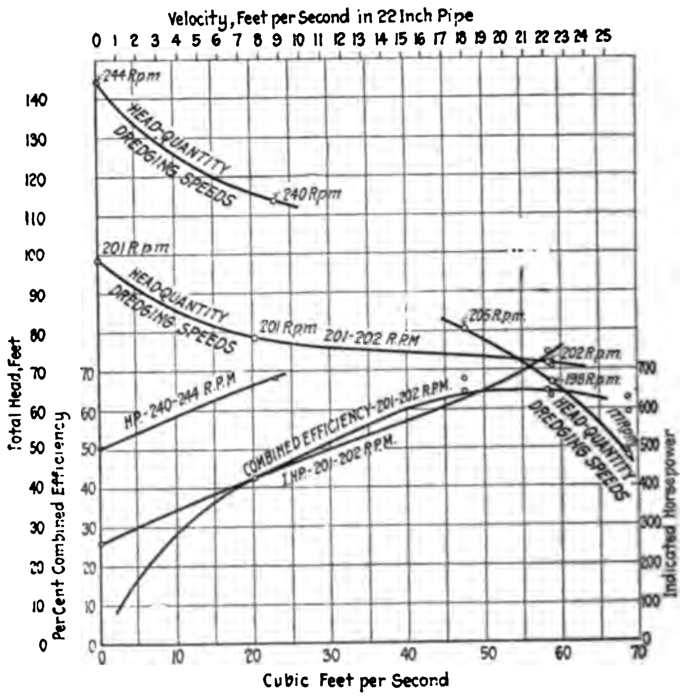


FIG. 8 CURVES OF DREDGE *Texas* EQUIPPED WITH IMPELLER DESIGNED BY A. B. WOOD

followed the usual procedure for dredges of this type, and it is not believed that any novel feature other than in connection with modification of pump design can be presented. The 22-in. dredge *Texas* was equipped to use ball joints as connections between sections of pontoon pipe, and it is believed that the passage of roots was made easier than if rubber sleeves had been used.

16 Pump impellers having two, three, four, six, and seven vanes and varying in diameter from 68 in. to 96 in. were used in the

different dredges engaged on this work. While it is the intention to present methods and actual results secured rather than to discuss the design and performance of centrifugal pumps and impellers, it is thought that some comparative observations made of the different pumps will be interesting. Tests to determine certain centrifugal dredge-pump characteristics while pumping water were carefully made. The results of these tests are given in Tables 1 and 2. Observers were selected and were rehearsed several times to make sure that they understood just what was required of them. All practicable precautions were taken so that accurate results would be secured. Readings were recorded as observed, the object being

FIG. 9 REAR VIEW OF TWO-VANED IMPELLER DESIGNED BY A. B. WOOD

to eliminate any intention to interpret or analyze readings while test was in progress. A study of the readings does not reveal any wide range in any set of readings for a given condition. All the figures in Table 1 are a mean of a number of readings and the writer has confidence in the correctness of all observation and computations, and it is believed that the data presented will be of value in analyzing and studying the design and performance of centrifugal dredging pumps. At the time tests were made, the dredges were in good operating condition, but no special preparation of machinery was made. All of the pumps tested were of the single suction type and all impellers, with one exception, were shrouded on both sides. The pumps were directly connected to vertical triple-expansion engines.

17 Upon the completion of the tests made on the dredges *Captain Huston* and *Dixie* both of which were equipped with pumps having modified propellers it was decided to test the pumps of the dredge *Pelican*, which was fitted with an open impeller. This pump,

Fig. [illegible] View Showing the Large Passageways of the Impeller Designed by A. B. Wood

shown in Fig. 11, was tested with impeller in the pump in the normal manner with the vanes curved backward and also with the impeller reversed, i.e., with the vanes curved in the direction of rotation.

[1] In Transactions, American Society of Civil Engineers, vol. 50, page [illegible], [illegible], Colonel C. W. Sturtevant, Member of The American Society of Civil Engineers, mentions a centrifugal dredging pump having been operated with impeller vanes curved forward instead of backward.

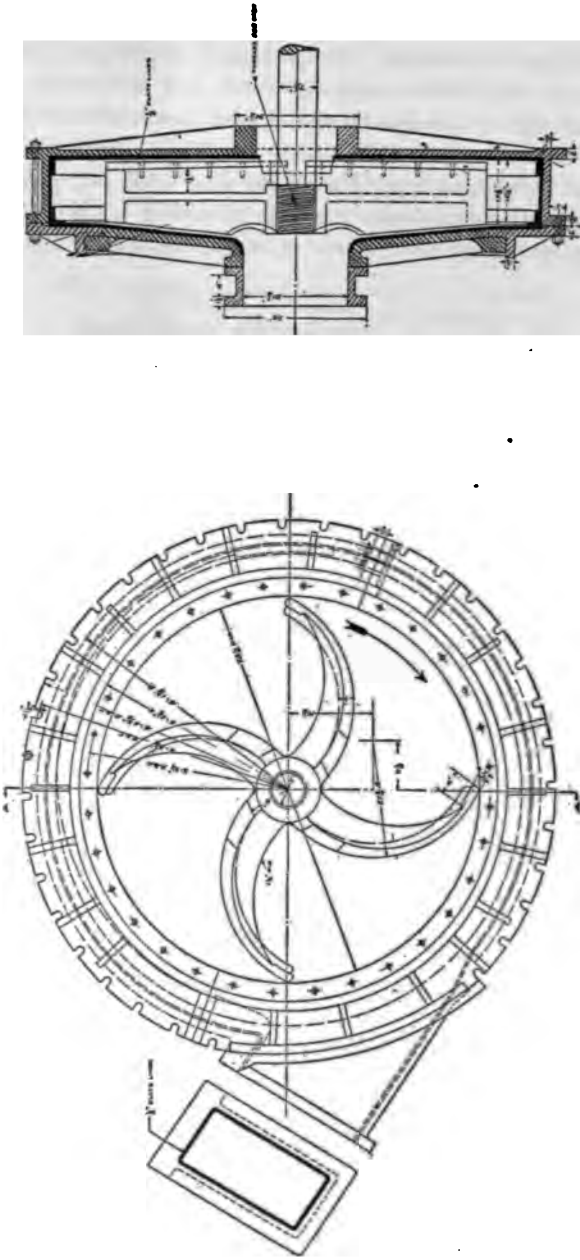


FIG. 11 DETAILS OF THE IMPELLER OF THE DREDGE *Pelican*

The results were surprising and are shown in detail in Table 1 and the curves of Figs. 12 and 13. So far as the operation of the dredge was concerned, both while pumping water and also while dredging there was nothing to indicate to the closest observer that the pump was not assembled in the usual manner. After completion of tests it was found by comparison that there was a 10 per cent loss of efficiency due to increased consumption of power with the reversed impeller. As a matter of interest it might be stated that dredge

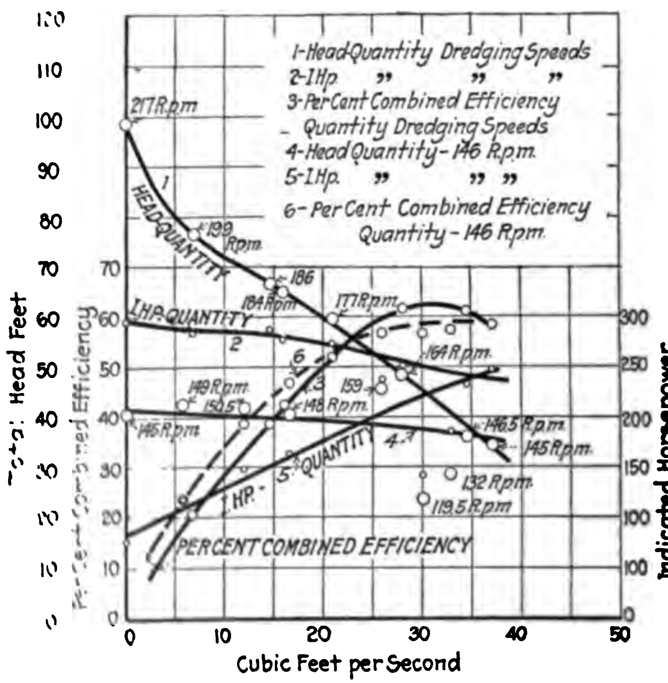


FIG. 12 TEST CURVES OF DREDGE *Pelican*

[illegible] was operated from June 1 to June 14, 1919, with the reversed [illegible] nothing unusual being noticed in operation and no diminu- [illegible] yardage being apparent.

[illegible] In the tests of dredges *Texas* and *Captain Huston* the quan- [illegible] discharged by the pump was measured by pitot tube and [illegible] While testing the dredge *Captain Huston* the pitot [illegible] and was not used in the tests of the other dredges. [illegible] venturi meter was located just astern of the dredge [illegible] section of pipe, where it would not be influenced by elbows. Suitable arrangements were made so that the discharge pipe could be throttled, or closed off entirely, and in this way a variation in head and quantity was created. Piezo-meters were ¼-in. tee-handled pet cocks, which were screwed into the pipe. Indicators were not calibrated, but are believed to have been in excellent condition. Cards were taken simultaneously on the three engine cylinders. It was unfortunate that during the drop in pressure test made on dredge *Dixie* it was impossible to obtain comparable

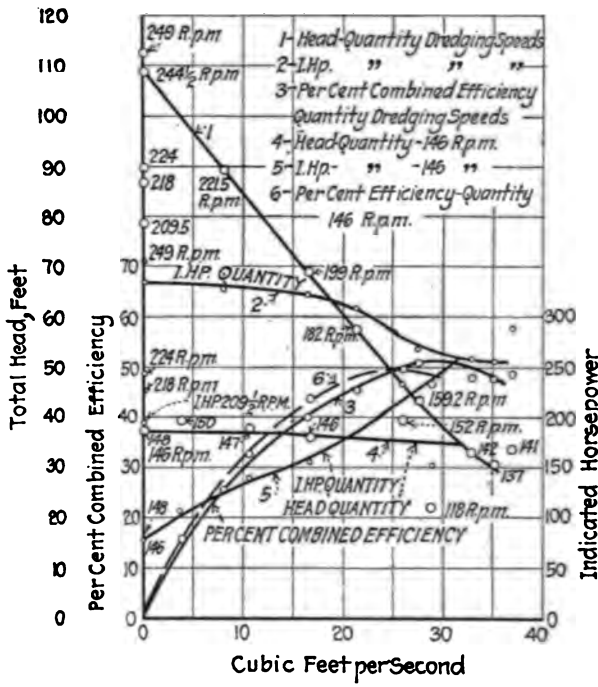


FIG. 13 TEST CURVES OF DREDGE *Pelican* WITH IMPELLER REVERSED

peripheral speeds due to valve of intermediate cylinder being improperly set.

19 From May 15, 1918, to August 1, 1919, there were employed in the construction of the Inner Harbor-Navigation Canal four 20-in. dredges and one 22-in. dredge. One of the 20-in. dredges was owned and operated by the Board of Port Commissioners; one 20-in. dredge was chartered at a monthly rate for the bare dredge, all operating and maintenance costs being borne by the Board; and two 20-in. and one 22-in. dredges were chartered by the

Board at a rate to include all operating and maintenance charges except fuel and shore pipe line, which was borne by the Board. In securing dredges for this work it was necessary to tow them to New Orleans from more or less distant points, two having been brought from Havana, Cuba, the Board assuming all towing charges and paying one-half the charter rate while dredges were in transit. A careful record of costs was kept and the total yardages by months and the total and the unit costs of the same will be found in Table 3. These costs are divided under three headings:

ARBITRARY

- Cost of towing to and from New Orleans
- One-half rental paid while in transit
- Cost of 477,193 cu. yds. of preliminary and auxiliary dredging
- Cost of design and manufacture of two-vane impeller for 22-in. dredge

CHARTER

- Charter cost of four dredges

ALL OTHER CHARGES

- Fuel
- Shore pipe lines
- Labor, subsistence, repairs, material, insurance and supervision.

20 The writer was present and assisted in the very complete test of hydraulic dredges made for the Mississippi River Commission by Colonel F. B. Maltby, Mem. Am. Soc. C. E., and a careful study of the latter's paper in Vol. LIV, Trans. Am. Soc. C. E., has been of material assistance in the preparation of this paper. Colonel Maltby has also very kindly offered suggestions and criticisms during the preparation of this paper.

In a hydraulic pipe-line dredge of the cutter type it is believed that all passages through pipes and pump should be designed so that clogging at any point by more or less irregularly shaped solids will be reduced to a minimum. As a result of tests and also based on observation of the dredges in operation, it is the conclusion of the writer that the modifications described did not decrease the efficiency or capacity of the main engines and pumps, but did greatly increase the output of the dredges as excavating machines.

TABLE 1 DATA AND RESULTS OF TESTS OF CENTRIFUGAL DREDGING PUMPS

Test Number	Steam Pressure lb. sq. in gage	r. p. m.	Suction ft. of water	Discharge ft. of water	Total Including Difference of Velocity Heads	Flow, in ft. per sec. by venturi meter	Indicated hp.	Water hp.	Combined Efficiency Engine and Pump	Peripheral Speed ft. per sec.
				Texas February 14, 1919						
[1]1	175	201	− .94 [a]	76.02	77.03	20.16	423.4	175.8	41.5	70.16
2	175	201	+ .86 [a]	99.34	98.48	0.0	254.45	0.0	0.0	70.16
3	173	202	− 12.94	57.57	71.18	58.37	742.5	471.0	63.3	70.51
4	170	240	− 1.34 [a]	112.04	113.47	22.86	681.9	293.0	43.0	83.78
5	175	244	+ .86 [a]	145.29	144.43	0.0	403.4	0.0	0.0	85.18
				Captain Huston February 23, 1919						
[1]1	200	141	− 30.20	25.82	54.26	52.38	657.9	322.1	49.0	59.06
2	198	161	− 15.23	85.11	99.56	35.17	706.7	397.8	56.3	67.44
3	200	165	− 12.74	93.65	105.69	33.05	729.5	396.0	54.1	69.12
4	200	166	+ .28 [a]	99.89	99.61	0.0	292.7	0.0	0.0	69.54
5	191	168	− 10.06	98.96	108.49	28.87	682.8	355.8	52.1	70.38
6	200	174	− 10.97	104.97	115.33	30.78	742.8	403.0	54.3	72.95
7	200	182.5	− 6.56	123.67	129.88	23.47	756.8	346.0	45.7	76.48
8	200	184	− 6.40	122.22	128.27	23.47	759.6	341.7	45.0	77.00
9	200	221.5	+ .28 [a]	186.94	186.66	0.0	426.0	0.0	0.0	92.78
				Dixie March 11, 1919						
[1]1	158	158	− 25.99	28.52	54.70	49.55	559.55	307.50	55.0	57.91
2	155	160	− 22.98 [b]	34.27	57.42	46.62	568.83	303.71	53.4	58.64
3	150	163	− 7.68	58.36	66.10	27.29	446.78	204.66	45.8	59.74
4	156	163.5	− 18.48 [b]	44.22	62.85	42.0	551.42	299.56	54.3	59.92
5	145	166	− 8.16	60.21	68.44	27.81	471.47	215.94	45.8	60.84
6	152	174.75	− 9.32	66.68	76.08	30.35	545.4	261.8	48.0	64.05
7	155.7	175.5	− 10.74	64.71	75.54	32.33	577.42	277.25	48.0	64.32
8	154.5	176	− 2.68 [a]	76.37	79.08	17.85	479.15	160.15	33.5	64.50
9	156	176	− .03	78.91	78.95	8.95	407.72	80.11	19.65	64.50
10	156	176.5	+ .92 [a]	82.61	81.79	0.0	310.24	0.0	0.0	64.68
11	151	186	− 3.60	83.76	87.40	19.47	547.96	193.06	35.3	68.17
12	150	197	− .38 [a]	99.46	99.85	10.21	552.39	115.66	20.95	72.20
13	154	219	+ .82 [a]	127.17	126.35	0.0	578.31	0.0	0.0	80.26
				Pelican May 28, 1919						
[1]1	174.5	119.5	+ 3.99	8.93	23.16	30.06	139.95	78.9	56.3	38.06
2	175	132	− 17.40	10.58	28.26	32.90	184.33	105.2	57.2	42.04
3	175	145	− 21.96	11.78	34.10	37.04	246.10	143.0	58.1	46.18
4	182.5	145	+ .88 [a]	41.45	40.57	0.0	75.66	0.0	0.0	46.18
5	175	146.5	− 18.96	16.63	35.90	34.58	231.32	140.3	60.7	46.66
6	180	148	− 3.87 [a]	36.38	40.32	16.61	162.82	76.0	46.6	47.14
7	175	149	+ .28 [a]	42.88	42.61	5.74	117.93	27.7	23.5	47.46
8	180	150.5	− 1.57 [a]	40.06	41.67	11.99	147.15	56.6	38.5	47.93
9	182.5	159	− 11.34	34.03	45.55	25.96	236.30	133.8	56.7	50.64
10	182.5	164	− 12.91	35.03	48.14	28.01	249.94	152.8	61.2	52.23
11	182.5	177	− 7.66	51.68	59.45	20.93	272.30	140.8	51.75	56.37
12	177.5	184	− 4.0	60.61	64.69	15.97	276.22	117.0	42.4	58.60
13	175	186	− 2.82 [a]	63.63	66.51	14.65	286.53	110.2	38.5	59.28
14	175	199	+ .08 [a]	76.78	76.71	6.79	283.2	59.0	20.75	63.38
15	177.5	217	+ .88 [a]	99.41	98.53	0.0	295.22	0.0	0.0	69.11

TABLE 1 — Continued

Test Number	Steam pressure lb. sq. in. gage	r. p. m.	Suction ft. of water	Discharge ft. of water	Total Including Difference of Velocity Heads	Flow in ft. per sec. by venturi meter	Indicated hp.	Water hp.	Combined Efficiency Engine and Pump	Peripheral Speed ft. per sec.
				Pelican May 31, 1919 During this test the runner was reversed						
[b]1	175	118	13.49	7.98	21.69	28.85	151.78	70.9	46.75	37.54
2	176	137	19.67	10.53	30.52	35.1	255.12	121.3	47.6	43.63
3	189	141	−21.75	11.28	33.39	36.9	288.57	139.3	48.35	44.91
4	172.5	142	−17.53	15.08	32.89	32.85	256.88	122.3	47.7	45.23
5	177.5	146	− 3.93 [a]	31.88	35.88	16.65	154.56	67.70	43.8	46.50
6	175	146	+ .88 [a]	37.98	37.10	0.0	75.85	0.0	0.0	46.50
7	175.5	147	− 1.13 [a]	36.68	37.84	10.60	138.93	45.5	32.7	46.82
8	172.5	148	+ .88 [a]	38.18	37.30	0.0	79.48	0.0	0.0	47.14
9	172.5	150	+ .64 [a]	40.08	39.44	3.76	106.66	16.78	15.73	47.78
10	181	152	−10.63	28.58	39.38	25.90	233.58	115.3	49.4	48.41
11	181	159	−11.84	31.18	43.22	27.54	266.79	134.7	40.55	50.64
12	177.5	182	− 7.14	50.18	57.44	21.38	307.91	139.0	45.2	57.97
13	187.5	199	− 3.88 [a]	64.88	68.83	16.49	321.50	128.5	40.0	63.38
14	181	209.5	+ .88 [a]	79.68	78.80	0.0	194.58	0.0	0.0	66.73
15	180	218	+ .88 [a]	87.83	86.95	0.0	227.89	0.0	0.0	69.43
16	170	221.5	− .43	88.78	89.23	8.17	327.96	82.50	25.18	70.55
17	185	224	+ .88 [a]	90.58	89.70	0.0	243.34	0.0	0.0	71.34
18	185	244.5	+ .88 [a]	109.78	108.90	0.0	334.57	0.0	0.0	77.87
19	185	249	+ .88 [a]	113.58	112.70	0.0	354.46	0.0	0.0	79.31

[a] Readings of suction gage incorrect because of whirl set up in pipe by pump impeller.

[b] Suction readings incorrect due either to pet cock being throttled down too much or getting temporarily choked up.

Discharge-pipe Line Conditions for Tests of Dredging Pumps Given in Table 1

Texas: Tests 1 and 4, 8-in. valve on end of discharge pipe wide open. Tests 2 and 5, 8-in. valve on end of discharge pipe cut off. Test 3, pipe line 570 ft. pontoon and shore pipe.

Captain Huston: Test 1, Pipe full opening. Test 2, Iron plate bolted on discharge pipe. Tests 3, 7, and 8, 8-in. valve on end of discharge pipe wide open. Tests 4 and 9, 8-in. valve on end of discharge pipe cut off. Tests 5 and 6, Pipe line 3538 ft. pontoon and shore line.

Dixie: Test 1, Pipe full opening. Test 2, Wooden piece bolted on discharge pipe. Tests 3, 5, and 6, Pipe line 1583 ft. pontoon and shore pipe. Test 4, 2 pieces of wood bolted on end of discharge pipe. Test 7, Iron plate bolted on end of discharge pipe. Tests 8 and 11, 8-in. valve on end of discharge pipe wide open. Tests 9 and 12, 8-in. valve on end of discharge pipe partly closed. Tests 10 and 13, 8-in. valve on end of discharge pipe cut off.

Pelican: Tests 1, 2, and 3, Pipe full opening. Tests 4 and 15, 8-in. valve on end of discharge pipe cut off. Tests 5 and 10, Wooden piece bolted on end of discharge pipe. Tests 6 and 11, 3 wooden pieces bolted on end of discharge pipe. Tests 7 and 14, 8-in. valve on end of discharge pipe partly closed. Tests 8, 12 and 13, 8-in. valve on end of discharge pipe wide open. Test 9, 2 wooden pieces bolted on end of discharge pipe.

Pelican (with reversed runner): Tests 1, 2, and 3, Pipe full opening. Test 4, Wooden piece bolted on end of discharge pipe. Tests 5 and 12, 3 wooden pieces bolted on end of discharge pipe. Tests 6, 8, 14, 15, 17, 18 and 19 8-in. valve on end of discharge pipe cut off. Tests 7 and 8, 8-in. valve on end of discharge pipe wide open. Tests 9 and 16, 8-in. valve on end of discharge pipe partly closed. Tests 10 and 11, 2 wooden pieces bolted on end of discharge pipe.

In all cases where length of discharge pipe is not given the length was less than 100 ft. from the stern of dredge. In general, it should be stated that the figures in Table 1 are a mean of a number of readings recorded during tests, which explains why the figures are worked out to two decimal places. The venturi meter had an inlet diameter of 22$\frac{5}{16}$ in. and a throat diameter of 18 in. For readings at cut off, the elevation of water at rest in suction pipe was taken as suction head and for other readings of low quantity the suction head was taken from loss pressure curve plotted from high quantity readings.

TABLE 2 DATA ON CENTRIFUGAL DREDGING PUMPS

	Texas	Capt. Huston	Dixie	Pelican	Pelican †
Diam. of impeller, in	80	96	84	73	73
Number of vanes	2	3 *	3 *	4	4
Diam. of suction pipe at suction gage, in.....	23$\frac{3}{4}$	19$\frac{31}{32}$	20$\frac{5}{16}$	20$\frac{3}{4}$	20$\frac{3}{4}$
Diam. of discharge pipe at discharge gage, in	22$\frac{11}{16}$	20$\frac{3}{32}$	20$\frac{7}{16}$	25$\frac{1}{2}$ x 12$\frac{11}{16}$	25$\frac{1}{2}$ x 12$\frac{11}{16}$
Triple expans. engine, in..	14 x 21$\frac{1}{2}$ x 35	14$\frac{3}{4}$ x 22$\frac{1}{2}$ x 40	14$\frac{3}{4}$ x 22$\frac{1}{2}$ x 36	11$\frac{1}{16}$ x 18 x 29	11$\frac{1}{16}$ x 18 x 29
Length of stroke, in......	18	20	18	18	18
Water at rest in canal above center-line of pump, in..	10$\frac{3}{4}$	3$\frac{1}{4}$	9$\frac{7}{8}$	....	....
Water in river above center-line of pump, in.....	....	....	....	10$\frac{1}{2}$	10$\frac{1}{2}$
Zero of suction gage below center-line of pump, in..	19	20	16.68	1.68	At center line of pump
Zero of discharge gage above center-line of pump, in..............	20	....	....	7	7
Zero of discharge gage below center-line of pump, in..................	....	53$\frac{13}{16}$	47.66	....	....

* Originally 6, parts of 3 vanes removed. † This test made with impeller reversed.

TABLE 3 DREDGING YARDAGES AND COSTS

Year 1918–1919	Yardage Excavated	COSTS							
		Arbitrary	Unit Cost	Charter	Unit Cost	All Other Charges	Unit Cost	Total	Unit Cost
May, June and July	383,613			74,887.19	.196	12,143.72	.031	87,030.89	.227
August	291,256	9,521.07	.015	90,547.20	.245	11,053.66	.034	111,124.93	.294
September	578,065	18,902.72	.022	63,993.60	.183	44,602.56	.054	127,498.88	.259
October	734,056	24,003.63	.027	62,486.40	.147	53,717.97	.061	140,208.00	.235
November	427,522	13,979.97	.028	54,566.40	.143	77,431.04	.082	145,977.41	.253
December	597,488	19,534.82	.028	66,628.80	.138	61,053.99	.086	147,217.61	.252
January	498,340	15,295.72	.029	64,145.76	.136	80,726.85	.097	160,168.33	.262
February	455,159	14,883.70	.029	57,874.80	.136	45,607.02	.096	118,365.52	.261
March	222,508	7,266.01	.029	24,807.55	.133	35,857.67	.101	67,931.23	.263
April	167,481	5,476.63	.029	19,027.80	.133	22,203.91	.102	46,708.34	.264
May	94,777	3,099.20	.030	12,724.80	.133	6,028.60	.101	21,852.60	.264
June	130,386	4,263.62	.030	13,915.20	.132	8,376.64	.100	26,555.46	.262
July	141,233	4,618.32	.030	14,275.20	.131	3,528.24	.098	22,421.76	.259
Preliminary and Auxiliary	477,193								
Total	**5,199,107**	**140,848.41**	**.027**	**619,880.68**	**.119**	**462,331.87**	**.089**	**1,223,060.96**	**.235**

All Yardages Computed from Sections Plotted from Soundings. Above Total Yardage is 70% of Estimated Yardage for Entire Project. Unit Cost Apply for Total Yardage Through Month Shown.

No. 1717

TURBO-COMPRESSOR CALCULATIONS

By Allen H. Blaisdell, Pittsburgh, Pa.
Associate-Member of the Society

After pointing out the features which differentiate the turbo-compressor from the multi-stage centrifugal pump, the author takes up the study of the pressure and volume changes occurring while a fluid is passing through one stage of the former and states the fundamental energy equations involved. He then discusses the number of stages to be used in a given case and follows this with an explanation of the method of laying out a compression diagram and of subdividing it according to the number of stages to be provided for.

The function of the impeller is next discussed and in this connection expressions are derived for the actual and effective impeller radii and the axial depth of channel of any radius. Blades and blade angles are also considered and a graphical method is described for calculating the theoretical head to be developed by an impeller.

The paper concludes with a brief study of the working of free-vortex and guide-vane diffusers, and of matters to be taken into account in designing in order to obtain a turbo-compressor of the highest possible overall efficiency.

IT is well known that the turbo-compressor is quite similar in its nature to the multi-stage centrifugal pump. Whatever differences exist between the two types of machines is due to the fact that one handles a gaseous medium while the other handles a practically incompressible fluid, water. Both are high-speed machines.

2 The impellers of multi-stage centrifugal pumps are of the same size throughout but those of turbo-compressors should be theoretically of continually diminishing dimensions because of the decrease in volume of the fluid as it is compressed from stage to stage. Actually the impellers are divided into two or more groups and each group calculated for its own average conditions. Since the capacities of these machines rarely exceed 50,000 cu. ft. per min. and in general range from 7000 to 10,000, the single-flow type of impeller is used in most turbo-compressors.

3 The only other important feature which differentiates the turbo-compressor from the centrifugal pump is the arrangement made for cooling the fluid as it passes through the compressor. In

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

the reciprocating compressor practically all cooling is accomplished by means of intercoolers located between the different stages, the water jackets of the cylinders having but slight effect on the temperature rise of the fluid which is being compressed in the cylinders. In turbo-compressors the cooling is accomplished in two ways. (*a*) by means of cooling coils between each stage and (*b*) by means of carefully designed water jackets. The water-jacket method is much more effective in the case of the turbo-compressor than in that of the reciprocating machine because the fluid being compressed has a more intimate contact with the cooling surfaces.

4 The discussion will be divided into three main divisions:

A Thermodynamic Considerations
- *a* Study of the nature of the compression
- *b* Fundamental energy equations
- *c* Layout of compression diagram

B Pneumatic Considerations
- *a* Theory of fluid flow
- *b* Essentials of impeller design
- *c* Essentials of diffuser design

C Power Consumption and Efficiency.

(*A*) THERMODYNAMIC CONSIDERATIONS

NATURE OF THE COMPRESSION

5 Fig. 1 shows in section a portion of two stages of a multistage centrifugal or turbo-compressor. The fluid being compressed enters the impeller at *a* with a definite pressure, velocity and temperature. The impeller, rotating with a high speed, discharges the fluid with an increased pressure, temperature and velocity. Leaving the impeller the fluid enters the diffuser where its velocity is greatly reduced and its pressure increased. The temperature will also be reduced somewhat, due to the cooling water which passes through the diffuser vanes. At *d* the fluid leaves the diffuser and passes down between guide vanes, during which interval it is in contact with water-cooled surfaces *f*, and suffers a still further reduction of temperature and volume. At *e* the fluid enters the succeeding impeller where it goes through a similar cycle.

6 The change of state of the fluid being compressed will follow the law pv^n = constant. The value of the exponent n is governed by the kind of fluid being compressed, the moisture condition of the fluid and by the surrounding thermal conditions. When the

fluid being compressed neither receives heat from, nor gives up heat to, surrounding bodies the compression is called adiabatic, and under these conditions $n = 1.41$ for air. When the nature of the compression is such that the temperature remains constant, then the compression is said to be isothermal and $n = 1$ for all gases.

7 It is very difficult to say exactly what pressure and volume changes occur while a fluid is passing through one stage of a turbo-compressor. The energy transfer to the fluid occurs in the impeller, which changes the angular momentum of each fluid particle, work being done upon the fluid as a whole. As a result of this energy

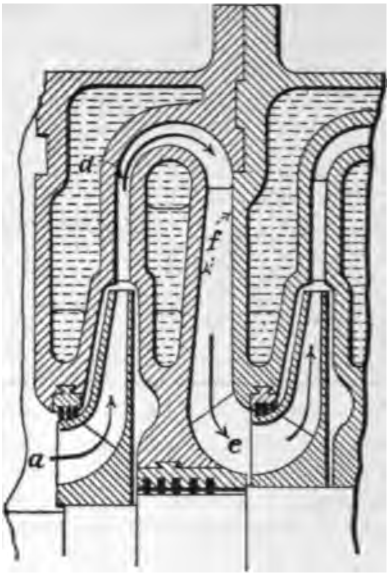


Fig. 1 Section of Portion of Two Stages of a Multi-Stage Centrifugal or Turbo-compressor

transfer the fluid on exit from the impeller possesses an increased pressure and velocity. But this pressure is not the total pressure developed in the stage, and herein the action in a turbo-compressor differs somewhat from the reciprocating class of machines. In the reciprocating compressor work is done by the piston upon the fluid enclosed in the compressor cylinder, first compressing the gas to the discharge pressure and then ejecting the fluid from the cylinder. In the case of the turbo-machine the impeller does work upon the fluid which results in an increase not only of pressure but also of velocity, and if the amount of work is the same in both instances it is quite evident that the pressure rise in the impeller will not be so great as in the case of the reciprocating compressor, since a portion of the energy imparted to the fluid has the kinetic form. Consequently, in order to secure as large a pressure rise as possible, use is made of a diffuser wherein no further work is done on the fluid, but a portion of its velocity head is changed over into pressure head.

8 It is possible to study the pressure and volume changes in the impeller of a turbo-compressor by means of a *pv*-diagram just as in the case of the reciprocating machines. Thus, in Fig. 2, *ABCD* may be considered as representing what occurs in an impeller. The line *BC* will be the actual compression curve and must be located outside of the adiabatic *BF*. This is quite easy to understand since there is likely to be but very little heat transfer through the

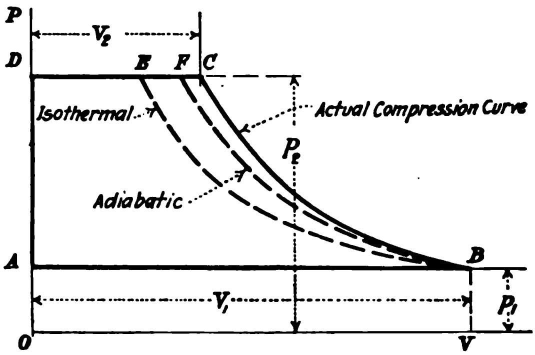


Fig. 2 Diagram Showing Pressure and Volume Changes in the Impeller of a Single-stage Turbo-compressor

walls of the impeller to or from the fluid because of the short interval of time which elapses while the fluid is passing through the impeller. Because of friction, impact, eddying and turbulent flow, extra work is done and this results in a compression which follows the polytropic curve *BC* rather than the adiabatic *BF*. This means that the value of the exponent n in pv^n = constant will be greater than 1.41. The temperature rise of the fluid must consequently be in excess of that due to adiabatic compression alone and hence it is very essential that every effort be made to cool the fluid after it leaves the impeller, since by so doing the temperature and volume will be reduced before entrance into the succeeding stage and consequently the compression work of that stage will also be diminished.

9 Of the total energy imparted to the fluid in the impeller

only a portion is in the form of pressure, the remainder being used up in increasing the velocity. Such being the case the expression for total work must contain a term to include the increase of kinetic energy as well as a term to cover the work of compression. The equation for total work in the impeller in ft-lb. can be stated then as follows:

$$W = \frac{144\ p_1 v_1}{\eta_p}\left(\frac{n}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{w_2^2 - w_1^2}{2g} \times \frac{1}{BT_2}\right) \quad . \quad [1]$$

where

W = work done in ft-lb. per lb. of fluid
p_1 = initial pressure in lb. per sq. in. abs.
p_2 = discharge pressure in lb. per sq. in. abs.
v_1 = initial volume in cu. ft. per lb.
n = compression exponent = 1.41 for air
B = 53.34 for air
T_2 = temperature of fluid on discharge from impeller
η_p = pneumatic efficiency
w_2 and w_1 = relative velocities of discharge from, and entrance into, impeller, respectively. In the first stage neglect w_1.

10 If we assume the pneumatic efficiency to be 0.72 and that $n = 1.41$, the work equation, in ft-lb., becomes

$$W = 200\ p_1 v_1\left(3.5\left[\left(\frac{p_2}{p_1}\right)^{0.286} - 1\right] + \frac{w_2^2 - w_1^2}{2gBT_2}\right) \quad . \ . \ . \quad [2]$$

The pneumatic efficiency corresponds to the hydraulic efficiency of centrifugal pumps. It is the ratio of the power theoretically required to pump the air against a given pressure to that actually required. Its value usually ranges between 0.7 and 0.8. The factors which affect the pneumatic efficiency will be discussed later on in the paper.

11 In the diffuser the kinetic energy of the fluid is partially converted into the form of potential energy or pressure, and by considering the diffuser as merely a gradual enlargement of the fluid path it is easy to show that the pressure rise in the diffuser, in lb. per sq. in., is given by

$$p_3 - p_2 = \frac{\delta_m}{144} \cdot \frac{v_R^2}{2g}\left[1 - \left(\frac{D_i}{D_o}\right)^2\right] \quad . \ . \ . \ . \ . \ . \quad [3]$$

where

p_3, p_2 = static pressure in lb. per sq. in. abs. at discharge from and entrance into diffuser, respectively
δ_m = mean fluid density for the stage
v^R = absolute velocity of discharge from the impeller

D_o, D_i = outside and inside diameters of the diffuser, respectively. In general the pressure rise in the diffuser will vary from 0.5 to 0.7 of that in the impeller. The total pressure rise for the stage, in lb. per sq. in. abs. is then

$$P = p_3 - p_1 = \frac{\delta_m}{9274} v_R^2 \left[1 - \left(\frac{D_i}{D_o}\right)^2\right] + (p_2 - p_1) \quad . \quad . \quad . \quad . [\mathbf{4}]$$

where

$$p_2 - p_1 = \frac{\delta_m}{9274}[(w_2^2 - w_1^2) + (u_2^2 - u_1^2)]$$

and

w_2, w_1 = relative velocities of discharge from and entrance into impeller, respectively

u_2, u_1 = peripheral velocities (Fig. 7).

NUMBER OF STAGES

12 The number of stages to be used depends upon the total pressure to be developed by the machine and the speed of the prime mover. Since the fluid being compressed is a gas and the terminal pressure for a single stage is proportional to the density of the gas, it is evident that a large number of stages will be required if very high delivery pressures are desired.

13 In selecting the proper number of stages a compromise has to be made between a few stages with large-diameter impellers and a large number with impellers of small diameter. If large impellers are used the frictional losses due to the fluid drag of the fluid on the impeller will be excessive, reducing the overall efficiency of the compressor. On the other hand, if small impellers are used the pressure rise per stage will not be so large and this means an increase in the number of stages with a consequent increase in the cost of production, but the efficiency will be better than that of the first machine. In addition to reduced frictional losses there will also be a decrease in the compression work, and this factor helps to raise the efficiency above that of the machine with a small number of stages.

14 For equal amounts of compression work per stage, the pressure developed will not be equal but will increase from stage to stage. If the stages are divided into two or more sets, all impellers of a single group having the same diameter, then the same fact applies, but in this case the pressure rise will be the average for the group.

LAYOUT OF COMPRESSION DIAGRAM FOR TURBO-COMPRESSOR

15 For illustrative purposes only, let us suppose that the diagram of Fig. 3 is that of a four-stage compressor. It is desired to study the peculiarities of the diagram and develop a graphical method of constructing such a diagram. The method to be explained was originated by M. de Stein.

16 The total energy transfer from all the impellers to the fluid passing through them is represented in the diagram by area $ABCDEFGHIK$. In other words, area $ABCD'$ is an equivalent pv-diagram for the first stage and represents not only the work of

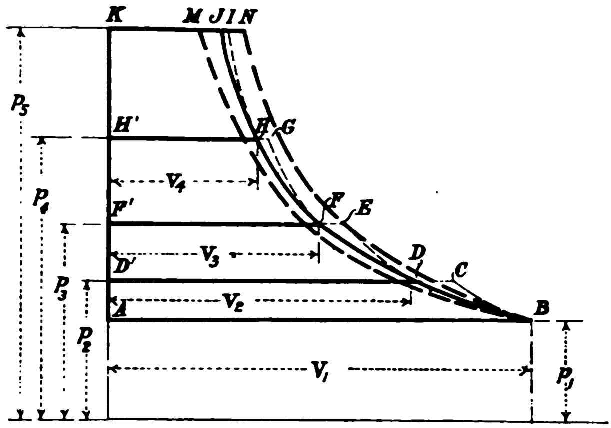


FIG. 3 COMPRESSION DIAGRAM FOR A FOUR-STAGE COMPRESSOR

compression in the impeller but also the increase of kinetic energy. In the same way does $D'DEF'$ represent the energy transfer for the second stage. In other words, if p_1 is the inlet pressure to the first stage, then p_2 is the delivery pressure from the diffuser, and not the impeller. Consequently, the total pressure rise for the stage (impeller and diffuser) is $p_2 - p_1$. Such being the case, p_5 will be the delivery pressure of the machine if the kinetic energy in the fluid when discharged is neglected. The pressures p_2, p_3 and p_4 may be considered as the static pressures which exist between the diffuser of one stage and the entrance to the impeller of the following stage.

17 Good design necessitates that the areas $ABCD'$, $DEF'D'$, etc., shall be equal or that the energy delivered to the fluid shall be the same for each stage. The lines BC, DE, FG are laid out as adiabatics. If the cooling is effective they will all lie in closer

proximity to the isothermal BM than the adiabatic BN. When there are a large number of stages, say eight or ten, it is allowable to assume the curve BJ (passing through points D, F, H) as representing the equivalent compression curve.

18 To lay out the pv-diagram $ABJK$ for the machine requires that we shall know the initial pressure p_1, the initial volume v_1, in

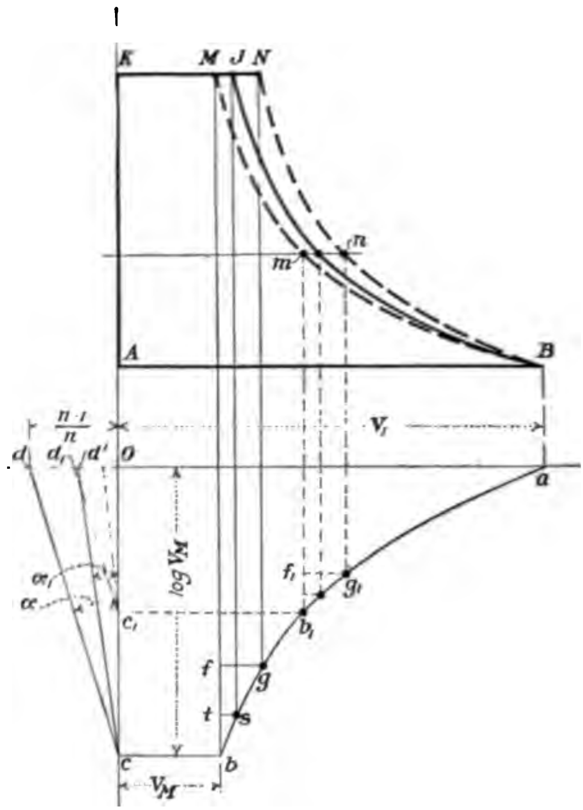


FIG. 1 METHOD OF LAYING OUT COMPRESSION DIAGRAM FOR TURBO-COMPRESSOR

cu. ft. per lb., the delivery pressure p_5 and the value of n in pv^n = constant in order that the curve BJ can be drawn.

19 The first step in the procedure will be to lay out the isothermal BM and the adiabatic BN. Graphical methods for drawing an isothermal curve are given in most handbooks and hence the drawing of this curve need not be explained. The adiabatic

can be drawn as follows: If v_1 is the initial volume in cu. ft. per lb., and v_N is the final volume for adiabatic compression (Fig. 3), then

$$v_N = (v_1)^{\frac{n-1}{n}}$$

if the supply volume v_1 for isothermal compression = unity.

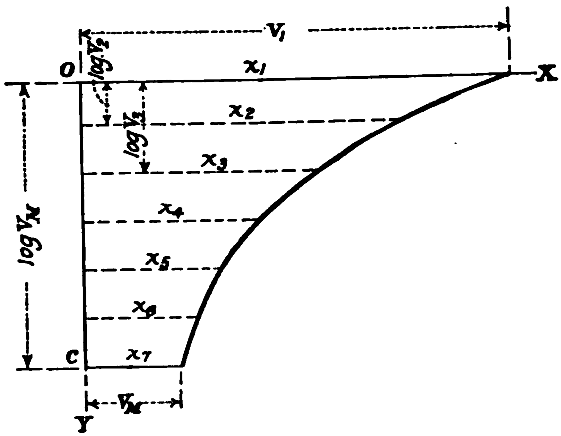


FIG. 5 CONSTRUCTION OF LOGARITHMIC CURVE EMPLOYED IN FIG. 4

Therefore,

$$\log v_N = \frac{n-1}{n} \log v_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

This expression when solved will give us the value of v_N for $v_M = 1$. Note that for $v_1 = 1$,

$$v_M = \frac{p_1}{p_M}$$

20 The location of the point N and the layout of the adiabatic curve BN can, however, be accomplished quite easily in the following manner. In Fig. 4 ab is a logarithmic curve constructed as indicated in Fig. 5. Along the vertical OY (Fig. 5), let any length Oc represent log v_M to scale. Divide Oc into any number of equal parts, say, six, and draw through these division points the lines x_2, x_3, x_4, etc., making the length of $OX = x_1 = 1$ and $x_6 = \frac{?}{?} x_7$, $x_5 = \frac{?}{?} x_6$, etc. Then by drawing a smooth curve through the extremities of these lines we have the desired curve. The vertical distance from any of the lines x_2, x_3, x_4, etc., to OX will represent the logarithms of the corresponding volumes to the same scale as Oc represents the logarithm of v_M.

21 Equation [5] can be changed to

$$\log v_N = \frac{\log v_M}{\dfrac{n-1}{n}} \quad \ldots\ldots\ldots\ldots \quad [6]$$

The division of $\log v_N$ by $\left(\dfrac{n-1}{n}\right)$ can be carried out graphically as follows: Through point M of the isothermal drop a vertical line intersecting the logarithmic curve ab at b. Through b draw the horizontal bc. Draw the line cd at the angle α with the vertical. The value of angle α can be determined from

$$\tan \alpha = \frac{1}{\log v_N} = \frac{\dfrac{n-1}{n}}{\log v_M} \quad \ldots\ldots\ldots \quad [7]$$

On bM lay off bf equal in length to Od. Through f draw the horizontal fg. If a vertical line is now drawn through g and extended to intersect the horizontal through K at N, then N will be a point on the adiabatic curve. To locate any other point n on BN which shall lie on a horizontal through m on the isothermal BM it is only necessary to repeat the procedure used in locating point N. Thus through any point m drop a vertical intersecting the curve ab at b_1. Next draw the horizontal b_1c_1. Lay off line c_1d_1 parallel to cd. On b_1m mark off b_1f_1 equal to Od_1. Now draw the horizontal f_1g_1 and the vertical through g_1 intersecting a horizontal through m at point n, the point desired.

22 Having laid out the isothermal and adiabatic curves it is now necessary to lay out the compression curve for the machine. In order to do this it is practically necessary to assume the position of point J of the compression curve such that

$$v_J \, \eta_a \, (v_M - v_N) + v_N \quad \ldots\ldots\ldots\ldots \quad [8]$$

where η_a = adiabatic efficiency.

23 When the compression is adiabatic, if the final temperature is T_N deg. abs., the adiabatic efficiency, or ratio between the theoretical internal work and the work actually furnished for the compression, can be expressed by

$$\eta_a = \frac{T_N - T_1}{T_J - T_1} \quad \ldots\ldots\ldots\ldots \quad [9]$$

where

T_N = final temperature in deg. abs. with adiabatic compression

T_J = final temperature in deg. abs. with actual compression

T_1 = initial temperature in deg. abs.

24 If frictional resistances are considered, then the value of η_a as calculated from [9] will be increased about 3 per cent.

25 Having located point J, project downward from J a line intersecting the log curve ab at S. Through S draw a horizontal intersecting bM at t. Then if the distance Od' is laid off on Od equal to bt, the line joining d' with point c determines the new value of $\alpha = \alpha'$. It is now only a matter of repeating the process used in laying out the adiabatic curve BN in order to locate the necessary points on the actual compression curve BJ.

26 Knowing the value of α' it is possible to calculate the value of n', the actual compression exponent. Thus from

$$\tan \alpha' = \frac{Od'}{Oc} = \frac{\frac{n'-1}{n'}}{\log v_M}$$

we get

$$n' = \frac{1}{1 - \tan \alpha' \log v_M} \quad \ldots\ldots\ldots \quad [10]$$

(B) PNEUMATIC CONSIDERATIONS

DIVISION OF THE PRESSURE–VOLUME DIAGRAM

27 On completion of the actual compression diagram $ABJK$ (Fig. 6) it is next necessary to divide its area into the same number of parts as there will be stages or groups of stages in the compressor. In this instance it is four.

28 The total energy head developed by a single impeller or group of impellers can be stated in the form

$$H = \frac{p}{\delta_m} = K \frac{u^2}{g} \quad \ldots\ldots\ldots\ldots \quad [11]$$

where

$K = \eta_p \times \theta$

and

δ_m = mean density for a single stage, or group of stages

u = peripheral velocity of the impeller (or impellers) in ft. per sec.

η_p = pneumatic efficiency for a single stage, or group of stages

θ = form factor determined by the blade shape, angles, etc.

p = pressure rise for a single stage, or group of stages.

29 It is evident from Equation [11] that the compression diagram must be so divided into areas that the law pv = constant shall hold true; that is, all the areas into which the pv-diagram is divided must be equal.

30 The following graphical method can be used with great facility in carrying out this subdivision, particularly when there are a good many stages to be considered.

31 Let $ABJK$ (Fig. 6) represent the compression diagram for the compressor. We will assume, as before, that the machine has four stages only and proceed to divide this diagram into parts such that the areas will all be equal.

32 Divide the distance KA into any convenient number of equal lengths, say, six, marked by the figures 1, 2, 3, etc. Through these points draw horizontal lines 1–1, 2–2, 3–3, etc., cutting the diagram up into six trapezoidal areas which can be quite easily converted into equivalent rectangles as indicated in Fig. 6.

33 The areas of these rectangles are to each other as their bases. Reduce these bases by one-half, say, and add to each one of these the sum of the reduced lengths of the preceding ones.

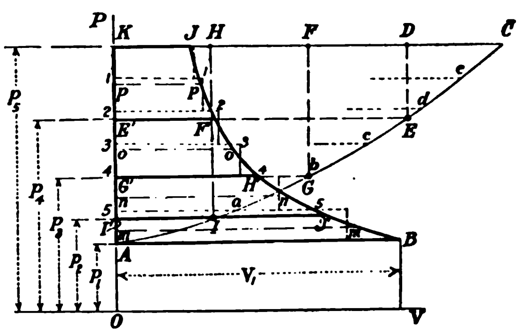


Fig. 6 Pressure-Volume Diagram for Compressor

Through the ends of the lines thus drawn pass a smooth curve, $AabcdeC$. The length of any horizontal such as KC represents to scale the area between it and the lines JB, BA, and AK. In other words, because of the construction the length KC represents the area of the compression diagram. Such being the case, if we divide KC into the same number of equal parts as there will be stages, or groups of stages (in this instance, four) and through the division points H, F, D, drop verticals intersecting the curve AC at I, G, and E, respectively, then horizontals through I, G, and E will divide the compression diagram into four sections whose areas will be equal. The lengths AI', $I'G'$, $G'E'$, etc., will represent the total pressure head to be developed by each stage, or group of stages.

34 If the line AB represents the total volume of fluid taken into the compressor, then mm, nn, oo, pp, etc., will represent, to the same

scale, the average volume for the various stages, or groups of stages. On the other hand, if AB represents the specific volume of the fluid drawn into the first stage, then mm, nn, oo, pp, etc., will represent to the same scale the average specific volumes of the different stages.

35 If Vm_1, Vm_2, Vm_3, etc., are the mean specific volumes just defined, Vm_1 being the mean specific volume for the first stage, then the mean densities are

$$\delta_1 = \frac{1}{Vm_1}; \qquad \delta_2 = \frac{1}{Vm_2}; \qquad \delta_3 = \frac{1}{Vm_3}; \text{ etc. } \quad . \; . \; . \; . \quad [12]$$

36 It has already been stated that the pressures developed per stage are proportional to the fluid densities. Hence the ratio of the total pressure developed in one stage to that of the following or preceding stage must be the same as the ratio of the mean densities. If p_2, p_3, p_4, etc., are the discharge pressures per stage, we have

$$\frac{p_2}{p_1} = \frac{\delta_2}{\delta_1}; \qquad \frac{p_3}{p_2} = \frac{\delta_3}{\delta_2}; \qquad \frac{p_4}{p_3} = \frac{\delta_4}{\delta_3}; \text{ etc. } \quad . \; . \; . \; . \quad [13]$$

where p_1 and δ_1 are the pressure and density of the fluid on entrance into the first stage.

37 If a group of stages is made up of n impellers and p is the total pressure increase for the group, we can assume the average increase per impeller and diffuser (or stage) as given by

$$\Delta p = \frac{p}{n} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [14]$$

38 In reality the pressure increases will not be the same, as we already know, but the resulting error has no effect on the accuracy of the calculations as a whole. By using Equation [14] we virtually assume that the fluid is incompressible for a group of stages, or that the mean density of one stage is the same as those of the remaining stages of the group.

39 It is possible to derive a general equation for the increase of pressure for the successive stages for any polytropic compression. For instance, let us consider isothermal compression; the compression must follow the law of areas and the law which is determined by the nature of the compression. From the law of areas we have that all the sections into which the pv-diagram is divided must be equal; that is,

$$(p_{n+1} - p_n) \times V_n = \text{constant} = C \quad . \; . \; . \; . \; . \quad [15]$$

From the nature of the compression we have that the product of

the mean pressure times the mean volume for any stage, or group of stages, must be constant. Hence for isothermal compression

$$\left(\frac{p_{n-1}+p_n}{2}\right) \times V_n = C' = \text{constant} \quad . \quad . \quad . \quad . \quad . \quad [16]$$

From [15] we get

$$V_n = \frac{C}{p_{n-1}-p_n}$$

and substituting this value of V_n in [16], we have

$$\frac{p_{n-1}+p_n}{2} \times \frac{C}{p_{n-1}-p_n} = C'$$

or

$$\frac{p_{n-1}+p_n}{p_{n-1}-p_n} = 2\frac{C'}{C} = a$$

from which we find that

$$p_{n-1} = p_n\left[\frac{a+1}{a-1}\right] = p_n k \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [17]$$

where

$$k = \frac{a+1}{a-1}.$$

40 As a general law for increase of pressure per stage, or group of stages, we then have the pressure increase represented by the terms of a geometric progression of which the argument is k.

IMPELLERS AND DIFFUSERS

41 In the design of impellers and diffusers for turbo-compressors we are forced to depend on the theory of centrifugal pumps, which at its best is far from being satisfactory or complete for centrifugal pumps even, and of course is far less so when applied to turbo-compressor calculations. However, by making certain assumptions it is possible to utilize the theory with quite satisfactory results.

42 In order to simplify the analysis of fluid flow it is necessary to make the following assumptions:

a That the fluid has stream-line motion, i.e., the fluid particles move in non-intersecting curves

b That the fluid will, since it is elastic, completely fill the channel of flow

c That the motion of the fluid is steady, i.e., each fluid particle has the same mean velocity as every other particle

which is moving along with it through any section and the static pressure is the same at all points of a section which is at right angles to the fluid flow

d That the fluid is non-compressible.

THE IMPELLER

43 The function of the impeller is to impart a certain amount of energy to the fluid passing through it, with a maximum of efficiency. That is, friction, fluid impact, and turbulence should be reduced to a minimum.

44 On the assumption that the speed of rotation and the rate of fluid flow are known, there will be derived expressions for the actual and effective impeller radii, and the axial depth of the impeller channel at any radius. In addition to these there will also be given a brief discussion on blades and blade angles, together with the explanation of a graphical method for calculating the theoretical head to be developed by an impeller.

45 If we assume the impeller discharge to be zero, then from Equation [1] we obtain

$$W = \frac{144\ p_1 v_1}{\eta_i} \cdot \frac{n}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \quad . \ . \ . \ . \ [18]$$

where η_i = impeller efficiency, being practically identical with η_R, but applying to the conditions of zero discharge only. This expression actually represents the work of compression performed in the impeller against the centripetal forces of the fluid particles. Since these forces vary from a minimum at the impeller entrance to a maximum at the discharge, the work done in forcing one pound of fluid from entrance to exit is equal to the average value of the centripetal force multiplied by the radial distance through which the pound of fluid is to move (with a negligible velocity). Such being the case, we have

$$F \times (R_2 - R_1) = W \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ [19]$$

where

F = centripetal force $= \dfrac{N^2}{5908}\ (R_2 + R_1)$

R_2, R_1 = actual outer and inner radii of the impeller in feet

N = r.p.m. of impeller

W = work of compressing 1 lb. of fluid in the impeller in ft-lb.

the mean pressure times the mean volume for any stage, or group of stages must be constant. Hence for isothermal compression

$$\left(\frac{p_{n-1} + p_n}{2}\right) \times V_n = C' = \text{constant} \quad . \quad . \quad . \quad . \quad . \quad [16]$$

From [15] we get

$$V_n = \frac{C}{p_{n-1} - p_n}$$

and substituting this value of V_n in [16], we have

$$\frac{p_{n-1} + p_n}{2} \times \frac{C}{p_{n-1} - p_n} = C'$$

or

$$\frac{p_{n-1} + p_n}{p_{n-1} - p_n} = 2\frac{C'}{C} = a$$

from which we find that

$$p_{n+1} = p_n\left[\frac{a+1}{a-1}\right] = p_n k \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [17]$$

where

$$k = \frac{a+1}{a-1}.$$

40 As a general law for increase of pressure per stage, or group of stages, we then have the pressure increase represented by the terms of a geometric progression of which the argument is k.

IMPELLERS AND DIFFUSERS

41 In the design of impellers and diffusers for turbo-compressors we are forced to depend on the theory of centrifugal pumps, which at its best is far from being satisfactory or complete for centrifugal pumps even, and of course is far less so when applied to turbo-compressor calculations. However, by making certain assumptions it is possible to utilize the theory with quite satisfactory results.

42 In order to simplify the analysis of fluid flow it is necessary to make the following assumptions:

a That the fluid has stream-line motion, i.e., the fluid particles move in non-intersecting curves

b That the fluid will, since it is elastic, completely fill the channel of flow

c That the motion of the fluid is steady, i.e., each fluid particle has the same mean velocity as every other particle

which is moving along with it through any section and the static pressure is the same at all points of a section which is at right angles to the fluid flow

d That the fluid is non-compressible.

THE IMPELLER

43 The function of the impeller is to impart a certain amount of energy to the fluid passing through it, with a maximum of efficiency. That is, friction, fluid impact, and turbulence should be reduced to a minimum.

44 On the assumption that the speed of rotation and the rate of fluid flow are known, there will be derived expressions for the actual and effective impeller radii, and the axial depth of the impeller channel at any radius. In addition to these there will also be given a brief discussion on blades and blade angles, together with the explanation of a graphical method for calculating the theoretical head to be developed by an impeller.

45 If we assume the impeller discharge to be zero, then from Equation [1] we obtain

$$W = \frac{144\ p_1 v_1}{\eta_i} \cdot \frac{n}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \quad . \ . \ . \ . \ [18]$$

where η_i = impeller efficiency, being practically identical with η_R, but applying to the conditions of zero discharge only. This expression actually represents the work of compression performed in the impeller against the centripetal forces of the fluid particles. Since these forces vary from a minimum at the impeller entrance to a maximum at the discharge, the work done in forcing one pound of fluid from entrance to exit is equal to the average value of the centripetal force multiplied by the radial distance through which the pound of fluid is to move (with a negligible velocity). Such being the case, we have

$$F \times (R_2 - R_1) = W \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ [19]$$

where

F = centripetal force $= \frac{N^2}{5908}(R_2 + R_1)$

R_2, R_1 = actual outer and inner radii of the impeller in feet

N = r.p.m. of impeller

W = work of compressing 1 lb. of fluid in the impeller in ft-lb.

the mean pressure times the mean volume for any stage, or group of stages, must be constant. Hence for isothermal compression

$$\left(\frac{p_{n+1} + p_n}{2}\right) \times V_n = C' = \text{constant} \quad . \quad . \quad . \quad . \quad . \quad [16]$$

From [15] we get

$$V_n = \frac{C}{p_{n+1} - p_n}$$

and substituting this value of V_n in [16], we have

$$\frac{p_{n+1} + p_n}{2} \times \frac{C}{p_{n+1} - p_n} = C'$$

or

$$\frac{p_{n+1} + p_n}{p_{n-1} - p_n} = 2\frac{C'}{C} = a$$

from which we find that

$$p_{n+1} = p_n\left[\frac{a+1}{a-1}\right] = p_n k \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [17]$$

where $$k = \frac{a+1}{a-1}.$$

40 As a general law for increase of pressure per stage, or group of stages, we then have the pressure increase represented by the terms of a geometric progression of which the argument is k.

IMPELLERS AND DIFFUSERS

41 In the design of impellers and diffusers for turbo-compressors we are forced to depend on the theory of centrifugal pumps, which at its best is far from being satisfactory or complete for centrifugal pumps even, and of course is far less so when applied to turbo-compressor calculations. However, by making certain assumptions it is possible to utilize the theory with quite satisfactory results.

42 In order to simplify the analysis of fluid flow it is necessary to make the following assumptions:

a That the fluid has stream-line motion, i.e., the fluid particles move in non-intersecting curves

b That the fluid will, since it is elastic, completely fill the channel of flow

c That the motion of the fluid is steady, i.e., each fluid particle has the same mean velocity as every other particle

which is moving along with it through any section and the static pressure is the same at all points of a section which is at right angles to the fluid flow

d That the fluid is non-compressible.

THE IMPELLER

43 The function of the impeller is to impart a certain amount of energy to the fluid passing through it, with a maximum of efficiency. That is, friction, fluid impact, and turbulence should be reduced to a minimum.

44 On the assumption that the speed of rotation and the rate of fluid flow are known, there will be derived expressions for the actual and effective impeller radii, and the axial depth of the impeller channel at any radius. In addition to these there will also be given a brief discussion on blades and blade angles, together with the explanation of a graphical method for calculating the theoretical head to be developed by an impeller.

45 If we assume the impeller discharge to be zero, then from Equation [1] we obtain

$$W = \frac{144\ p_1 v_1}{\eta_i} \cdot \frac{n}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \quad . \ . \ . \ . \ [18]$$

where η_i = impeller efficiency, being practically identical with η_R, but applying to the conditions of zero discharge only. This expression actually represents the work of compression performed in the impeller against the centripetal forces of the fluid particles. Since these forces vary from a minimum at the impeller entrance to a maximum at the discharge, the work done in forcing one pound of fluid from entrance to exit is equal to the average value of the centripetal force multiplied by the radial distance through which the pound of fluid is to move (with a negligible velocity). Such being the case, we have

$$F \times (R_2 - R_1) = W \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ [19]$$

where

F = centripetal force $= \dfrac{N^2}{5908}\ (R_2 + R_1)$

R_2, R_1 = actual outer and inner radii of the impeller in feet

N = r.p.m. of impeller

W = work of compressing 1 lb. of fluid in the impeller in ft-lb.

the mean pressure times the mean volume for any stage, or group of stages, must be constant. Hence for isothermal compression

$$\left(\frac{p_{n+1} + p_n}{2}\right) \times V_n = C' = \text{constant} \quad . \; . \; . \; . \; . \quad [16]$$

From [15] we get

$$V_n = \frac{C}{p_{n+1} - p_n}$$

and substituting this value of V_n in [16], we have

$$\frac{p_{n+1} + p_n}{2} \times \frac{C}{p_{n+1} - p_n} = C'$$

or

$$\frac{p_{n+1} + p_n}{p_{n-1} - p_n} = 2\frac{C'}{C} = a$$

from which we find that

$$p_{n+1} = p_n\left[\frac{a+1}{a-1}\right] = p_n k \quad . \; . \; . \; . \; . \; . \; . \quad [17]$$

where

$$k = \frac{a+1}{a-1}.$$

40 As a general law for increase of pressure per stage, or group of stages, we then have the pressure increase represented by the terms of a geometric progression of which the argument is k.

IMPELLERS AND DIFFUSERS

41 In the design of impellers and diffusers for turbo-compressors we are forced to depend on the theory of centrifugal pumps, which at its best is far from being satisfactory or complete for centrifugal pumps even, and of course is far less so when applied to turbo-compressor calculations. However, by making certain assumptions it is possible to utilize the theory with quite satisfactory results.

42 In order to simplify the analysis of fluid flow it is necessary to make the following assumptions:

a That the fluid has stream-line motion, i.e., the fluid particles move in non-intersecting curves

b That the fluid will, since it is elastic, completely fill the channel of flow

c That the motion of the fluid is steady, i.e., each fluid particle has the same mean velocity as every other particle

which is moving along with it through any section and the static pressure is the same at all points of a section which is at right angles to the fluid flow

d That the fluid is non-compressible.

THE IMPELLER

43 The function of the impeller is to impart a certain amount of energy to the fluid passing through it, with a maximum of efficiency. That is, friction, fluid impact, and turbulence should be reduced to a minimum.

44 On the assumption that the speed of rotation and the rate of fluid flow are known, there will be derived expressions for the actual and effective impeller radii, and the axial depth of the impeller channel at any radius. In addition to these there will also be given a brief discussion on blades and blade angles, together with the explanation of a graphical method for calculating the theoretical head to be developed by an impeller.

45 If we assume the impeller discharge to be zero, then from Equation [1] we obtain

$$W = \frac{144\ p_1 v_1}{\eta_i} \cdot \frac{n}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \quad . \ . \ . \ . \ [18]$$

where η_i = impeller efficiency, being practically identical with η_R, but applying to the conditions of zero discharge only. This expression actually represents the work of compression performed in the impeller against the centripetal forces of the fluid particles. Since these forces vary from a minimum at the impeller entrance to a maximum at the discharge, the work done in forcing one pound of fluid from entrance to exit is equal to the average value of the centripetal force multiplied by the radial distance through which the pound of fluid is to move (with a negligible velocity). Such being the case, we have

$$F \times (R_2 - R_1) = W \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ [19]$$

where

F = centripetal force $= \dfrac{N^2}{5908}\,(R_2 + R_1)$

R_2, R_1 = actual outer and inner radii of the impeller in feet

N = r.p.m. of impeller

W = work of compressing 1 lb. of fluid in the impeller in ft-lb.

the mean pressure times the mean volume for any stage, or group of stages, must be constant. Hence for isothermal compression

$$\left(\frac{p_{n+1} + p_n}{2}\right) \times V_n = C' = \text{constant} \quad . \quad . \quad . \quad . \quad . \quad [16]$$

From [15] we get

$$V_n = \frac{C}{p_{n+1} - p_n}$$

and substituting this value of V_n in [16], we have

$$\frac{p_{n+1} + p_n}{2} \times \frac{C}{p_{n+1} - p_n} = C'$$

or

$$\frac{p_{n+1} + p_n}{p_{n-1} - p_n} = 2\frac{C'}{C} = a$$

from which we find that

$$p_{n+1} = p_n\left[\frac{a+1}{a-1}\right] = p_n k \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [17]$$

where
$$k = \frac{a+1}{a-1}.$$

40 As a general law for increase of pressure per stage, or group of stages, we then have the pressure increase represented by the terms of a geometric progression of which the argument is k.

IMPELLERS AND DIFFUSERS

41 In the design of impellers and diffusers for turbo-compressors we are forced to depend on the theory of centrifugal pumps, which at its best is far from being satisfactory or complete for centrifugal pumps even, and of course is far less so when applied to turbo-compressor calculations. However, by making certain assumptions it is possible to utilize the theory with quite satisfactory results.

42 In order to simplify the analysis of fluid flow it is necessary to make the following assumptions:

a That the fluid has stream-line motion, i.e., the fluid particles move in non-intersecting curves

b That the fluid will, since it is elastic, completely fill the channel of flow

c That the motion of the fluid is steady, i.e., each fluid particle has the same mean velocity as every other particle

which is moving along with it through any section and the static pressure is the same at all points of a section which is at right angles to the fluid flow

d That the fluid is non-compressible.

THE IMPELLER

43 The function of the impeller is to impart a certain amount of energy to the fluid passing through it, with a maximum of efficiency. That is, friction, fluid impact, and turbulence should be reduced to a minimum.

44 On the assumption that the speed of rotation and the rate of fluid flow are known, there will be derived expressions for the actual and effective impeller radii, and the axial depth of the impeller channel at any radius. In addition to these there will also be given a brief discussion on blades and blade angles, together with the explanation of a graphical method for calculating the theoretical head to be developed by an impeller.

45 If we assume the impeller discharge to be zero, then from Equation [1] we obtain

$$W = \frac{144\ p_1 v_1}{\eta_i} \cdot \frac{n}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \quad \ldots\ldots \quad [18]$$

where η_i = impeller efficiency, being practically identical with η_R, but applying to the conditions of zero discharge only. This expression actually represents the work of compression performed in the impeller against the centripetal forces of the fluid particles. Since these forces vary from a minimum at the impeller entrance to a maximum at the discharge, the work done in forcing one pound of fluid from entrance to exit is equal to the average value of the centripetal force multiplied by the radial distance through which the pound of fluid is to move (with a negligible velocity). Such being the case, we have

$$F \times (R_2 - R_1) = W \quad \ldots\ldots\ldots\ldots \quad [19]$$

where

F = centripetal force $= \dfrac{N^2}{5908}\ (R_2 + R_1)$

R_2, R_1 = actual outer and inner radii of the impeller in feet

N = r.p.m. of impeller

W = work of compressing 1 lb. of fluid in the impeller in ft-lb.

Then, from [26] we have

$$k = \frac{m^2 \times \overline{OC} \times \overline{OD} \times \cos \alpha_2}{n \times \overline{OK} \times \overline{OE}} \quad . \ . \ . \ . \ . \ . \ . \quad [27]$$

and, in feet, $$H = k \times \overline{OE} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad [28]$$

58 The value of the angle α_2 is dependent upon the angle made by the vector w_2 with a tangent to the arc whose radius is r_2 (Fig. 7) at the point from which w_2 is laid off, and the magnitude of the discharge. It varies between 30 and 50 deg., a good average being 40 deg.

59 *Second Case.* When the water does not enter the impeller radially the same construction can be used with some additions

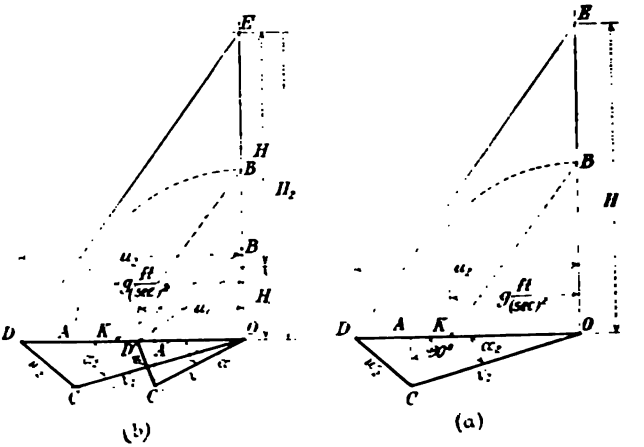


FIG. 10 GRAPHICAL METHOD FOR CALCULATING THEORETICAL HEAD OF FLUID

to account for the term $u_1 c_1 \cos \alpha_1$ in [24] which is subtracted from $u_2 c_2 \cos \alpha_2$. The method of laying out the diagram, see Fig. 10 *b*, is self-evident and requires no further explanation.

DIFFUSERS

60 On discharge from the impeller the energy content of the fluid, as previously shown, is largely in the potential and kinetic form. For a perfect impeller the division between the two would be nearly equal. Actually, however, this is not the case, the potential energy being smaller in value than the velocity energy. That is, the pressure head will be less than the velocity head. In

order to convert part of this velocity head into pressure head, use is made of so-called diffusers. These can be made in two forms, the free vortex and the guide vane. As a matter of fact, the second or guide-vane diffuser is really at normal loads a free-vortex diffuser, providing the guide vanes have been properly designed. Of course this is not the case under ordinary conditions, particularly in a multiple-stage machine where it is desirable that the discharge from the diffuser shall be radial, or at least not far from radial.

61 On discharge from the impeller a fluid particle, if unhindered, will trace out a spiral path providing the space surrounding the discharge side of the impeller has sufficient radial depth. In other words, the fluid as a whole will have a free vortical motion and it can be shown that the path of any fluid particle is a logarithmic spiral, the equation of which is

$$r = e^{k\theta} \quad \ldots\ldots\ldots\ldots [29]$$

where

e = 2.7183

k = constant

θ = angle of radius vector with entrance position.

62 With free vortical motion of the discharge the maximum possible gain of pressure is given by Equation [4]. Due to instability of flow, friction of side walls, impact, etc., which cause the stream lines to be widely divergent from the theoretical the gain of pressure given by [4] is not attainable in practice. If a free-vortex diffuser is used it is essential that its radial depth be sufficient for the vortical motion to develop and thus secure as large energy conversion as possible. The axial depth of the diffuser on the entrance side should be the same or slightly greater than that of the impeller discharge. This axial depth may remain constant or gradually diverge from inlet to outlet of diffuser. It is of course quite essential that the inner surfaces of the diffuser shall be as smooth as possible in order that friction shall be eliminated.

63 Because it is practically impossible to obtain non-turbulent motion in a free-vortex diffuser the use of guide vanes has been introduced. These are located in the diffuser channel and divide it up into several separate sections, through which the fluid can flow with a reduced tendency to become turbulent. As a consequence the conversion efficiency is higher than in the case of the diffuser without guide vanes. In practice it is difficult to transform much more than 50 per cent of the kinetic energy possessed

by the fluid on entrance to the diffuser into pressure energy by the time the fluid is discharged. With a guide-vane diffuser the best conversion efficiency is obtained at normal load when the fluid leaves the impeller in such a direction as to enter the diffuser sections with a minimum of shock, eddying, etc.

64 The number of guide blades should be sufficient to give the desired uniformity of flow without undue friction, usually one-half to two-thirds as many guide blades as impeller blades. If the same number of blades are used in both there may result severe periodic impulses or vibrations due to the fact that the clear cross-section of the fluid channels at discharge of impeller and entrance to diffuser are increasing and decreasing in unison.

65 Theoretically, the shape of the guide blades should be that of a logarithmic spiral to conform with the free path which the fluid tends to follow on leaving the impeller. Actually, the blade shape is governed by other requirements. A favorite form of guide-blade construction is that shown in Fig. 7, and is similar to those used on turbine pumps. The cavity inside the blades is connected with the cooling-water circuit and thus some reduction in the temperature of the fluid is obtained while it flows through the diffuser. The blade curves are customarily volutes or involutes on the entrance side of the diffuser and a combination of circular arcs and straight lines for the remainder of the blade length.

(C) POWER CONSUMPTION AND EFFICIENCY

66 In order to compete successfully in the industrial field with the reciprocating compressor, it is of great importance that the overall efficiency of the turbo-compressor be made as high as possible. That is, the difference between the compressor input (work delivered to the compressor) and output (work obtained from compressor) should be reduced to a minimum. The area of the *pv*-diagram represents the useful work done in the compressor but it does not cover the pneumatic, mechanical and volumetric losses. If η_p, η_m, η_v, η, are the pneumatic, mechanical, volumetric, and overall efficiencies, respectively, then the shaft horsepower must be given by the following expression neglecting the kinetic energy of discharge:

$$\text{Shaft Hp.} = \frac{0.261 \times L \times p_1 v_1}{\eta} \times \frac{n}{n-1}\left(\left[\frac{p_2}{p_1}\right]^{\frac{n-1}{n}} - 1\right) \quad . \quad [30]$$

where

η varies between 0.55 and 0.65 and equals $\eta_p \times \eta_m \times \eta_v$

L = lb. of fluid compressed per sec.

$p_2 - p_1$ = total pressure rise in the machine in lb. per sq. in. abs.

n = compression exponent and is calculated by Equation [**10**].

67 The mechanical efficiency η_m is the ratio between the total power actually delivered to the fluid and the shaft or brake horsepower. It is a measure of the power losses, made up of rotation loss (friction between outer surfaces of impellers and the surrounding fluid) and bearing loss or similar losses due to friction between metal surfaces. The rotation loss will vary between 10 per cent and 20 per cent of useful output of the machine, being determined by the density of the fluid, the speed of rotation, and the impeller diameters. For a single impeller it can be calculated very approximately from the following expression:

$$\text{Friction Hp.} = 0.44 \times 10^{-16} \times \delta_m \times N^3 \times D^5 \quad . \; . \; . \quad [\mathbf{31}]^1$$

where

N = r.p.m.

D = impeller diameter in ft.

δ_m = mean density of fluid for the stage.

68 Equation [**31**] is based upon experiments with constant-thickness disks in open air. It is readily seen from [**31**] why it is preferable to employ a large number of small-diameter impellers in place of a lesser number with large diameters. With a given discharge any increase in pressure must be accompanied by either an increase in speed or else an enlargement of the impeller diameter. Since the frictional resistance varies as the speed cubed and the impeller diameter to the fifth power, it is more economical to use high speeds with small impellers.

69 The volumetric efficiency is, of course, the ratio of the fluid actually delivered by the compressor to that taken into the compressor, and will vary between 96 and 98 per cent, a good average being 97 per cent. Most of this leakage will occur in the last stage between the inlet to that stage and the atmosphere, and is not materially affected by the number of stages.

70 Under ideal conditions the compression would follow the

[1] See pp. 220 and 223, Bulletin of the Bureau of Standards, vol. 10, on Windage Resistance of Steam-Turbine Wheels.

isothermal curve BM on the pv-diagram, Fig. 3. The compression efficiency will be, when referred to the isothermal,

$$\eta_c = \frac{\text{isothermal work}}{\text{actual work}} = \left(\frac{n-1}{n}\right)\frac{T_1}{T_2 - T_1}\log\frac{p_2}{p_1} \quad . \quad . \quad . \quad [32]$$

and will vary between 60 and 67 per cent.

71 The need of cooling the fluid as it passes through the compressor is emphasized by [32]. The heating of the fluid during compression renders it difficult to obtain efficiencies which are materially better than those attainable with reciprocating machines of modern construction. The amount of heat to be dissipated is more than that due to adiabatic compression since the mechanical energy loss due to short-circuiting, eddy currents, fluid impact, and fluid friction reappears as heat energy stored very largely in the fluid itself. By careful design of the cooling-water passages in the casing and diffusers, and by use of a large number of stages, it is possible to carry off most of this sensible heat. In spite of the use of a water-cooled housing, the temperature rise of the air in some turbo-air compressors has been over 200 deg. fahr., the inlet temperature being in the neighborhood of 70 deg. fahr. In the first few stages the cooling effect is small because of the small temperature difference between the air and the cooling water, but in the later stages the compression approaches very closely to being isothermal, as we already know, and the temperature of the air when discharged will range between 150 deg. fahr. and 175 deg. fahr. for discharge pressures of from 75 to 120 lb. per sq. in. gage.

DISCUSSION

Arthur M. Greene, Jr. (written). The paper is of the type which is helpful to the Society in pointing out the theory underlying the action and design of commercial apparatus.

The writer was unable to study all parts of the paper with care as it was not received in time, but he does not agree with certain formulae used by the author and certain statements do not seem to be correct.

In Eq. [3] Bernoulli's equation for an incompressible fluid is employed. The compressor uses a compressible fluid and there is a removal of heat during the action. Eq. [3] should include a term for the heat removed and for the change in internal energy which is called sensible heat in a later paragraph. The same criticism also applies to Eq. [4], in which case a complete energy equation must be applied.

The writer could not follow the reasoning of Par. 20, especially that part which refers to the fractional values $\frac{3}{4}$ for the ratios of the successive steps.

Eq. [11], which is similar to one for hydraulic apparatus, does not seem to the writer to be a good one for gases unless θ depends on all dimensions of the runner.

In Par. 42, the writer does not see how assumption a is at all possible in the design of turbo-compressors.

Eq. [18] would not yield anything if the discharge were zero, because in this case v_1 would be zero. Eq. [18] is the form to which Eq. [1] would reduce if there is no change in kinetic energy.

Eq. [22], [23] and [24] cannot be simplified by the omission of the internal energy term, here called the sensible heat, as this term in adiabatic action is equal to $\frac{1}{k-1}$, or 2.5 times the pressure term.

The Author. In reply to Professor Greene's criticisms, the author would like to say that a rigid thermodynamic analysis of the fluid processes in a turbo-compressor is an exceedingly difficult undertaking, particularly so in view of the meager experimental facts with which to check one's theory. Such being the situation, we are practically obliged to solve a difficult problem in an exceedingly crude manner and this condition, with respect to turbo-compressors, will continue to exist until more exhaustive investigations have been undertaken on such machines and placed at the disposal of the designing engineer.

Referring to criticisms of Eq. [3] and [4], a more exact value for $p_3 - p_2$ can be derived with the assistance of the fundamental equations for the flow of an elastic fluid. These equations are

$$\frac{w_e^2 - w_d^2}{2g} = (u_e + p_e v_e) - (u_d + p_d v_d) - Jq \quad . \quad . \quad . \quad [33]$$

$$\frac{w_e^2 - w_d^2}{2g} = -\int_{p_d}^{p_e} v\,dp - z \quad . \quad . \quad . \quad . \quad . \quad . \quad [34]$$

w_e and w_d = radial components of the relative entrance and discharge velocities of the diffuser

u_e and u_d = internal energy in foot pounds

p_e and p_d = static pressures
v_e and v_d = specific volumes
q = heat subtracted by cooling water
z = friction work

It is evident in order to utilize Eq. [33] and [34], that some fairly definite knowledge must be available relative to the values of q and z, and the temperatures of the fluid on entrance into and discharge from diffusers. Such information we do not possess and in view of this fact we are forced to utilize an equation such as Eq. [3], which of course strictly applies to the flow of an incompressible fluid only. It is possible to make rough estimates of these necessary values and calculate the total pressure rise from the more exact thermodynamic formulæ, but I seriously doubt if the results secured would be much more accurate than if Eq. [3] and [4] are used.

With respect to Par. 20, the use of the fraction $\frac{3}{4}$ is based on the following reasoning. Let $x_3 = x_2^2$, $x_4 = x_2^3$, etc. Then

$$\frac{x_1}{x_2} = \frac{1}{x_2} = \frac{x_2^2}{x_2^2} = \frac{x_2^2}{x_2^3} = \frac{x_2^3}{x_2^4} = \frac{x_2^{n-1}}{x_2^n} = \text{constant} \quad . \quad . \quad [35]$$

from which relations it is easily seen that (Fig. 5)

$$\left.\begin{aligned} \log x_2^2 &= 2 \log x_2 = \log r_2 \\ \log x_2^3 &= 3 \log x_2 = \log r_3 \\ \log x_2^n &= n \log x_2 = \log r_n \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad [36]$$

The ratios of Eq. [35] show that for $x_2 = \frac{3}{4}x_1$ (where $x_1 = r_1 =$ unity), $x_3 = \frac{3}{4}x_2$, $x_4 = \frac{3}{4}x_3$, etc. It is evident that convenient ratios other than $\frac{3}{4}$ might be utilized in laying out the logarithmic curve ab of Fig. 4.

Referring to Eq. [11], θ is determined by the shape of the runner, as well as the blade type, angles, etc.

The criticism of assumption a, Par. 42, is undoubtedly correct, and a can be accepted as applying to turbo-compressors only to the extent that assumption d can be applied. As a matter of fact, practically nothing is definitely known about the conditions of flow in such a machine and assumptions such as I have made in Par. 42 can only apply to a limited extent.

With respect to Eq. [18]. I cannot agree with Professor Greene. In the first place there will always be some flow, if only a short-circuiting around the outer side of the impeller and back into the impeller entrance, and consequently W will not be zero. Again, v_1, the specific volume, does not become zero, or say, of greatly diminished value with reduced or zero discharge. As a matter of fact v_1 will diminish as the discharge is diminished, until for the first stage v_1 will be the specific volume of the fluid in the suction line, under conditions of zero flow, while the initial specific volumes for the other stages will be determined by the static pressures developed by each preceding stage. It is of course quite evident that if v_1 were zero, then the density would become infinite, an impossible condition.

Referring to the criticism of Eq. [**22**], [**23**], and [**24**], I believe that Professor Greene is in error when he states that the sensible heat cannot be neglected, because the sensible heat term is 2.5 times the pressure term. The factor for an adiabatic change is equal to $\frac{1.4 - 1}{1.4}$ and is approximately 0.286 power of the pressure term instead of 2.5 times as stated by Professor Greene.

No. 1718

KEROSENE AS A FUEL FOR HIGH-SPEED ENGINES

By Laurence F. Seaton, Lincoln, Neb.
Member of the Society

Although considerable work has been done toward the use of kerosene as a fuel for internal-combustion engines, there still exists a controversy regarding the correct design of a vaporizing system. There has also been very little work done toward determining the distribution of the heat evolved by the fuel. To study these problems the author accordingly made a series of tests on a high-speed heavy-duty type of engine, using kerosene as a fuel. These tests form the basis of the paper, and the apparatus used and the manner of conducting the tests are discussed in detail. The author concludes his paper by pointing out that in his opinion a kerosene motor would be successful if designed so that the piston displacement were higher than is commonly used; if the intake passages were larger and shorter; and if the incoming gas were heated to a temperature considerably above that of the boiling point of kerosene.

At the present time there is a great deal of controversy among the designers of high-speed gas engines regarding the correct design of a vaporizing system to successfully handle kerosene. There has also been very little done in regard to determining the distribution of the heat of the fuel which is used in high-speed engines. Another idea which is likewise much disputed is the effect on the crankcase oil due to particles of unvaporized fuel passing by the pistons and entering the crankcase, and it was therefore decided to fit up a high-speed engine in such a way that the foregoing conditions could be carefully and accurately studied.

DESCRIPTION OF APPARATUS

2 The apparatus used, and upon which these tests were conducted, is shown in Fig. 1. The motor which was tested was a 4½-in. by 5½-in. high-speed, heavy-duty type and was equipped with a Kingston carburetor connected to a specially designed vaporizer.

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

The exhaust gases were piped directly from the exhaust manifold to the calorimeter. There was also provided a bypass pipe from the top of the exhaust manifold through the vaporizer and back to the calorimeter, and by means of dampers any amount of the exhaust gas could be sent through the bypass, making it thus possible to maintain any desired temperature in the vaporizer. Pyrometers were placed both in the bypass and in the main-line exhaust pipes. A thermometer was placed in the intake manifold leading to the motor block. The air leading to the carburetor was maintained at a constant temperature by means of an electric heater so arranged that the air was heated when passing through it. A manometer was connected to the intake opening into the carburetor, and in this way the suction pressure was measured.

3 Two Tabor engine indicators were used, one fitted with a 240-lb. spring for measuring the average combustion pressure, and the other with an 80-lb. spring for determining the compression pressure. A Dixie high-tension magneto provided with a starting coupling was used throughout the tests for ignition. A valve was placed in the bottom of the crankcase through which the oil could easily be drained in order that its temperature, gravity and weight might be accurately determined. A water-spray nozzle was located in the intake manifold in order that water might be introduced at high loads to prevent preignition.

4 The weighing system for the cooling water consisted of two large tanks and a scale, and the water was weighed every fifteen minutes. The fuel was weighed on a specially designed scale connected with a beam-bell contact, which thus made accurate readings possible. Fuel was piped from the fuel drum up to the kerosene tank, the drum being so arranged that air pressure could be used to force the fuel through the pipe into the tank on the scale. At high loads it was necessary to fill the tank between readings, and it was therefore only necessary to correct for the time which it took to fill the tank. The water which was used in the manifold to prevent preignition was weighed on a sensitive scale.

5 The calorimeter used consisted of a Wheeler surface condenser, the water passing through the tubes and the exhaust gas [illegible] The exhaust gas thus coming in contact with [illegible] temperature. Since the [illegible] in this apparatus, it was [illegible] gases entered [illegible] plate. By means

of these it was possible to determine the amount of liquid condensed during each run. A gasoline tank, mounted on the side of the dynamometer, was used for starting purposes. A thermometer well was placed in the bypass line near the vaporizer so that the temperatures might be compared with those shown by the pyrometer, also placed in the bypass line.

6 The load carried by the engine was maintained by means of a Sprague electric dynamometer, illustrated in Fig. 1, and the load was so constant from this source that no governor was used on the engine. It was possible, therefore, to study the force of

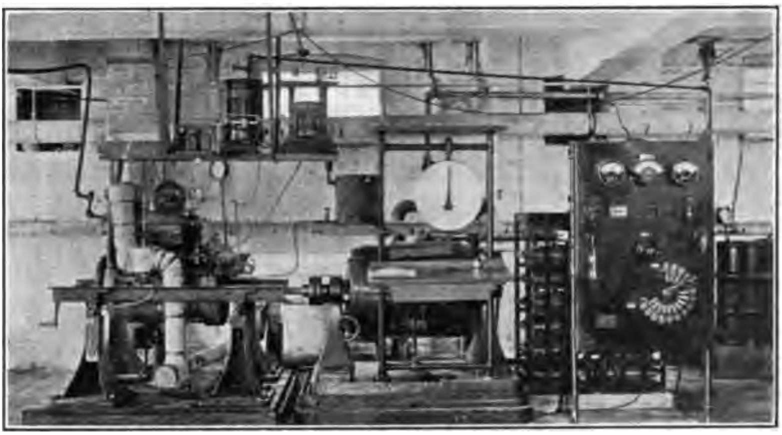


Fig. 1 Apparatus Used in Conducting the Tests

explosions under constant throttle opening. The water used in the calorimeter to cool the exhaust gases was weighed in a set of weighing tanks carried on Fairbanks scales having a range of 0 to 12,000 lb. A water manometer was placed on the exhaust line to determine the back pressure at a point just before entering the calorimeter.

MANNER OF CONDUCTING TEST

7 The engine was started and brought up to a speed of 1100 r.p.m., the speed recommended by the makers, by means of a load connected to a throttle butterfly valve. The load was then put on the engine by means of allowing an electric current from the motor-generator set to pass through the coils surrounding the pole pieces of the dynamometer. This made a generator of the dyna-

TABLE 1 LOG SHEET OF TESTS ON A HIGH-SPEED, HEAVY-DUTY ENGINE USING KEROSENE AS A FUEL

		Engine						Fuel				Cooling Water			Exhaust Calorimeter						Dynamometer			Oil	
Load	Test No.	Temp. of gas intake, deg. fahr.	Temp. of air intake, deg. fahr.	Main pyrometer deg. fahr.	By-pass pyrometer, deg. fahr.	Intake manometer	Max. comb. pressure	Compression	Oil pressure	Wt. lb.	Wt. per b.hp. (hr.)	Increase in temp., deg. fahr.	Weight of water lb	B.t.u.	Incr. in temp. deg. fahr.	Weight, lb.	B.t.u.	Condensate	Temp. exh. gas (out), deg. fahr.	Exh. pressure	Net load, lb.	R.p.m.	Hp.	Temp. deg. fahr.	Wt., Gain and Loss
Load 20 lb.	1	112.8	114.5	1261	150.0	7.5	228.5	..	26.5	42.40	.8872	112	1957.5	219140	4.19	61720	258600	..	77.43	5.5	21.02	1137.5	23.78	160	+ .66
	3	141.5	114.4	1227	351.0	7.4	222.6	..	22.4	43.15	.9295	114.1	1808	206100	18.2	11914	225700	45.5	83.9	5.38	20.90	1129.0	23.63	170	+1.14
	5	148.74	114.9	1132	545.9	6.19	258.0	..	21.6	53.40	1.098	114.6	1713	196300	2.55	85240	217300	32.0	81.8	2.84	20.14	1176.5	24.12	168	+2.00
	E1	147.5	125.3	1129	650.0	7.0	266.8	59.0	21.3	49.77	1.105	114.5	1648	188400	12.15	19685	239000	49.5	80.4	4.44	19.6	1148.0	22.50	164	+2.25
	E4	173.2	154.5	1104	650.0	6.3	224.0	55.7	21.2	47.20	1.066	115.2	1580	182000	4.74	65795	312000	59.5	82.5	4.35	20.2	1094.0	22.64	164	+2.95
	E8	202.5	195.5	1064	650.0	6.3	287.0	60.7	21.5	51.2	1.179	117.2	1709	200100	15.8	16255	257000	39.0	82.9	3.4	19.7	1103.0	21.70	159	+1.82
	F	271	279	310	848.0	6.5	242	58.8	21.3	51.95	1.314	116	1660	192500	4.6	35980	165500	26.75	79.6	2.0	18.0	1100.0	19.80	175	+1.95
Load 10 lb.	1	118.4	116.9	1190	145	2.8	117.4	..	21	27.39	1.275	111.2	1433	159700	4.93	23515	116000	..	74.0	3.4	9.88	1119	11.06	150	− .45
	3	146.9	115.1	1006	346	2.91	142.3	..	22	27.25	1.122	118.2	1480	174900	19.85	4256	84500	19.00	78.1	3.1	10.19	1200	12.22	170	− .56
	5	158.8	115.2	979	551	2.2	148.0	..	21	25.10	1.154	115.0	1405	161500	15.20	6160	93600	16.25	79.3	1.5	9.63	1124	10.86	162	+ .45
	E1	158.0	125.0	982	652	2.4	161.0	40	20	23.65	1.050	114.6	1503	172300	20.05	4070	81700	20.00	80.1	1.9	10.2	1107	11.29	162	+ .26
	E4	179.6	154.2	890	650	2.4	181.5	36	20	24.60	1.115	116.0	1550	179800	19.29	3880	74850	17.75	79.8	1.6	10.00	1104	11.04	161	+1.1
	E8	210.4	195.0	927	650	2.7	137.1	33.4	19	28.00	1.265	117.15	1425	167000	15.4	5935	91450	14.75	78.8	1.48	10.00	1107	11.07	160	+ .5
	F	379	404.0	227	820	3.6	114	38.3	19	29.2	1.334	121.9	1426	173600	19.2	4935	84800	17.00	78.7	1.4	10.00	1096	10.96	170	− .06
Load 5 lb.	1	119.2	124.6	853	141	1.28	120.5	..	20.0	25.80	2.330	114.8	1009	115800	3.05	54825	161200	3.22	84.1	1.58	5.0	1118	5.60	140	+ .92
	3	138.9	114.8	941	350	1.50	79.9	..	21.4	19.45	1.582	117.4	1285	150900	18.20	2894	52650	14.25	79.3	1.78	5.4	1132	6.07	160	− .72
	5	166.0	114.9	861	550	1.34	85.9	..	21.0	19.15	1.553	118.5	1233	146200	14.60	3110	45450	12.75	79.9	1.65	5.3	1157	6.21	160	− .80
	E1	165.0	125.1	807	649	1.29	94.0	37	19.9	16.47	1.511	115.6	1298	151000	18.45	2920	53800	11.25	80.4	.66	5.0	1086	5.46	161	− .98
	E4	192.5	155.6	722	650	1.20	82.4	29	20.0	15.55	1.400	114.0	1330	151600	16.93	2885	49800	13.00	79.9	.52	5.0	1111	5.55	158	− .90
	E8	233.7	195.2	726	651	1.2	64.4	25	19.5	16.58	1.52	111.91	1303	146000	15.61	2615	40800	12.75	79.5	.38	5.0	1102.3	5.51	162	− .64
	F	371.4	381.4	136	689	1.7	84.0	21	20.2	7.85	1.434	115.9	647	75800	20.00	1275	25500	6.00	78.8	.38	5.0	1096	5.48	..	..
Load 0 lb.	3	140.3	115.2	779.0	350	.69	28.6	..	21.0	15.90	10.58	116.4	1046	121800	15.35	1860	28550	13.50	78.6	1.37	.7	1122	.763	158	−1.22
	5	178.9	115.3	616.5	552	.60	34.2	..	20.0	10.80	8.25	117.0	1012	118600	11.75	1915	22500	9.25	77.3	.60	.6	1153	.661	154	−1 9
	E1	180.5	125.0	306.0	650	.66	33.3	20	20.0	10.20	7.85	115.1	1145	131900	18.64	1885	35150	11.00	80.0	.20	.6	1081	.649	160	−1.48
	E4	200.7	155.0	495.0	650	.60	26.0	21	19.9	11.65	7.82	113.6	1169	132800	17.35	1845	32050	7.75	80.5	.13	.6	1098	.665	159	−1.5
	E8	245.2	194.9	476.1	628	.60	25.	17	19.67	9.95	6.54	112.61	1078	125200	13.39	1900	25420	7.75	76.2	.2	.69	1097.4	.762	156	−1.55
	F	382.4	432.0	150	665	.8	24.5	..	19.8	4.95	7.63	113.8	557	63400	12.90	955	12310	..	78.6	.36	.60	1087	.652	..	..

Length of test, two hours.

mometer, and the reaction of the pole pieces was measured through a series of calibrated beams to the dial on the dynamometer.

8 A series of tests was first run with various temperatures in the vaporizer, air entering the carburetor at a constant temperature. The temperature of the intake air was maintained at 115 deg. fahr. and the temperature around the vaporizer was increased from an initial 150 deg. by 100-deg. increments until a temperature of 650 deg. was maintained on the bypass pyrometer. This corresponds to a temperature of 785 deg. in the vaporizer, and in order to obtain this condition it was necessary to shunt all the gases through the bypass at low loads. The loads carried under this condition were approximately 5, 10, 15, and 20 hp. The length of test run on each load was two hours. These series of tests are known as tests 1, 2, 3, 4, 5 and 6, and the average results of tests 1, 3, and 5 for loads of 0, 5, 10, and 20 hp. are given in Table 1.

9 Tests were also made increasing the temperature of the intake air by increments of 10 deg.; that is, the first test was run maintaining a temperature of 650 deg. on the bypass pyrometer and 125 deg. on the intake air. This test is known as E-1. The next test, with a constant temperature in the vaporizer and another 10-deg. rise in the intake air, is known as test E-2. This process was continued until a maximum was obtained at which it was thought practicable to run the engine, according to both the amount of power obtained from the engine and to the fuel consumption per hp. The average result of these tests will also be found in Table 1.

10 At the end of the series of E tests, which was thought to cover all conditions commonly met in practice, it was decided to run a test in which the limit of the apparatus was reached as far as furnishing heat to the mixture was concerned. This test is known as Test F. All heat from the exhaust gases was thrown through the vaporizer and all possible current was passed through the heating coils in the air heater.

11 Indicator cards were taken at 15-min. intervals in order that the combustion pressure might be ascertained and also to determine the evenness with which the fuel was burning in the cylinder. A typical card is shown in Fig. 2. The indicator drum was not connected to the engine but simply turned by hand, leaving the indicator cock open. The pressure of successive explosions was clearly shown at the end of each test, when the spark plug was

short-circuited and a light spring used to determine the compression pressure.

12 The oil was cleaned out of the crankcase at the end of each run, when the temperature, gravity and weight were determined. This was done in order that some definite information might be obtained as to the effect of kerosene on the lubricating oil, especially when run without sufficient heat to more or less perfectly vaporize the fuel. The water and condensed fuel were collected in the calorimeter and are indicated in Table 1 as condensate. The condensate was weighed at the end of each two-hour run to determine its amount.

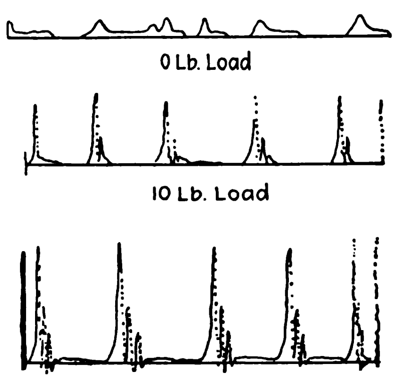


Fig. 2 Typical Indicator Cards

RESULTS

13 From the summary of results and from the curves plotted between delivered hp. and lb. of kerosene per hp-hr. as shown in Fig. 3, it will be seen that the thermal efficiency of the engine was practically constant throughout all heat changes made as previously described. It will be observed, however, that the motor became much more flexible, especially when the temperature of the ingoing air was raised, as this in turn raised the temperature of the ingoing gas materially; much more, in fact, than was possible by increasing the temperature in the vaporizer, for the reason that a small area was exposed to the hot gases in the vaporizer. It was found that at low heats the motor would not idle down below 800 or 900 r.p.m. At the exceptionally high temperatures of the ingoing gases — about 375 deg. it was found that the engine could be idled down to

150 r.p.m. and would pick up almost as well as when burning gasoline. It was also possible to start the engine on kerosene under these conditions.

14 Since the fuel-consumption curves shown in Fig. 3 are practically identical throughout the conditions covered in these tests, it seems probable that about the same percentage of the fuel was burned in the cylinders under all the conditions covered in the test. This is also substantiated by the fact that the amount of kerosene deposited in the crankcase was practically constant for all conditions. The only advantage in heating the ingoing air

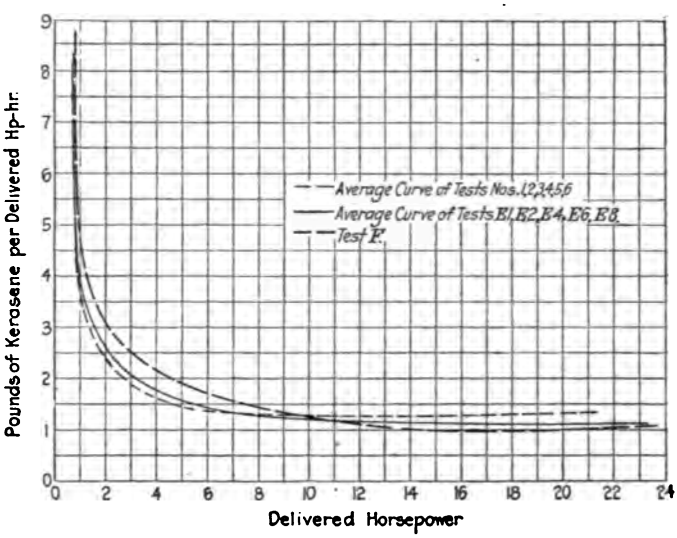


FIG. 3 CURVES OF DELIVERED HORSEPOWER VS. POUNDS OF FUEL

and gas was that it made the motor more flexible. In fact, when the engine was kept up to speed it was found that a greater horsepower could be developed when a low temperature of gas was taken into the cylinders. It seemed quite evident that during all these tests a wet mixture was taken into the cylinders, and that in no case was the kerosene thoroughly vaporized before entering.

15 It was also observed that at any time during the test of higher heat conditions the engine could be made to preignite by lowering the temperature of the air entering the carburetor. A reasonable explanation of this seems to be that in the process of cracking the fuel a gas is formed which is quite explosive, and that when the conditions are slightly changed this explosive gas

is not formed, which does away with preignition; it being fair to assume that this explosive gas, which probably is acetylene, is ignited instantly by the spark, which in ordinary practice occurs several degrees before dead center.

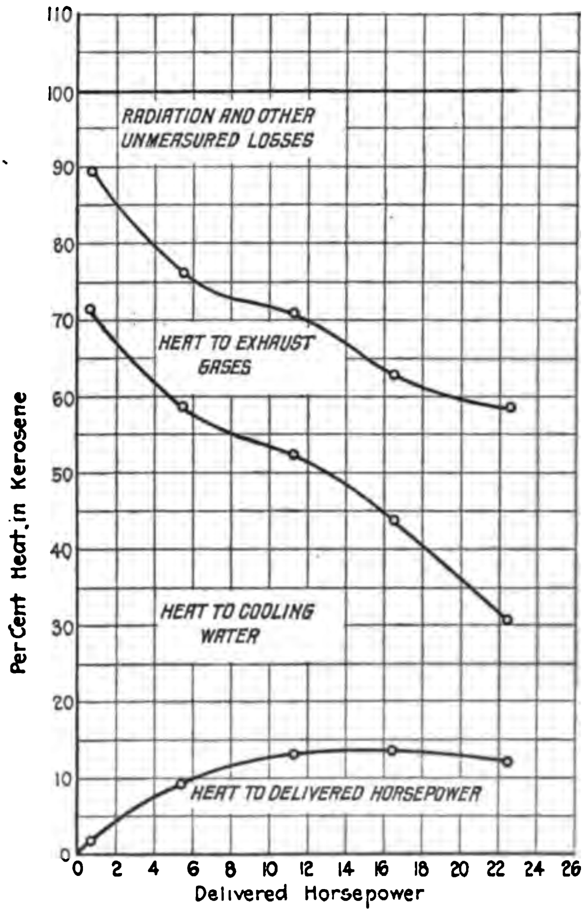


FIG. 4 A TYPICAL HEAT BALANCE

16 The curves of Fig. 4 represent a typical heat distribution throughout the motor, and it will be observed that the maximum thermal efficiency is about 13 per cent. The percentage of heat going to the cooling water slightly decreases after the load increases, while the percentage of heat to the exhaust gases increases after the load increases. It will also be observed that the percentage of heat

dissipated through radiation and other unmeasured losses, which includes the amount of fuel deposited in the crankcase, increases as the load increases. This no doubt accounts for the increase of unmeasured losses as the load increases.

CONCLUSIONS

17 Some of the conclusions to be drawn from these tests are as follows:

a It seems evident that to successfully burn kerosene in the type of internal-combustion engine now in general use, higher heats must be used so that the motor may be made sufficiently flexible to be practicable

b While the tests do not show the motor to be as efficient when operating on kerosene, it is nevertheless believed that this defect can be overcome by designing the motor with higher compression and large intake ports

c It was observed that the lubricating-oil temperature was higher when additional heat was added, which means that a high-flashpoint oil would have to be used.

18 It is the belief of the writer that a motor designed as follows would handle kerosene at all loads successfully: The piston displacement should be greater per horsepower than that commonly used, a higher compression pressure should be obtained, the intake passages should be large and short, and the intake gas should be heated to a temperature considerably above the boiling point of kerosene. This probably would be done with the exhaust gases, necessitating an automatic temperature control at all loads.

DISCUSSION

D. B. Prentice (written). The paper is of especial interest because of two features of the experimental apparatus — first, the preheating of the air by electric heating wires, and second, the condensation of part of the exhaust in a standard steam condenser. The condensation of the moisture in the exhaust gases seems to offer information of as much value, perhaps, as the analysis ordinarily made by Orsat apparatus. The use of a condenser for the determination of this condensate, however, leads to some figures in Table 1 which are hard to understand. The writer would like very much to have the author's explanation of the fact that the condensate in most of the tests is less than the fuel consumed.

With very little error we can consider kerosene as being 15 per cent hydrogen and 85 per cent carbon. One pound of fuel will, therefore, furnish hydrogen sufficient to burn with the oxygen of the air to 1.35 lb. of water. If all the kerosene supplied as fuel is burned, the condensate should be about 35 per cent in excess of the fuel weight. In none of the tests reported in Table 1 is this true. The nearest approach to it is in Test E-4 with a 20-lb. load in which 47.2 lb. of fuel produce 59.5 lb. of condensate. In contrast with these figures there is Test F with the 20-lb. load in which the fuel weighs about twice as much as the resulting condensate. The kerosene must either be burned in some way, pass through the engine to the exhaust unburned or leak past the piston rings to the crankcase. In the last case one would expect a large gain in the weight of lubricating oil, which is not a fact according to the author's figures. It is improbable that much of the kerosene could pass through the engine unburned with an exhaust temperature of 1000 to 1300 deg. fahr. and air enough for the engine to develop full power, but even if it were so, an equal weight would appear in the condensate. It is possible that some of the kerosene is converted to volatile hydrocarbons which pass through the engine unburned and through the condenser as gases, but this is not probable as sufficient air must have been supplied for combustion. Apparently there is a discrep ancy in the weights of fuel and condensate which cannot easily be reconciled.

A second situation presented by the data in Table 1 which the writer finds difficult to explain is the increase in heat loss by radiation and other means as one passes from Test 1 to Test 3 and to Test 5 in the 20-lb. load runs. The actual heat lost by radiation and otherwise unaccounted for doubles between Tests 1 and 3 which develop virtually the same power and speed.

Preheating of the air above 115 deg. fahr. does not seem to increase the thermal efficiency of the engine nor does raising the temperature in the vaporizer. The gain in flexibility, of course, is not indicated clearly by the data in Table 1, although this gain is doubtless considerable. The writer does not understand why, with preheated air, a higher efficiency cannot be secured with this engine. He has made many tests of gas motors of corresponding size which showed fuel consumption as low as 0.5 to 0.65 lb. per hp. per hr. consistently. The best performance represented in Table 1 is

0.8872 lb. of kerosene per hp. per hr. which is 50 per cent higher than can often be secured with gasoline.

Charles E. Lucke thought that there must have been an error in the author's suggestion of heating the intake above the boiling point of kerosene, as a temperature of 900 deg. fahr. is sometimes reached in distilling kerosene. He also called attention to the fact that in operating the engine under load, either no precautions were taken, or if taken not discussed, to control the proportion of fuel to air passing through the carburetor. Unless some effort was made to secure such control, the ratio of fuel and air would change with the temperature. He considered the paper as presenting data of a semi-corroborative nature to that already presented by other investigators of the use of kerosene as motor fuel, and doubted the value of the conclusions as a new contribution to the subject.

The Author. The discrepancy in condensate formed in the first tests can be explained in part, at least, by the collection of tarry and moisture deposits on the condenser tubes. This was explained in the complete report of the test but was not mentioned in the abstract which formed the substance of the paper presented to the Society.

It was found that when the in-going gas was at low temperature a large amount of asphaltic deposit was placed on the tubes. It was necessary to clean the tubes quite often in order to make the condenser efficient enough to keep the out-going gas corresponding with room temperature.

As stated in the original report, little dependence can be put on the results shown by the condensate. However, at high temperatures very little deposit was found, which should make the results shown by the tables more nearly correct, which is the case.

That the efficiency does not increase with the increased temperature is, no doubt, accounted for by the fact that a fixed carburetor setting was kept. As outlined in the original report the test was run to conform as nearly as possible to conditions met in practice with such a motor used for automotive purposes.

Plans are being made to carry on similar work in which a constant ratio of fuel and air will be maintained. In this series of tests the temperature of the gas will be carried above the point at which it will condense before entering the cylinder. It is my opinion that under this condition the condenser or calorimeter will work sufficiently well to give accurate results in regard to the condensate.

No. 1719

A PERFECTED HIGH-PRESSURE ROTARY COMPRESSOR

By Chester B. Lord, St. Louis, Mo.
Member of the Society

Of the three types of rotary compressors, namely, the centrifugal blower, the gear-type and the eccentric rotor with telescopic blade, the last is the least efficient. Particulars are given by the author, however, of a new and improved construction of this type in which leakage, friction and packing troubles have been eliminated to such an extent that a volumetric efficiency of 92 *per cent at seven compressions* (100 *lb. per sq. in.) is said to be commercially obtainable. It is also stated that this compressor will, without adjustment or alteration, handle gas or air, or a liquid, or a gas and a liquid at the same time, and that it is possible to obtain a pressure of* 500 *lb. in a single-stage machine and* 1000 *lb. in three stages with the three units on the same shaft.*

The uses of the rotary compressor are enumerated in the paper and its special advantages are set forth, these being (1) *small weight per unit of capacity,* (2) *small amount of floor space required,* (3) *small initial cost of installation,* (4) *employment of direct drive,* (5) *small amount of headroom required,* (6) *simplification of construction and small amount of adjustment or repair, and* (7) *auto-lubrication.*

ANY one familiar with the various lines of mechanical endeavor must realize that no vital change has been made in compressors in the last thirty years or more. While steam-engine practice has profited greatly by the development of the turbine, and generators and motors have been vastly improved, refrigerating machinery, at least as far as the compressors are concerned, has not been developed.

2 The same cumbersome reciprocating machines that were in use thirty years ago are being used today, with perhaps the difference that they are more cumbersome. There have been improvements in the way of better valves, a little higher speed and minor matters of this kind, but none of them has changed the principle of the machine or resulted in any great economy either in weight,

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

operating space required, or cost of installation. Any reciprocating machine, because of the very principle of reciprocation, must be built exceedingly heavy per unit of capacity and is confined to comparatively low speeds in operation. Because of this and its slow speed of operation the power unit for driving such machines is of necessity also exceedingly heavy and cumbersome. High-speed machines such as electric motors and steam turbines are very rarely used because of the impracticability of connecting them directly to a low-speed reciprocating machine.

Moreover, these machines require a great deal of attention. [illegible] repairs are expensive because in almost every case [illegible] to get at the parts [illegible] by reason of their [illegible]

[illegible] bearings that must [illegible] the delicate [illegible] compression and requires several stages to secure any considerable degree of compression. The gear-type compressor, in which there are two moving members, which, roughly speaking, compress air between two gears with their teeth in mesh, taking it in at one side of the teeth and delivering it at the other, has a very low efficiency, but is able to compress to a higher degree, single-stage, than either of the other types. None of them, however, compresses to over 8 or 10 lb. single-stage. There are many modifications and variations of these three types, but they represent the vast majority of rotary compressors.

7 Once again the mechanical world is about to be startled, perhaps, with a story of a perfected rotary compressor, or at least, it is hoped, interested. Since the writer is not the inventor and has a reputation to consider, it may be assumed that when he makes unqualified statement that it is possible with a rotary compressor to get commercially 92 per cent volumetric efficiency at seven compressions, or 100 lb. per sq. in., that the compressor will, without adjustment or alteration, handle gas or a liquid, or a gas and liquid at the same time, close investigation has been made and the statement is based upon something more than paper facts. Until explained it will raise still further doubt when it is stated that the perfected high-pressure rotary compressor is built on the eccentric-rotor-and-telescopic-blade principle, this being the least efficient of the three types above referred to and the only one that has had no commercial applications except in small water and oil pumps.

8 Before proceeding with the description of the perfected compressor let us analyze the troubles which have been experienced with this type of machine. The first and greatest was that it was necessary to maintain a seal at one point between the rotor and the cylinder. This was a source of large friction losses and the seal was soon broken by wear. Second, the blade pressure against the end plates and the inner surface of the cylinder created great friction loss and still further wore the cylinder at the contact point between the rotor and the cylinder wall. Third, the pressure being all on one side of the blade created such friction that the blade did not work in and out freely. Next, it was found impossible to devise a method of packing square corners, and hence the only commercial application of this type was in the pumping of oil or water at a comparatively high velocity, which service produced the least friction and no important losses.

9 The compressor about to be described was perfected by

W. G. E. Rolaff, of the Research Engineering Company, St. Louis. It is not an invention but the result of long, patient research and the ingenious application of obvious principles. As stated, it is of the eccentric-rotor-and-telescopic-blade type, such as just described, except that the faults mentioned have been eliminated and the elimination has tended toward simplification.

10 Taking the three difficulties enumerated in their order, the first mentioned, that of maintaining a seal at one point between the rotor and the cylinder, is fundamental. Some years ago a Mr. Hermansen developed an engine in which the cylinder revolved at the same peripheral speed as the rotor and was in fact carried by the rotor. Having both moving parts traveling at the same peripheral speed, not at the same *angular* speed, means that there is no slip between the two, and consequently little or no friction. It also reduces to a minimum the blade friction against the outside of the cylinder because the movement of the blade on the wall of the cylinder per compressive revolution is only that represented by the difference between the circumferences of the outside of the rotor and the inside of the cylinder. In other words, in a 100-cu.-ft. size with a cylinder $10\frac{5}{8}$ in. in diameter and a rotor 10 in. in diameter, the blade travel per compressive revolution is only the difference between 31.41 in. and 33.38 in., or less than 2 in., so that it requires approximately 17 revolutions of the rotor for one revolution of the blade upon the cylinder wall. In order that this may be brought out clearly the writer will state that there are no valve openings in the cylinder and that as it travels always in the same plane it makes no difference whether the cylinder is revolving with the rotor, standing still, or reversing itself, as far as compression is concerned. Hermansen's principle, however, solves but one of the many difficulties and by itself is valueless. Mr. Rolaff was unaware of this patent when he started his research work, and even after he had evolved the principle he did not think it patentable, due to its similarity to a roller-bearing detail. Hermansen had had no difficulty in doing so, however, and Mr. Rolaff was obliged to acquire the Hermansen patent.

11 The second difficulty, that of excessive end pressure, was overcome by floating plates at the ends of the cylinder held in intimate contact with it and the rotor by pressure generated by the machine itself. The pressure upon the outer surface of the plates is greater than upon the inner surface by reason of the difference in the area exposed to this pressure. This supplemental pressure is

always in direct proportion to the pressure generated by the machine, hence the additional friction is only what is necessary to keep the cylinder heads in close contact with the rotor and cylinder ends; and it is obvious that where the pressure is even slightly greater on the outside than on the inside there can be no leakage.

12 It will be seen that any motion imparted to the rotor shaft will immediately be communicated to the cylinder by reason of the metal-to-metal contact between the two. In other words, if the rotor is revolved the cylinder will revolve with it and the cylinder plates will float with the cylinder or piston or between the two as the case may be. The friction developed is merely the friction of the wiping contact between the rotor ends and the cylinder plates, since there is very little or no friction in the rolling contact between the rotor and the cylinder.

13 There still remained the question of packing square corners and of friction between the ends of the blades and the end plates. When the rotor is revolved the piston blades which fit loosely in the slot fly out by centrifugal force and remain in contact with the cylinder wall and the cylinder heads, since centrifugal force acting on a wedge keeps them in such contact.

14 It has already been stated that the cylinder revolves with the rotor. Since the piston blades also revolve with the rotor and at approximately the same speed as the cylinder, it follows that the piston blades create very little or no friction on the inner wall of the cylinder. This is also true of the cylinder heads, hence the friction at this point is negligible. At each revolution of the rotor the piston blades travel in the slot to the extreme limit of the crescent-shaped displacement space and back, and in such travel they rub back and forth against the cylinder heads. This is the principal point of friction. But calculating this friction on basis of speed in ft. per min., we find, for instance, that on a machine capable of compressing 100 cu. ft. of free air per min. the piston-blade travel is approximately $\frac{5}{8}$ in., and since a machine of this type would run at about 900 r.p.m., we have the following calculation: $\frac{5}{8}$ in. $\times$ 2 $\times$ 900, which gives 93 ft., and since the surface in contact with the cylinder heads is very small, the total amount of friction generated by this reciprocating action of the piston blade is proportionately small.

15 These blades as developed are of two types, centrifugal and pressure. At present the centrifugal type shown in Fig. 1 is used commercially. This blade consists of four parts, one of which

A, might be termed a blade in itself. Part of this is cut away as shown and on it are imposed three thin blades, *B* and *C*, of any desired metal. It will be noted that the two outside pieces *B* are beveled and that the center piece *C* fits between these, forming a wedge, and that the top of *C* is cut away to allow of movement vertically. Revolving at speed the center piece *C* is thrown out by centrifugal action, the force exerted on each piece *B* being in the direction of the arrow. This presses the *B* pieces against the sides and ends of the cylinder. These blades are self-aligning and self-adjusting, also self-compensating. The force with which these blades *B* strike against the corners is regulated by the angle of their inner sides and the weight of the center piece. There are no springs whatever and the blade is balanced for back pressure by having both sides exposed to chamber pressure with a seat at the top of

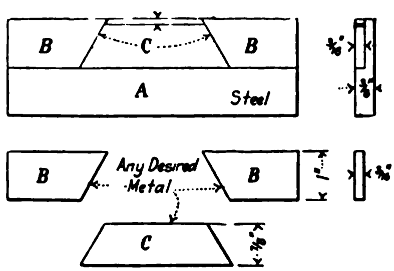


FIG. 1 CENTRIFUGAL TYPE OF BLADE FOR USE IN ROTARY COMPRESSOR WITH ECCENTRIC ROTOR

the slot in back, and as oil from the cylinder wall is scraped off by the blade into the slot, an effective seal is maintained. The blade *A* is about $\frac{1}{32}$ in. short on each end and makes contact with the cylinder wall except at the extreme corner and side. The discharge valve is immediately in front of this blade. Fig. 2 is reproduced from a photograph of a 20-ton rotary refrigerating machine of the type under consideration, a cross-section assembly of which is shown in Figs. 3 and 4.

16 It is evident that in machining the rotor and the cylinder where a rolling contact is maintained between the two, the turning of the cylinder too large or the rotor too small would destroy contact; also that it might happen that the engineer would find himself with too small capacity or that either the rotor or cylinder, or both, might need remachining at any time. This is taken care of ingeniously and all that is necessary if the engineer wishes to increase capacity

is to take a cut off the cylinder, the rotor, or both. For making contact, what might be called a circular wedge is used. This is an eccentric cradle carrying the bearings for the revolving cylinder and is made with a threaded stud led through the outer casing. In practice the rotor and cylinder are assembled without contact and started on test. After a time, by drawing up on the wedge, the proper contact is made. When this is found, the nut is backed off about a quarter of a turn, so that on a 500-cu.-ft. compressor the space is approximately 0.01 in., the oil making a proper seal and contact. Expansion is automatically compensated for.

17 One of the great advantages of the compressor is that it may be direct-connected. It can be run at any speed at which a

FIG. 2 TWENTY-TON ROTARY REFRIGERATING MACHINE

prime mover can be run safely. A 100-ton refrigerating machine is run at 600 r.p.m., a 1-ton or 5-cu.-ft. size at 1800 r.p.m. The first thought of a mechanical engineer at this statement is that the piston speed is entirely too high for efficiency. Let us take from a catalog the data on a reciprocating compressor of 14 in. stroke and having 100 cu. ft. capacity at 200 r.p.m. The piston speed would be 466 ft. per min., or over 200 per cent in excess of a rotary compressor of the same capacity run at 900 r.p.m., with a relative speed of 150 ft. per min. between the rotor and cylinder. The next advantage is in its space requirements. A 100-ton rotary compressor on its base with its prime mover requires only 60 sq. ft. of floor space, whereas a horizontal reciprocating unit of the same capacity requires approximately 1200 sq. ft. plus a building, plus a boiler room,

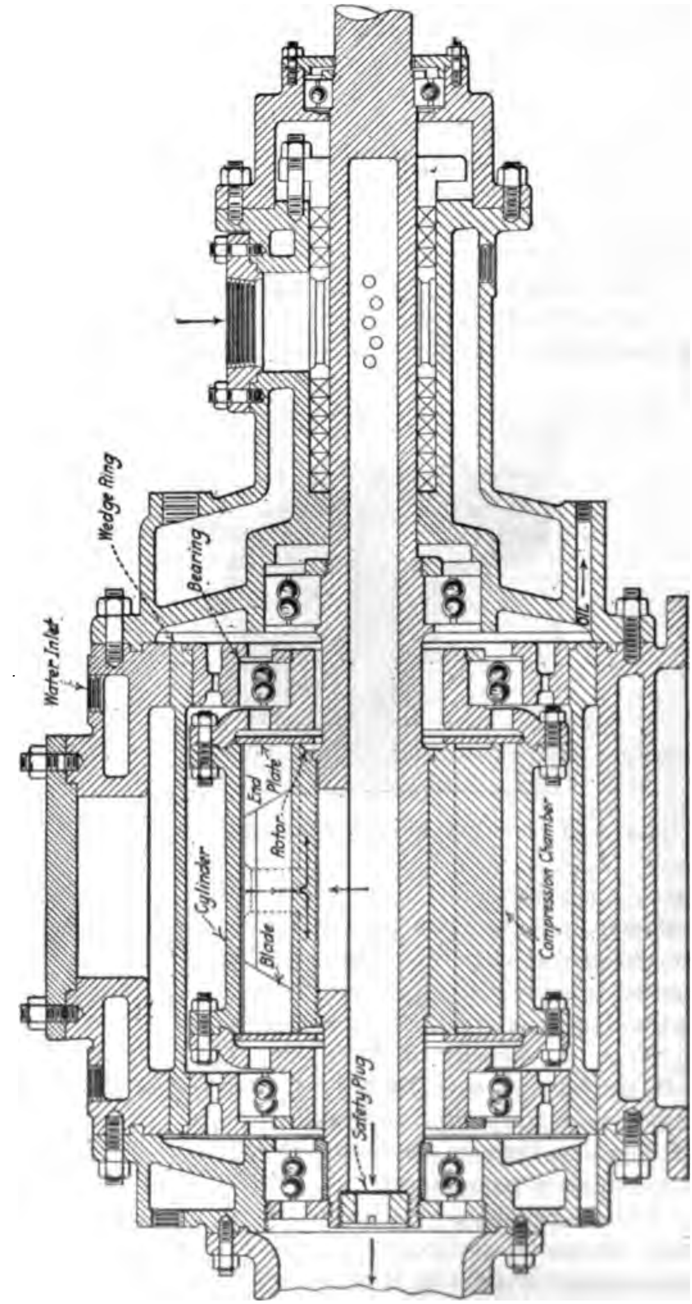


FIG. 3 CROSS-SECTION ASSEMBLY OF 20-TON REFRIGERATING MACHINE SHOWN IN FIG. 2

and weighs 100 lb. per ton capacity as against approximately 8 lb. for the rotary. It is usually possible to replace any pump or compressor with a machine weighing less than the flywheel of the displaced unit. With engineers the prime consideration is always efficiency. With the high-pressure rotary compressor, as before stated, the volumetric efficiency at 100 lb. pressure is 90 to 92 per cent, depending on size, as against 65 to 85 per cent with the reciprocating machine. A number of engineers who tested one of these machines were nonplussed by finding that at no discharge pressure it had a volumetric efficiency of 107 per cent. Never having encountered this phenomenon before they were at a loss to know whether the capacity as given by the company, the instruments furnished by it, or their own instruments or arithmetic was wrong, and for several days they did not give out any conclusions. The manufacturer was of course familiar with it and suspected what their troubles were as such action is not uncommon in uniflow machines, but the best information obtainable is to the effect that the greatest volumetric efficiency found is usually about 102 or 103 per cent.

18 It has been stated that these compressors without adjustment or alteration would pump gas, air or liquid, or all three. Those who have had experience with refrigerating machines know that when ammonia condenses in the refrigerating coils and a "slug," as it is called, happens to get into the compressor, usually a valve or a cylinder head is blown off. One of the tests of the rotary compressor, for the benefit of visitors, is the pouring of a 5-gal. tank of oil into a compressor running at full speed under load, as fast as the inlet pipe will take it. The secret of its ability to take care of this is simple. Since the end plates are held to the cylinder only by differential pressure, when the oil enters the cylinder and cannot be discharged fast enough by the valves, the end plates open up, allow the oil to pass out and then close. It is obvious that when pumping fluid continuously the pressures inside and out would again be different and be relatively maintained, so that no leakage would occur. In other words, the loose end plates act as safety valves without effort, as any necessary area of exit is secured and the oil leaves the cylinder at just sufficient pressure to raise the end plates. This feature is valuable in pumping hot water, or gas and oil.

19 Another feature of this construction which as yet has not been dealt with is the possibility of making the compressor double-acting by placing two blades 180 deg. apart, which would add from 60

to 70 per cent to the capacity of the machine. Until recently, however, no effort has been made to do this as it requires an intake valve, which is troublesome. In the simple type no intake valve is used, the seal between the rotor and cylinder acting as such. The major trouble with all air compressors, reciprocating or otherwise, lies in the valves, and the same trouble was found initially in the rotary compressor. It was aggravated somewhat in the rotary by the fact that cracking or splitting of the valve in its cage was not immediately perceptible, and it was possible for a valve to gradually break up into fine pieces and get into the cylinder. Experiments were undertaken to find a suitable material, and this has been done for cases where the temperatures do not exceed 200 deg. fahr. In the 100-ton size the discharge valve is about 24 in. long by ½ in. wide, so it will be seen that the problem differs from that of the reciprocating machine.

20 The next and most important things to consider are the commercial uses of the rotary compressor. Broadly speaking, it may be said that these embrace every one possible to a compressor of the reciprocating type and some that are not. It is possible to get 500 lb. pressure single-stage; to get 1000 lb. pressure in three stages with the three units on the same shaft; to make it a low-pressure machine, giving it any desired capacity by making the rotor relatively small; and to give even more than 1000 lb. pressure single-stage by reducing the displacement. As a vacuum pump it will pull 28½ in. against a pressure of 180 lb., but to do this continuously with a compressor it is necessary to shut off the oil pipe, and for commercial practice it would be necessary to have a self-contained oiling system. These high single-stage pressures and vacua are of course interesting only as illustrating the possibilities of the rotary compressor as compared with the reciprocating type. It is of course not desirable to compress to 500 lb. without intercooling, and to pump 28½ in. vacuum against 180 lb. pressure is unnecessary and merely demonstrates the excellence of the machine.

21 The firm handling the rotary compressor in question has already been asked to consider the making of blowing engines for blast furnaces, as compressors of the rotary type do not take up more than 5 per cent of the room now necessary. These rotary compressors are in use at the present time as boosters on refrigerating units for both high and low pressures. One of the large packers has four. The small household refrigerating business has not grown because it has been found impossible to secure a dependable compressor.

One manufacturer is at present building units using this rotary compressor. It is a 0.5-cu.-ft. size, runs at 1760 r.p.m., is direct-connected and is perceptibly noiseless (practically noiseless means nothing, absolutely noiseless means too much, so "perceptibly" is used advisedly), as are all other sizes of this compressor. The rotor in this size is $1\frac{7}{8}$ in. in diameter and 1 in. long. Another manufacturer is making the 3-ton refrigerating size. Still another has a contract on which is guaranteed a minimum of 10,000 cu. ft. per year for street-car compressors. Its use for this purpose should be extensive, as no satisfactory reciprocating pump has ever been designed. Another use is as a small individual unit for refrigerating cars, run either from the axle direct or with a motor from a turbine in the engine. Upon arrival at terminals, instead of the expense and waste of icing, it will only be necessary to plug in. Dining cars are still another use to which, among a multitude of others, it may be put. Imagine a cooled Pullman compartment on a hot, dusty day!

22 Used as an air compressor the rotary compressor is ideal, as the air is taken off at the center and the oil thrown out centrifugally to the periphery.

23 No rotary pump is at present made that will prime itself, but this is not true of the compressor when used as a pump. Further, so far as the writer is aware, no reciprocating pump manufactured handles CO_2 successfully for any length of time. Those that do so at all are usually packed with leather, which wears rapidly. Because of the high efficiency and the character of its cylinder construction there would seem to be little doubt of the rotary's ability to handle CO_2 continuously without repairs. This, however, is in the future, which also applies to hydraulic applications.

24 The compressor may be built either vertical or horizontal and in the small sizes up to 5 cu. ft. it is air-cooled. Between this and the 15-cu.-ft. size it is partly water-cooled. In the 20-cu.-ft. size and above it is completely jacketed.

25 Summing up, the advantages of the rotary compressor are:

1 Small weight per unit of capacity, about $\frac{1}{10}$ of that for reciprocating machines
2 Small amount of floor space required, about $\frac{1}{20}$ of that for reciprocating machines
3 Small initial cost of installation
4 Most economical drive known, employing either electric motor or steam turbine direct-connected
5 Small amount of headroom required. The machine is

usually about the same size as the motor required to drive it

6 Simplicity of construction and hence small cost for adjustment or repair

7 Self-lubrication, since the entire machine operates in a bath of oil.

All of the foregoing statements have been demonstrated commercially and hence it would seem that the long-looked-for rotary compressor has arrived.

DISCUSSION

H. V. CONRAD[1] (written). In the claims set forth covering the advantages of the compressor described by the author, those who have followed the development of rotary compressors will recognize history repeating itself.

In the paper as presented, some of the details of design and construction are not sufficiently described or illustrated to give a thorough understanding of them; such, for instance, as how the rotor rotates the cylinder, the arrangement of inlet and discharge ports and discharge valve. In consequence, the writer's comments will be in the nature of seeking further information.

In rotary compressors of this type the sliding blade has been the failing detail, and how long it will maintain its efficiency in actual service in the machine in question can be determined only by duration tests.

The unbalanced side thrusts on the blade during the compression stroke or revolution still exist, but as the in-and-out movement is reduced in this design, wear will be slower. How will this blade be lubricated? Centrifugal force will prevent oil that is scraped off the cylinder wall working its way into the blade slot, where the sliding friction takes place.

The floating end plates, apparently, will rotate with the cylinder, but will have a sliding or wiping action against the ends of the rotor. Judging from the drawing, full final air pressure is always against the outside of the end plates, opposed on the inner side by a pressure varying from atmospheric to 100 lb. per sq. in., say, for about 80 per cent of the revolution of the rotor, producing wear on its ends due to the unbalanced pressure on end plates.

While a volumetric efficiency of 92 per cent is claimed at 100

[1] 122 East 40th St., New York, N.Y.

lb. per sq. in. final pressure, nothing is stated concerning overall efficiency — that is, the actual number of cubic feet of air compressed per horsepower delivered to the machine, which is the ultimate result to be arrived at.

The discharge valve, referred to in Par. 19, designed for a temperature not to exceed 200 deg. fahr. will be subjected to a temperature of 450 to 500 deg. fahr. when air is compressed to 100 lb. per sq. in.

Oscar R. Wikander (written). The advantages of a successful high-pressure rotary compressor, enumerated by the author, will certainly be conceded by anyone familiar with the subject.

In addition to the difficulties mentioned, which are encountered by designers of such machines, the writer would like to add the difficulty of balancing the rotary piston, as the telescopic blade, which moves radially in and out during the revolution of the piston, will naturally tend to unbalance the latter, and for this reason noise and vibration will be produced at higher speeds.

The author has not shown how this difficulty has been overcome, nor does he give in his paper a very clear conception of the arrangement of the cylinder wedge to which he refers in his description. The above points would probably have been brought out if the author had shown a transverse section of his compressor. Such a section would probably also have shown the arrangements of the intake opening and discharge valve, both of which are very important and interesting features.

The paper gives the interesting information that the volumetric efficiency of the compressor is as high as 92 per cent at seven compressions, which is remarkable, but no information is given of the mechanical efficiency which is one of the most important features of the machine.

The arrangement of the blade which automatically packs the square corners and the plates at the ends of the cylindrical rotor is a very ingenious solution of one of the difficulties encountered in the design of such machines.

It would be interesting to know whether machines of this type have been in commercial operation for any length of time and what results have been obtained.

E. H. Bridenbaugh (written). The writer agrees with the author that no vital changes have been made in compressor design in the past thirty years and that a number of devices for compressing air on the rotary principle have been designed, built and marketed. Most of these compressors have developed defects due to excessive wear, heat and horsepower required for operation to the extent that they were considered impracticable. The writer has gone through this experience, having been connected for six years with a concern developing a rotary compressor which showed these defects. About two years ago, however, a radical change in the design of the compressor resulted in placing a machine on the market whose operation has been entirely satisfactory to the extent that difficulty is now experienced in supplying the demand.

The writer believes that the advantages claimed for rotary compressors by the author are realized in the Rolaff compressor. These same advantages are also to be obtained in the Jackson compressor, made by his company, there being, however, differences in speed, capacity and weight in the two machines.

The writer would like to know if the author has attempted using the compressor for pumping coal or oil gas and what effect the tars and other residues have on the lubrication of the compressor. He would also like to inquire about the power requirements for a given capacity and pressure.

In the compressor described by the author is it possible to maintain proper rolling contact between rotor and cylinder when expansion takes place? It is not possible that the two will expand equally, especially in a water-cooled compressor. What will be the result when the bearings become worn? Will new bearings be needed or can the old ones be adjusted?

A volumetric efficiency of 90 to 92 per cent is also obtainable with a Jackson compressor, and a vacuum within 0.3 in. absolute when exhausting to the atmosphere.

The writer would like to ask how it is possible to operate the compressor in a bath of oil without considerable loss in the discharged air.

A. M. Greene, Jr., thought that a cross section of the compressor should be shown and that the author should include a statement of the power required to compress a given amount of air under certain limiting conditions of pressure.

H. Schreck stated that he considered, as the author had said that there was nothing new in the compressor except the arrangement of the blades. However on former designs the wear of the cylinder was so great and uneven as to make engines constructed on this principle unmarketable. The fundamental principle has been used for a quarter of a century.

The Author stated that the compressor was not built by the Research Engineering Company but that manufacturers wishing to utilize the principle were supplied with drawings and licensed to build the compressor. One firm was expected to produce in 1920 between 300,000 and 500,000 of these compressors for use with household refrigerating units.

In answer to a question by Mayo D. Hersey regarding the performances mentioned in Par. 12 of the paper the author stated that these had actually been obtained with a 500-lb. single-stage compressor.

In closing, the author said that an overall efficiency of 79.7 per cent, including motor drive, at 900 r.p.m. had been obtained with one of these compressors.

A discussion centering around the operation of the compressor with rotating rotor and cylinder was finally closed by an explanation by the author. Those taking part in the discussion were Benjamin F. Tillson, L. T. Lowenstein, A. M. Greene, Jr., P. B. de Schweinitz, H. L. Benner, H. P. Porter, H. H. Supplee, Walter J. Wohlenberg, Frank E. Mathewson and Gardner T. Voorhees.

The author, in reply, to H. V. Conrad, wrote that the rotor imparts its own rotating motion to the cylinder by reason of the contact between the rotor and cylinder.

The inlet and discharge port and valves are located in the rotor at opposite sides of the blades, hence their relative position to each other and to the blade which is the compressing element remains the same. This is shown in the cross-section view in Fig. 4.

The internal surface of the cylinder is constantly spread with lubricant which is imparted more or less completely by the blade traveling over this surface. The lubricant is also contained within a compression space by the squeezing action between the rotor and the cylinder. This lubricant is forced against the blade and, of course, into the blade slot because that is the line of least resistance, hence the blade slot really presents the characteristics of a cylinder in which the blade is the piston, the cylinder being without valves but ported to receive incoming pressure and the same ports act as relief ports for the oil that comes out as the blade is pushed back into the slot. The lubricant also forms wedges both on the forward and backward sides of the blade and the blade really travels upon these wedges of oil rather than upon a definite metallic surface. The wear on this blade is very light. The centrifugal force of the oil is totally overcome by the pressure force sending this oil back into the blade slot.

The floating end plates as stated have a frictional load where they are in contact with the rotor and cylinder and ordinarily there should be some wear. This wear however, is very light as proven by actual experience in over two years operation. Since

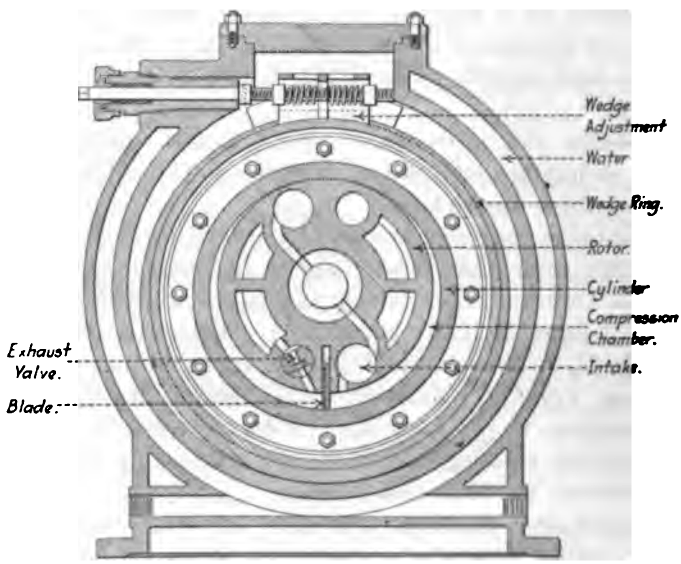


Fig. 4 Cross Section View of Compressor.

the motion between the elements in question is very small the surfaces are well lubricated and the nature of the metal (cast iron) is such that it takes a high-glazed polish, and after that, wear practically stops. Whatever wear takes place must be uniform and hence of no moment.

The discharge valve mentioned as being designed for 200 deg. fahr. was one that was equipped with a cushion. Ordinarily straight metal valves are used which can easily stand the normal temperatures of high compression.

Referring to the discussion by O. R. Wikander, the rotor is balanced statically with the blade out of the slot, the blade itself is an element which only directly can become an unbalancing factor by reason of its own inertia and the unbalanced pressure against the exposed portion of it. This feature of unbalanced running parts is utilized to overcome the compression, by merely calculating the weight of the blade to be such as to act as a counter weight for the compression. Noise and vibration are so thoroughly absent from the machine in operation that they do not require any elaborate foundation irrespective of size or normal speeds.

The overall efficiency in the test referred to was 79.7 per cent.

Some machines have been in operation for a period of very nearly two years although they have not been in constant operation. One machine of 600 cu. ft. capacity has been in constant operation against low pressures of 30 lb. for a period of 100 months operating from 12 to 24 hr. per day including Sundays. The only service necessary on this machine has been in connection with defective bearings.

In answer to the question put by Benjamin F. Tillson, the author wrote that the pump had not been used as a motor or prime mover except in experiments in which the machine was used as an expander of compressed air. No readings were taken of the power recovered but temperature readings were taken and while they were not conclusive they implied that the machine might prove a very successful prime mover such as would be needed for drills and even simple steam engines.

In answer to E. H. Bridenbaugh, the author said that no attempts had been made to pump coal or oil gas and if such gases were unscrubbed and not free of all tar it would not be attempted as this would materially interfere with proper lubrication of the machine.

The power requirements for a given size, so far as experience goes, are less than an equivalent size of reciprocating machine.

Expansion and contraction do not effect the contact between rotor and cylinder because, while this contact is made originally when machine is assembled, it is automatically adjustable to prevailing conditions. This applies particularly to expansion.

It will depend upon the amount of wear on the bearings as to whether they can be repaired or must be renewed. This is true of all other bearings.

The machine will operate in a complete bath of oil without any appreciable loss of oil in the discharged air.

Mr. Schreck is correct in saying that the principle of the eccentric cylinder is not new, and all the author claims is the perfection of an old principle by means of new fundamentals, some of which have never been used, and others never in similar combinations, otherwise they would not have been patentable.

No. 1720

COMMON ERRORS IN DESIGNING AND MACHINING BEARINGS

By Christopher H. Bierbaum, Buffalo, N. Y.
Member of the Society

After presenting ten principles relating to the design, construction and operation of journal bearings, based on what is now known concerning the laws of lubrication, the author discusses among other things oil grooves and their proper distribution, the disadvantages of tight-fitting bushings, proper methods of finishing brasses to provide for expansion when running, and proper methods of clamping bearings during tooling. He then takes up the matter of the tools employed in machining bearings and shows by numerous photomicrographs of finished surfaces the importance of using sharp tools with the proper rake in order that the crystalline structure of the surface material of the bearing may not be so crushed and compacted that it will fail to function as a normal bearing alloy.

IN considering the subject of bearings, their design, construction and lubrication, it is desirable to have in mind the fundamental laws relating thereto — those discovered by Tower, Thurston, Goodman, Lasche, Stribeck and others. Taking the work of these investigators in the light of what is now known concerning these laws, it follows that in the operation of a properly designed, constructed and lubricated bearing we may lay down the following ten principles:

a The bearing surfaces are completely separated by a supporting film of oil

b The friction of operation is the fluid friction in the oil film, and adequate thickness of film is essential

c During construction proper clearance or space should be provided for a normal thickness of oil film

d The advance edge of a bearing surface must be rounded or chamfered off in order to permit a supporting film of oil to form

e The oil film forms most effectively upon a bearing surface whose advance edge is at right angles to the direction of motion

---

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

f An increase of speed increases the thickness of film, all other conditions remaining constant and clearance permitting

g An increase in the viscosity of the oil increases the thickness of film, all other conditions remaining constant and clearance permitting

h The larger the unbroken film of oil, the greater will be the average pressure-supporting capacity per unit area, other conditions remaining constant

i Every unnecessary oil groove or interruption in the continuity of the oil film reduces the supporting capacity of the film

j For every bearing condition there is a film thickness corresponding to maximum lubrication efficiency.

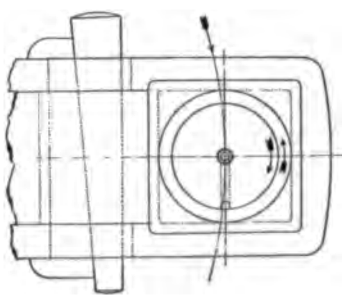


Fig. 1 Practical and Efficient Method of Lubricating a Crankpin Bearing

2 The law governing the proper thickness of oil film has not as yet been investigated to the extent that the importance of the subject demands. In modern machinery the average thickness of film varies from 0.0002 in. to 0.006 in.

PRECAUTIONS REGARDING OIL GROOVES

3 Since the lubrication of every bearing is a study in itself, the application of the known laws can be discussed to the best advantage by considering specific cases. Fig. 1 represents a theoretical ideal, as well as practical and efficient, method of lubricating a crankpin bearing, the journal receiving oil through the crankpin. The rotation of the crankshaft is in the direction indicated by the upper arrow, and in the position shown the engine is on a dead center at a point of reversal of pressure. The direction of relative

motion of the rubbing surfaces is shown by the two arrows at the right of the figure. The oil film enveloping the right half of the crankpin has been completely restored during the stroke just finished, since the oil groove passed over this half of the bearing while no pressure was being exerted upon it. After the dead center has been passed, the entire pressure is then exerted upon the crankpin with its fully restored oil film, and at the same time the oil groove wipes over the other half of the bearing and restores its oil film while no pressure is being exerted upon it, after which it in turn is ready to receive a reversal of pressure upon a fully restored oil film. Thus both halves of this crankpin bearing present alternately, for

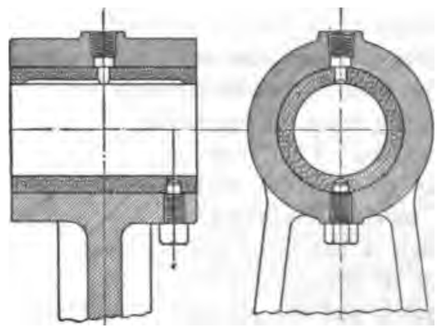


FIG. 2 TRANSVERSE AND LONGITUDINAL SECTIONS OF AN ORDINARY BUSHING DRIVEN INTO A CAST-IRON HOUSING

the maximum pressure of each stroke, a complete and uninterrupted surface for maintaining the film on an area equal to the projected area of the crankpin, but it is impossible to obtain so perfect a condition of lubrication in a bearing having the old-fashioned cross oil grooves which are still too often found in this class of bearing.

4 In case the direction of rotation is to be reversed, the oil groove in Fig. 1 should be placed diametrically opposite to its present position. The function of the oil grooves is of course that of supplying oil. In many cases they are necessary evils, which should be minimized as much as possible by avoiding a useless excess of grooves and especially grooves in the direct line of maximum pressure of the bearing. Unless good and sufficient reason exists to the contrary, oil grooves should be cut parallel with the journal.

DISADVANTAGES OF TIGHT-FITTING BUSHINGS

5 One matter very frequently neglected, or not thoroughly understood, is that of driving in bushings. Fig. 2 shows a transverse and a longitudinal section of an ordinary bushing driven into a cast-iron housing. As all bearing alloys have a temperature coefficient of expansion higher than that of cast iron, and as the bearing is directly subject to the friction of the journal, it follows that the bearing is at a higher temperature than the cast-iron housing and that all bearings must have an appreciable outward expansion when in operation. For this reason bushings should be driven in with just enough pressure to prevent looseness during operation.

6 The practice of driving in bushings so tight that they require reaming before they can be put into service cannot be condemned too severely. The subsequent reaming does not remedy the evil, in that a bushing driven in so tight will and must continue to contract inwardly when in operation, since the outside pressure upon it is not removed by the reaming. In the tooling of a bearing a definite amount of clearance should be provided for, in order to insure the best service conditions. When a bushing is driven in too tight, although it has been reamed before being placed in service, the amount of clearance which it will then have becomes a matter of guesswork and the amount of oil-film space provided an uncertainty. This condition frequently gives rise to heating and bearing troubles, even though the amount of internal contraction may not be sufficient to positively grip or stop the journal. The idea that bushings must be driven in tightly in order to hold them in place is fundamentally wrong, and excessive tightness of the bushing in its housing should always be avoided. For all ordinary machinery it will suffice if a bushing may be driven in place with a blow of the hand or with a small block of wood. The provision for fastening the bushing in place should be such that it will not bind or clamp the latter against outward expansion. In the construction shown in Fig. 2 a set screw engages a hole in the bushing, but does not, however, bottom in the hole.

PROPER METHOD OF FINISHING A CRANKPIN BEARING

7 The finishing of bearings in such a manner that they may expand while warming up is of very considerable importance, and this can be done in many cases by simply giving the matter the full consideration which it deserves. Fig. 3 shows a crankpin bearing

finished in a manner providing for necessary expansion in service. The two edges of the two half-bearings should be "brass bound," that is, they should bear solidly against each other and should exert a pressure against each other somewhat in excess of the maximum crankpin load. The outer surfaces of both bearings near

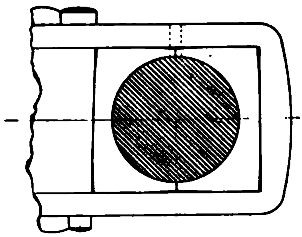


FIG. 3 PROPER METHOD OF FINISHING THE BRASSES OF A CRANKPIN BEARING TO PROVIDE FOR NECESSARY EXPANSION IN SERVICE

these edges should be relieved so as not to bear upon the straps, as shown, somewhat exaggerated, in Fig. 3. The four corners of the bearings should be relieved in like manner so that the horizontal thrust will be mainly borne upon the surfaces falling within the area of the horizontal projection of the crankpin. A bearing thus constructed expands with the first slight increase of temperature, relieving the crankpin instead of clamping it as when these precautions are not taken.

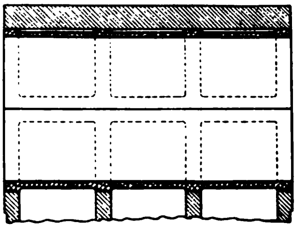


FIG. 4 SPLIT BRONZE BUSHING OF TOO LIGHT A DESIGN TO WITHSTAND THE STRESSES IMPOSED UPON IT

DESIGNING FOR STRENGTH IMPORTANT

8 The matter of proper design of bearings as to strength is likewise of importance. Fig. 4 shows a bearing put in service by one of the oldest-established machine builders and which gave very considerable trouble. It is a split bronze bushing 12 in. long, supporting a journal 6 in. in diameter; the maximum thickness of the

bushing was $\frac{3}{8}$ in. and it was recessed on its back to a depth of $\frac{1}{8}$ in. as shown, leaving a thickness for the larger part of the bearing of only $\frac{1}{4}$ in. The cap of the bearing was held down by ten $\frac{5}{8}$-in. studs. Now it is obvious that by clamping this bearing the bearing surfaces were materially distorted by the pressure consequent on the tightening of the ten nuts. It is also apparent that the thickness of the recessed and unsupported part of the shell was not sufficient to withstand distortion under a fluid film pressure such as would normally be exerted upon it. The trouble with this bearing was therefore due entirely to the fact that it was too light a design.

9 Mention may here be made of another class of bearings which together with their supports are often improperly designed, namely, self-oiling ring bushings. These are nearly cut in half at mid-length in order to provide space in which the ring may operate, and the supporting wall beneath the bushing is slotted correspond-

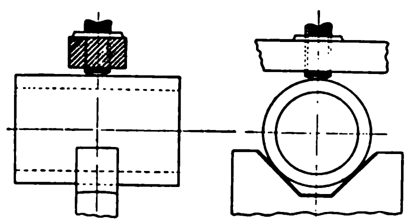


FIG. 5 IMPROPER METHOD OF CLAMPING A BUSHING FOR TOOLING

ingly at the center to permit the insertion of the ring into the oil well. This often unduly weakens the bearing at a point where normally the greatest film pressure is exerted. In general, much of the trouble at present encountered in bearings can be overcome by simply increasing the strength and rigidity of the bearings and their supports, since an insensible amount of distortion and deflection is sufficient to produce all manner of annoyances.

BEARINGS SHOULD BE CLAMPED ENDWISE FOR TOOLING

10 Another matter very often overlooked is that of the manner in which bushings or bearings in general are clamped during tooling. Fig. 5 shows end and longitudinal views of an improperly clamped bushing. With a $\frac{5}{8}$-in. 16-thread set screw and the application of, say, a 50-lb. pull upon a 12-in. monkey wrench, a pressure of 6000 lb., can be carelessly exerted upon the bushing in question, and it must be remembered that the composition of a

bearing material is never one that would be selected for maximum strength. Bushings tooled when clamped in the manner shown in Fig. 6 seldom have a bore that even approaches a true cylindrical surface, and if they are then driven upon an arbor and finished on the outside a very inaccurate product is the result, a bushing that can never give the most satisfactory service. The best and most satisfactory method of holding bushings for tooling is that of clamping them endwise. The importance of this is very generally underestimated; in general, bearings should be held in a manner such as to produce the least possible amount of strain and distortion.

CHAMFERING THE EDGES OF OIL GROOVES

11 The matter of chamfering oil-groove edges deserves special attention in that all advance edges of a bearing should be rounded and chamfered off, and the fact that this work should be done last,

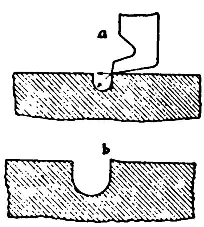
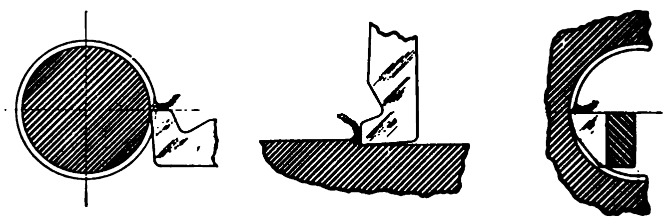


Fig. 6 Sharp Edge Produced at the Edge of an Oil Groove by an Ordinary Tool

after all of the other tooling of the bearing has been completed, is important. At *a* in Fig. 6 is shown an ordinary lathe tool at the edge of a groove on a finished surface. It is obvious that there are being exerted two forces upon any surface which is being tooled, one horizontal or parallel to the finished surface and the other vertical or in a direction normal to that surface. The resultant of these two forces is a force indicated by the dotted arrow, in a direction tending to deflect the edge of the groove obliquely downward, producing an effect which is shown somewhat exaggerated at *b*, Fig. 6. It can readily be appreciated what an injurious effect is produced in a bearing if any tooling is done after the oil grooves have been cut, especially so if the direction of rotation of the journal is opposite to that of the tooling. In all cases sharp edges of the groove prevent the formation of an adequate oil film and should be carefully avoided.

WHY BEARING SURFACES SHOULD BE MACHINED WITH SHARP TOOLS

12 For best results it is very necessary and it is also general practice that a cutting tool should have rake. The tools shown in Fig. 7 would be readily condemned even by a person with a very limited shop experience, and no journeyman machinist would think of setting a lathe tool as shown at *a*, a planer tool as at *b*, or a boring tool as at *c*. In the last mentioned case the amount of normal or radial pressure which it exerts upon the finished surfaces is far in excess of what is necessary. Consider now the standard multiple-cutting-edge reamer which is very often used in finishing bearings. A 2½-in. reamer would have, say, 14 edges and the amount of bursting or internal pressure that it would exert within a bushing would be at least 14 times that exerted in case the bushing were



a b c

FIG. 7 IMPROPER FORMS OF LATHE, PLANER AND BORING TOOLS

reamed with only one improper tool as shown at *c*, Fig. 7. This practice is one that cannot be condemned too severely.

13 The final finishing in bearings should be done by reamers or cutting heads having only one or a very limited number of cutting edges, and these cutting edges should have the proper amount of rake — such as would be given to any other proper cutting tool. Experiments made in a large manufacturing plant on bushings of the same dimensions, from the same lot of material, finished at the same time, some, however, being reamed with the standard multiple-cutting-edge reamer and others with a single cutting blade, showed that after a storage of six months those bushings which had been reamed with a proper single cutting edge retained their accuracy and shape much better than those finished with the so-called standard reamer. The latter bushings exhibited a decided tendency to decrease in inside diameter and to assume inaccurate, elliptical forms. It was also brought out at this time, that not

only did a superior product result from the use of a proper tool, but that by its use the production could be increased from 15 to 20 per cent.

DEFECTS IN BEARING SURFACES REVEALED BY THE MICROSCOPE

14 Another reason why bearing surfaces should be tooled with sharp tools having the proper amount of rake is brought out by a microscopic study of bearing surfaces. In order to obtain the full value of bearing alloys it is necessary that these alloys should be presented as bearing surfaces having their natural crystallization undisturbed. The reason for this is given in the report of the Society's Research Sub-Committee on Bearing Metals.[1] The hard or bearing crystals should be embedded in a softer material, per-

FIG. 8 CROSS-SECTION OF ONE OF THE TEETH OF A MOTOR-TRUCK-DRIVE WORM WHEEL

mitting the former to adapt themselves to the journal surface. The softer crystals under proper service conditions will wear slightly below the surface of the harder crystals. In order to retain these conditions it is necessary to preserve the natural crystallization upon the bearing surfaces, but this cannot obtain where they have been mutilated by improper tooling. This mutilation of the bearing surfaces gives rise to the crushing of the harder crystals and embeds these crushed particles into a compressed material which does not function as a normal bearing alloy.

[1] Presented at the Spring Meeting, June 1919.

15 A very forcible illustration of this is furnished by the bronze worm wheel of a certain motor-truck drive in which the teeth had been finished with a dull hob, Fig 8, showing a cross-section of one of the teeth. After giving unsatisfactory service the worm wheel was examined in the usual way by chemical and physical tests, neither of which showed any defect whatsoever. On the other hand, microscopic examination showed that the trouble was due to improper tooling. Fig. 9 shows a photomicrographic section of a field at *a*, Fig. 8. This view clearly shows that the natural crystallization of the greater part of the area has been disturbed, and that the edge of the tooth has had a cold-rolled or wire-drawn effect produced upon it by improper tooling. It is true this effect is not very deep, nevertheless it is on the very surface which is brought

FIG. 9 PHOTOMICROGRAPH OF FIELD AT *a*, FIG. 8 (60 MAGNIFICATIONS), SHOWING DISTURBANCE OF CRYSTALLINE STRUCTURE DUE TO THE USE OF A DULL HOB IN TOOLING

into play when the worm wheel is put in operation. Fig. 10 shows a part of Fig. 9 more highly magnified, in which the cold-rolled effect upon the surface is seen to be even more complete, a condition which proved to be the sole cause for the very unsatisfactory performance which this wheel gave in service. Fig. 11 shows a magnification of part of the edge *b* of Fig. 8, the inner edge or surface of this wheel from which it is centered accurately to within a thousandth of an inch upon its spider. This surface had been tooled in a horizontal boring mill with a single cut, the tool set so as to produce a smooth finished surface. The cutting of this surface, however, was done with a tool that was sharp and had sufficient rake, showing that it is an easy matter to cut a bronze surface satisfactorily without distorting the natural orientation of its crystalline structure.

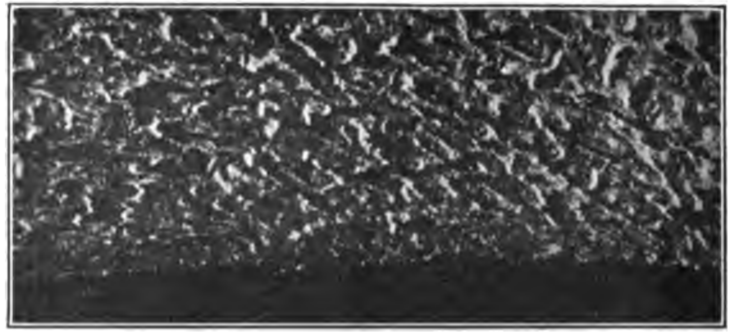


Fig. 10 Portion of Fig. 9 More Highly Magnified (120 Magnifications), Showing Clearly the Cold-Rolled Effect on the Bearing Surface of the Tooth

Fig. 11 Photomicrograph of Edge *b*, Fig. 8 (90 Magnifications), Showing that a Sharp Tool with Sufficient Rake Will Not Disturb the Crystalline Structure

16 Figs. 12 and 13 are photomicrographic sections of the inner edges of a bronze bushing which had been finished or tooled by broaching with a dull broach. These micrographs distinctly show

FIGS. 12 AND 13 PHOTOMICROGRAPHS OF THE INNER EDGE OF A BUSHING FINISHED WITH A DULL BROACH (60 MAGNIFICATIONS), SHOWING MUTILATION OF THE SURFACE MATERIAL

what not to do; they show that the surface material has been distinctly mutilated and that the compression of the material upon the bearing surface is very uneven, and they prove conclusively that this treatment is not conducive to best service requirements.

DISCUSSION

SELBY HAAR (written). Bearings for small alternating-current motors of the induction type must be manufactured with additional limitations in mind to those mentioned in the author's paper. This is mainly due to the small clearance or radial air gap between stator and rotor. The small motor which has a solid bearing usually has a small air gap, some being from 15 to 20 mils. A bearing for such a motor with a looseness of a few mils only would cause a considerable percentage variation of air gap around the circumference of the rotor, which variation gives rise to relatively large magnetic forces pressing against the bearings and tending to wear them elliptical. Since the maximum allowable wear is small, reasonable bearing life requires that the fit of new bearings be close. The close working limits may be illustrated by an actual occurrence. A motor with a shaft approximately 2½ in. in diameter was built for direct coupling to its load and the shaft extension was finished large for a coupling press-fit. When the bearing, of the same diameter as the normal shaft, was forced over the oversize shaft extension, it was expanded so that it became too large.

Another factor affecting bearing clearances, bearing pressures and thickness of oil films is the bending of the motor shaft if there is a large belt pull or gear thrust to be withstood. A journal of a shaft loaded as described does not bear evenly over the whole surface of the bearing and therefore special allowances are necessary. Usually the exact directions of all the forces acting on the motor bearing cannot be foretold at the time the motor is manufactured, and in many cases motors are shifted from one class of work to another so that it is necessary that motor bearings be more liberally designed than plain machine bearings.

Frank L. Fairbanks (written). The principles set forth in the author's paper are theoretically correct, but commercially it is impossible to obtain these conditions with the present wages, attitude and ability of so-called skilled labor.

To maintain a full oil film between journal and bearing, thereby eliminating all friction except internal molecular friction of the oil, with no metallic friction due to metal contact at any point, is perfection itself. If this were obtainable in practice, it would make little or no difference what metals were used, in as much as they would never come in contact. This, however, would mean a projected area so large and a rubbing speed so low that it is doubtful if the machine could compete with one obtaining 750 to 1000 lb. per sq. in. by using proper metals, and letting the metal take some of the wear and tear.

Aside from the many practical reasons why the author's theory cannot be carried out, there are only a few minor details which call for criticism. For instance, in Par. 7 he says "the two half bearings should be brass bound; that is, they should bear solidly against each other and should exert a pressure against each other somewhat in excess of the maximum crank-pin load."

If this were done, especially with large reciprocating engines, a week would be required to go over the bearings of such a machine, whereas with the boxes planed "open" and with a micro-wedge take-up, a 1000-ton compressor can be shut down, keyed up and started up within an hour. This is true more particularly of small high-speed engines, especially in hotels and apartment houses, that have to be keyed up once or twice a week to prevent objectionable pounding. On the other hand, if a complete film of oil between pin and boxes could be maintained, there would never be any wear, and consequently they would never have to be keyed up.

George N. Van Derhoef said that the author's paper had made clear many of the points he had wished to emphasize, and that many bearing difficulties had been clarified, but he wished particularly to call attention to the effect of the stiffness of the shaft in the journal. He had found it almost impossible, in many cases, to maintain a proper oil film in such a bearing as, for instance, that of a clutch pulley sleeve, because of the deflection of the shaft within the limits of the sleeve bearing. It was a question of providing a much larger shaft; or of reducing the length of the bearing surface, which would result in a much greater apparent unit pressure, but actually in a lower one. He had in one case calculated such a deflection and found it to be 0.006 in. within the limits of the length of the bearing.

Arthur Falkenau spoke particularly of the poor design represented in Fig. 4 of the author's paper, and said that he had come to the conclusion that such designs failed because of non-uniformly distributed pressures.

C. H. Norton said that oil grooves were of no practical value — in fact were a detriment to the bearing and that heat was of no possible consequence — that it usually is produced by the action of the oil itself and not by the friction of the metals together. Therefore heat up to 200 degrees could do no possible harm. The heat occasioned by the absence of oil increases rapidly and the bearing would stick before very much time had elapsed anyway, but the heat that allowed the bearing to continue hour after hour was not caused by a lack of oil but by the oil itself and that the greater the viscosity the more heat would be generated in any good bearing; that the bearing would run cooler as the oil viscosity becomes less. He called attention to some experiments conducted by the Texas Oil Company the results of which were published in pamphlet form.

The Author, in answer to a question by H. M. Bunting, referred to Pars. 14, 15 and 16. Experience, in general, he wrote, has fully established that the crushing, compressing or cold-rolling of the natural crystalline formation of a bearing alloy upon a bearing surface is injurious in both the bronzes and the babitts.

Replying to Mr. Van Derhoef and the latter part of Mr. Haar's comments the author would say that where the punishment on a bearing is such that the journal itself is bent or distorted it is impossible to obtain satisfactory service; the only remedy in such cases is that of increasing the diameter or stiffness of the journal. The amount of actual bending of the journal permissible within the limits of a bearing is very slight, it should be but a small fraction of the thickness of the supporting oil film.

After many years of experimenting Mr. Norton has become enthusiastic over the oil film; he now fully appreciates that he can run his grinding wheel-spindles upon a film of oil while doing accurate work.

In the first part of Mr. Haar's remarks he seems to have missed entirely the intent of the paper, namely: "bushing should be driven in with just enough pressure to prevent looseness during operation." It was the experience of a large manufacturer with this very type of induction motors that first led the writer to see the fallacy of an excessive drive fit, and years of subsequent experience has proven the wisdom of avoiding the same.

Primarily the paper was not intended to be theoretical, rather the contrary, in that its title might have been "A bearing trouble man's experience of twenty-one years." All the illustrations given have proven, in each case, to be an economically practical remedy for overcoming bearing troubles on an extended scale. All the progress that has been made on this subject of bearings has been by approaching the theoretical nearer and nearer, that is, by having more and more of the load carried upon the oil film. In many cases, what appeared theoretical years ago, today is plain "horse sense." There must always be more or less metallic contact between bearing surfaces; during stopping, starting and "running in" of the same it is unavoidable; therefore, the subject of bearing metals must remain a matter of importance.

Fortunately, Mr. Fairbanks agrees that the principles in the paper are theoretically correct. It, therefore, becomes simply a question of how much theory he can realize, since a practice that requires keying up once or twice a week is certainly in such a condition where improvement is both desirable and possible.

No. 1721 a

THE CAUSES OF INDUSTRIAL UNREST AND THE REMEDY

By Frederick P. Fish,[1] Boston, Mass.
Non-Member

THE attitude of the workmen throughout the world is such that individually and as a whole their efficiency and productive power are much less than before the war. While in other parts of the world they are working under unfavorable conditions, that is not the case in this country. Here the influence of the labor unions, not only in the "closed shop," where with the acquiescence of the employer they largely dominate the situation, but in many of the so-called "open shops," in which are labor-union workmen influenced by the principles of the labor unions, is such as to promote inefficiency on the part of the workmen.

I believe the fundamental difficulty to be that industry has not had time to adjust itself to the extraordinary new conditions that have prevailed during the past forty or fifty years. Those who were in control of the industries and directed their progress through these years of development and upon whom was the duty of assuring satisfactory employment relations, were so absorbed in the task of meeting the material advances in the arts that they overlooked the matter of properly adjusting employment relations to new conditions

Through extensive growth, personal touch between employer and employee became supplanted by organization which failed to adjust itself to the new conditions. We now see clearly that enough attention was not given to the conditions under which men worked, to the principles upon which their compensation should be determined, to their health and comfort not only in the factory but at home.

It was but natural that labor unions should succeed in bringing many workmen into labor organizations which seemed to them to

[1] Chairman, National Industrial Conference Board. Presented at the Annual Meeting, New York, December 1919, of The American Society of Mechanical Engineers. For complete address, see Mechanical Engineering for January 1920.

promise relief from what they regarded as unsatisfactory working conditions. There is no doubt that labor organizations have accomplished much good. Among other things they have focused the attention of industrial management and of the public upon those features of employment relations which needed study and revision.

But from the beginning the labor unions have adopted social and economic principles that were unsound and methods of propaganda and of action that should not be tolerated in any civilized society. They have worked for the suppression of production upon the false theory that the less men worked, the more men could be employed.

There can be no specific general cure, no panacea for the evils that exist, either by way of legislation or of reorganization of our social or industrial methods; but the present unfortunate antagonism between employers and employees and the existing disturbance of the cordial feeling that should exist between them can be eliminated and right conditions restored only by definite personal effort to develop and maintain in the industrial establishments of the country, suitable employment relations, based in each instance upon fair, cordial and sympathetic coöperation and a recognition of what is right and proper in the particular establishment.

No hard and fast rules can be laid down. Each employer should study his own problem and attempt to deal with it in view of the conditions in his own establishment, exactly as he works out his mechanical and technical problems in harmony with the requirements of his own work and with only general reliance upon the experience of others.

Management can never be asked to sacrifice its function of judgment and direction. The workmen would be the very first to recognize that ultimate control must be with the employer. Naturally, however, they cannot be satisfied if no opportunity is given them to state their point of view as to matters which are of the utmost importance to them in their life's work.

It is obvious that such relations as should exist between management and workmen are only possible in an establishment which is not tied up with the trade unions. There alone are the workmen free from outside influence which may be and often is used to stimulate antagonism and to breed dissatisfaction as well as to reduce efficiency.

It is the duty of each member of this Society to study the question of industrial relations and to take a definite position as to what seems to be the single important issue we are now facing.

No. 1721 b

SYSTEMS FOR MUTUAL CONTROL OF INDUSTRY

By William L. Leiserson,[1] Rochester, N. Y.
Non-Member

"It is difficult for us to change our conception of those conditions and relations as we have learned to know them from years of experience," said Dr. Leiserson in introducing his discussion of shops committees, works councils and the coöperative or industrial democracy plans by which employers propose to give their employees some voice in the control of industry, as contrasted with trade-union collective bargaining by which organized wage earners attempt to force employers into a system of joint control of wages and working conditions.

He pointed out that the world war and the shortage of labor since the signing of the armistice have materially changed the status of the employee, and that however much the employer may realize the changed industrial conditions, he finds it difficult to conceive of his employees as having a voice in any action that may affect them.

"To talk about mutual or democratic control of industry or to establish plans for coöperative or industrial democracy without a clear realization of the revolution that has taken place in the status of the wage earner can lead only to confusion and to failure of the plans for mutual control of industry," Dr. Leiserson asserted. He compared the present situation with that in Europe when an industrial revolution was brought about at the time of the Black Plague resulting from very much the same industrial ills as those from which we now suffer. He traced the history of the labor movement in this country and in others and continued: "In a democratic country with rising standards of living and large numbers in what we call the middle classes there is no great danger of labor completely dominating the situation and becoming the only masters of industry.

---

[1] Chairman, Labor Adjustment Board. Presented at the Annual Meeting, New York, December 1919, of The American Society of Mechanical Engineers. For more complete account see Mechanical Engineering for January 1920.

[illegible] of law [illegible] industrial

[illegible] authority [illegible] have in industry [illegible] democratic control [illegible] To maintain law, order [illegible] western countries there [illegible] consent of the governed. Without [illegible] democracy, our industrial governors [illegible] to control their organizations, to have their [illegible] and order obeyed. This is the essence of the industrial problem today."

No. 1721 c

WHAT MAY WE EXPECT OF PROFIT SHARING IN INDUSTRY?

By Ralph E. Heilman,[1] Chicago, Ill.
Non-Member

THAT the subject of profit sharing is not one which is new to this country but dates back to the early 70's, when there was a very widespread interest in the whole subject of profit sharing, was pointed out by Dean Heilman. Many experiments were introduced at that time and profit sharing was heralded as a panacea for industrial problems. Most of those experiments were soon abandoned however, and profit sharing came to be regarded as a subject of interest mainly to the social reformer or theorist and of little interest to the business man. In the last few years there has been a marked increase in the attention paid to the subject, which justifies a reexamination of the profit-sharing plan in the light of modern business practice and methods.

"Profit sharing," said Dean Heilman, "is an agreement between an employer and his employees whereby the latter participate in some way in the profits of the business, either of the business as a whole or of some unit part of the business."

The speaker than summed up the reasons for introducing profit sharing and the advantages its introduction will bring to the firm. First, it is claimed that profit sharing will increase efficiency and output. "But only to a certain extent is this true, and only under certain circumstances and subject to special limitations; in other words, generally speaking, more effective results are obtained from labor if compensation is on the basis of the measurable results of each individual so that he will be induced to put forth his best effort."

Secondly, profit sharing may be introduced for the primary purpose of promoting permanency and stability among the employed

[1] Dean, School of Commerce, Northwestern University. Presented at the Annual Meeting, New York, December 1919, of The American Society of Mechanical Engineers. For a more complete abstract, see Mechanical Engineering, for January 1920.

staff and as a consequence reduce the labor turnover. Such a scheme contains one of two features, either there is a preliminary period which the employee is required to serve before he can participate in the profits, or profits are to be issued only at stated periods, with a proviso that employees leaving between these periods surrender their right to participate in the profits.

"Again, profit sharing is sometimes introduced with the hope that it will promote industrial peace. Profit sharing has possibilities in this direction, but they have been exaggerated."

The most important purpose for which profit sharing has been introduced, in the opinion of Dean Heilman, is to promote effective management upon the part of managerial executives, semi-executives, junior executives, and salaried employees all the way from foremen or sub-foremen up.

Speaking of the relations of the effectiveness of profit sharing to the rank or responsibility of the participant, he said: "In the first place, the higher the rank the greater is the opportunity which the individual has to exercise an important influence upon the profits. In the second place, the greater the responsibility the easier it is for the employee to see the relation of his efforts to the profits of the business. In the third place, the managerial or executive group are more familiar with business vicissitudes and uncertainties than the rank and file. This has an important effect during a lean year."

In the opinion of Dean Heilman, profit sharing will operate successfully for the rank and file of wage earners only when applied in a small or medium-sized shop, because the larger the group of participants, the more difficult it is to educate them in the essentials of profit sharing, and the larger the group, the less effect each one can exercise on profits.

"Profit sharing is not a social panacea. It will not solve the vexing problems which arise between employer and employee. It will not prove a substitute for good management on the part of a corporation, nor will it prove a substitute for the wage system. But within the wage system it has a place, a real place. In my judgment, we are destined to see a much wider and a much wiser use of profit sharing than in the past."

No. 1721 d

WAGE PAYMENT

By A. L. De Leeuw, New York, N. Y.

Member of the Society

This paper calls attention to the fact that the main terms in controversies between capital and labor are ill defined or not defined at all. The fact is emphasized that there is at present no equitable way of determining wages, and, as a result, that there is no possibility of collective bargaining. Wages at the present time are paid for time given, whereas they should be paid for work produced.

Employers alone cannot correct this condition, but employers and organized labor together, assisted by qualified engineers, might go far in correcting this unsatisfactory condition of wage payment.

UNREST among the laboring classes has taken on such proportions and is so widespread that it is timely to discuss the various items which enter into the present relations between employer and employee, even if no prescription for a cure can be given at the present time.

2 It is perhaps a little out of date to speak of differences between "employer" and "employee." In the first place, these terms are gradually being looked upon as the equivalents of slaveholder and slave, though there was a time when such was not the case. In the second place, it is beginning to be realized that there is really a third party, the general public. The present strife has a number of angles, of which one or more present themselves in individual cases. Quite often an honest attempt is made to reconcile the varying viewpoints of employer and employee, but more often it is purely and simply a question of war, of brute force; and a peace of victory, not of understanding.

MISUNDERSTANDINGS IN REGARD TO TERMS EMPLOYED

3 Many times one or both parties in the conflict try to befuddle the third party, the general public, by statements which are

[1] Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

misleading or which are couched in such words as will hide that which is not true.

4 It is well recognized that where two parties have a difference of opinion the first step should be to find some common ground from which to start, and that then the second step should be to define clearly and precisely the chief terms which enter into the controversy. It is never difficult to find the common ground between employer and employee. When the railroad men strike or threaten to strike for higher wages or shorter hours or both, and when the railroad managements refuse to concede these demands, it may seem to the casual observer that there is no common ground; and yet this common ground is unmistakably at hand and is wide enough for all to stand on. The common ground in this case is the desire on the part of both employer and employees to have the railroads running. Without this the employer cannot get his profit and the employee cannot get his wage; and as this common ground is a matter of intense interest to both parties, there should be no trouble in starting the discussion under very favorable circumstances.

5 One reason why such discussions so often fail to bring the desired results and why they are often not started at all is that there is no clear understanding of some of the most commonly used terms in these controversies. The writer wishes to call attention to some of these terms, and to some of the misunderstandings in regard to their meaning.

6 We read practically every day about war between "capital" and "labor," about the demand of the laboring man for the "right to organize," and of his right to insist on "collective bargaining," and yet these three terms, which are perhaps used in every controversy that comes up, are not only not defined but they are actually misleading in themselves.

WAR BETWEEN CAPITAL AND LABOR

7 If war between capital and labor means anything at all, it must mean war between capitalists and laboring men. Capital is the result of past labor used progressively. Without capital, civilization is unthinkable. Capital is the tool with which the laboring man works. There can no more be war between labor and capital than between the laboring man and his tools. It may be said that his is merely a fine distinction and not a difference, but the writer cannot take this view. Recent events in Russia have shown that

there are a large number of people who actually believe and act upon the idea that capital must be destroyed in order to give the laboring man his own. It may be that the leaders who presented this thought did not have capital in mind, but were either referring to capitalists or to the present system of control of capital. But their slipshod way of expressing themselves, or the lack of existing definition, causes a large number of people to think that *capital itself* is at fault.

8 In late years the control of capital has been more and more centralized, or, in other words, in the hands of fewer individuals. And though it is well recognized that this condition has its advantages in many ways, yet the dangers to which such centralized control may well lead have caused it to be regarded with suspicion and enmity. Capital being the tool with which labor must work, it is but natural that labor should look with suspicion on an attempt to corner the tool supply. For this reason we may naturally expect that labor will insist upon a better control of their tools, capital. They cannot have war with capital, and there is no reason why they should make war against capitalists in general, for it need not be pointed out that many capitalists are good and honorable men who do not deserve the enmity of any one. On the other hand, it may well be conceded that labor has cause to dislike, or even hate, the present system of control of capital.

9 It has been pointed out many times quite recently that there are really three parties, not considering the general public, which have a right to be heard in this controversy, namely, capital, labor, and management. It seems to the author that this merely leads to complication without any compensating features. Management and labor are both labor, but of different kinds. If at times they clash it is not due to unavoidable conditions but merely to defects in one or the other. We find this clash between management and labor whenever an apprentice boy works under the instructions of a full-fledged mechanic; we find it between the mason and the hod carrier; we find it between the plumber and his helper; it does not touch the broad question but is only one of the phases of one of the details.

THE RIGHT TO ORGANIZE

10 Another term which should be quite familiar to all of us by this time is the "right to organize." Whether this term was purposely created by labor organizations as a slogan which ought to

appeal to every fair-minded man, or whether it has just naturally slipped in, the writer does not know nor does he care. What he does care about is to have this term defined in such a way that no one with a fair amount of intelligence can mistake it for something else.

11 The writer does not believe that there is an appreciable percentage of employers who would deny their employees the right to organize; for instance, for a baseball team, or a brass band, or sick benefits, or coöperative stores or a Bible class. On the other hand, nobody would blame the employers for denying their employees the right to organize for some other purpose; such, for instance, as burning down the plant, or sabotage in the shops, or blowing up the works of his competitors — however beneficial this might be to him in a business way, or for many other purposes.

12 It follows, then, that the "right to organize" must be further defined or at least limited in its scope before we can judge as to the real right of the employee to organize. If the purpose for which the employee wishes to organize is legal and proper, the third party, the general public, will naturally concede this right and quite as naturally inquire as to whether the employer has really denied this right to his employee.

13 Now, what are actually the facts in most cases where this right to organize has been brought to the foreground, are these: A labor organization thinks it proper that the employees of a certain establishment should be gathered into its union. This, by the way, is perfectly proper, and no objection could be taken to it. The organization succeeds in gathering in part of the employees. The employer, fearing that this activity may lead to a strike, attempts to keep the other employees from joining the union; in other words, he tries to organize them with him instead of with the union. This also is perfectly proper. To make the claim at this point that the employee has been denied the right to organize may be good politics but is not quite correct. It is doubtful whether in late years there have been many cases where the employer has denied the employee the right to join the union, even if he was not in favor of it. When undesirable conditions or relations existed between employer and employee, or perhaps when union activities led to a strike, it was quite customary to make a demand on the part of the men for the right to organize — yet this right had seldom been denied; but put in this way the general public would get the impression that all kinds of organizations of the employee

are taboo. What is really meant by the "right to organize" is that the employer shall give, not to *his men*, but to *some labor union*, the right to come into his shop and organize his men into a union; and apart from the question as to whether this is desirable or equitable, it is certainly different from the idea which the expression "the right to organize" conveys.

COLLECTIVE BARGAINING

14 Another term which is misleading, even more so than those mentioned before, is "collective bargaining." So far collective bargaining has been collective, it must be admitted, but it has not been "bargaining." In collective bargaining it is supposed that some or all of the employees have delegated the right to bargain for them to some attorney or business agent. In practically all cases this business agent is their union, and this union, according to the term used, is supposed to bargain with the employer. As a matter of fact, however, there is no bargaining, and, under the present conditions, there can be no bargaining. There have been cases where the men were entitled to all they asked for, and possibly more; there have been cases where the men were not entitled to as much as they asked for, but the method of bargaining has never been employed, and, in the writer's opinion, *could not* have been employed, to settle the differences. The issue has been brought to a conclusion by strikes, or threats of strikes, which is no more a method of bargaining that when a man points a gun at his debtor to collect a bill. The fact that the bill may be due to him, and even overdue, does not make this a method of bargaining.

15 There is still another point about this question of collective bargaining which should have all the light possible thrown on it. Is collective bargaining, bargaining by *all* the employees? Is it bargaining for a group or part of a group? If it is bargaining by a union, acting for a group, should the representative, the union, show credentials before bargaining starts? Such a course of action might reveal the weakness of the union or of the employer, but would it not be for the best interests of the third party, the general public?

16 When the writer stated that under the present conditions bargaining is not possible, he had in mind the essential feature of bargaining, which is a process of readjustment of the value of the object of barter in the minds of the two parties concerned. If a house is offered for sale and a price of five thousand dollars is asked and an offer is made of four thousand dollars, then it cannot be said

that any bargaining has taken place so far. But if now the seller readjusts his ideas in regard to the value of the house or the desirability of receiving money; or if the buyer readjusts his ideas as to the value of the house or his ability to pay money, then true bargaining takes place, even if, in the end, neither of the two gives in. In that case there has been bargaining, but no results were obtained. In order, then, that there shall be true bargaining, there must be an object to be sold, an object to be given in barter therefor, and an attempt of buyer and seller to modify each other's conception as to the value of what they are offering.

17 Now, in offering labor and demanding money, one of the principal requirements of a bargain is missing. When we say that the price of eggs is seventy-five cents, we mean, per dozen. When we say that the price of land is two hundred dollars, we specify, per acre. But when we say that the price of labor is seventy-five cents, we do not know what this means. There are measures or quantities by which to sell eggs or real estate. If a house is sold, though there may not be a measure of the value of the house, there is at least a chance for the prospective buyer to look it over and make his own estimate, or if you wish, guess, as to what the house is worth. But when it comes to buying labor we have no measure nor can we estimate its value by personal observation — except after the labor is done, which is long after the so-called bargaining took place, long after the price was set. It is true that labor is sold by the hour, but the hour is a measure of time, not of labor. We might just as well sell eggs by the yard, and without even specifying whether we lay them lengthwise or crosswise.

18 At the present time labor is not sold by measure but a workman sells his time. *This inability to strike a bargain because the value of the product to be sold is entirely unknown, is, in the writer's opinion, one of the greatest difficulties to be overcome before there can be an equitable adjustment of the differences between employer and employee.* The fact that labor is sold by the hour is equally unjust to both parties. Sometimes the employer suffers, sometimes the employee. In all cases it is a source of curtailment of production, and this leads at once to a discussion of wage system.

WHAT WAGES ARE PAID FOR

19 "Wage" is defined as remuneration for work done. It is also defined as remuneration for time given. The second definition seems to be defective. In the first place, when money is paid

for time given it is called "salary." In the second place, and this objection is more serious, time alone is not bought or paid for. There are many occupations in which the employee must wait until something turns up for him to do, and then he is supposed to do it. Take the example of the stenographer, the watchman, the bridge tender, the crane tender, the machinist's helper. Such people may be idle for a long time, but finally there comes some occasion when they have to do work, and they are supposed to do it. It would not be proper, then, to say that they are paid for their time. It would be more proper to say that they are paid for the labor which they perform within a specified amount of time. Besides time and labor there enters into such cases the requirement of physical presence. When a stenographer is supposed to give eight hours' time, he or she is supposed to be at a given place within that time, say, at the office. That such people are not merely paid for their time is emphasized still further by the fact that they are not paid more when they move further away from, or less when they move closer to, their place of employment.

20 There are cases where, practically speaking, time alone is sold. Such is the case, for instance, with a stationary watchman. Even in such an extreme case something more than time is paid for: the man's watchfulness and faithfulness. In other cases time and labor are paid for: for instance, with a machinist's helper, or a blacksmith's helper. In still other cases time and skill are paid for: for instance, the man running a large planer or boring mill. The writer cannot think of any case where wages or salary are paid for time alone.

21 In the great majority of cases it is labor that is wanted, or rather the results of labor, and not time. Now, the great problem before us is to find a measure for this labor, in order to obtain an equitable way of paying for it. In even the simplest operations of labor there are so many factors which modify the result that it is difficult to establish a unit of measurement. Digging for foundations is a simple operation, and it should be possible to measure the result by the cubic yard if no disturbing factors had to be reckoned with. The amount of material removed per man per hour depends on the depth of the foundation, the width of the trench, the kind of material to be removed, the weather of the day, and even of the previous day, et cetera.

22 In some of the more complex work it has been possible to subdivide the operations to such an extent as to set a fixed time for

a given operation. Such is the case where single operations are done by an automatic machine to which the operator feeds the material, or where he operates a simple machine and performs simple operations such as drilling with a sensitive drill press, using jigs, etc. But even in such simple operations some factors enter which will cause trouble at times — trouble with the machine, breakages, replacement of tools and the like are such disturbing factors. However, in such work the disturbing factors are a small percentage of the total and can be estimated close enough to establish a fair value. It would be possible to extend the number of such operations very largely, but it would require the active coöperation of labor to do it successfully.

23 In a large portion of work there is a combination of time, physical presence, labor, skill, knowledge, judgment, and probably other factors which are hard to define, such as reliability, steadiness, enthusiasm, loyalty, ambition, and whatnot. To make up a formula which would embody all these elements and from which a man's value could be calculated is, of course, impossible; and yet, unless this can be done there is no possibility of avoiding differences of opinion between employer and employee as to the value of a man's labor. The writer has gone into this matter somewhat at length in order to make clear that a mathematical solution is not possible, and in order to realize that a compromise system *must* be developed.

SYSTEMS OF WAGE PAYMENT

24 There are various systems of payment in existence at the present day — there are straight wages, piece-work system, various systems of bonus payment, premium payment, and combinations of these systems.

25 The wage system considers nothing but time and physical presence. A man is selected for a certain task because he or some one else claims that he is fit for that task. If his work is satisfactory he is retained; if not, he is dismissed. Both employer and employee have made a guess. The employee knows exactly what he will get, and guesses at what he will have to deliver. The employer knows exactly what he must deliver, and guesses at what he will get. He has no means to bring the employee's output up to his conception of what it should be, unless the employee falls so far behind the employer's expectations that he makes a new guess and fires the man.

26 With the piece-work system the employer knows exactly what he will get for his money, but the employee makes a guess as to how much he can reasonably do; he has no control over the conditions which will enable him to do, or prevent him from doing, as much as he is expected to.

27 There are various kinds of bonus systems. The employee may earn his regular wages and get a bonus at stated intervals if his work exceeds the expectations of his employer; or he may be working under the task-and-bonus plan, in which case there is less guessing and more of an attempt at a definite bargain. If the employer is in earnest with such a system, and sees to it that it is possible for the employee to fulfill his task, then this system approaches quite closely to true bargaining.

28 The premium system is an attempt at gaining the interest of the employee by making him invent improvements in the method of working and sharing the profits with him; but unless the time originally set was carefully studied, and unless all conditions of machinery, tools, existing knowledge and related matters remain unchanged, this system also will lead to controversies and injustices.

29 To sum up, the pure wage system is no bargaining in any sense of the word; the piece-work system is only bargaining if employer and employee have reached an agreement, and if there is some mechanism by which this agreement can be changed as soon as conditions change; the pure bonus system is nothing else but a wage system with a kind of profit-sharing plan; the task-and-bonus system, if properly worked out and if accompanied by complete instructions to the employee, is a perfect bargain, but must be constantly revised as conditions change. However, the last mentioned system carries with it the necessary mechanism to effect these changes.

30 It will be seen from the foregoing that wherever bargaining is possible it must be done, not only by each employee individually, but by each man separately every time his work is changed; in other words, each new operation he performs should lead to a new bargain. Unfortunately, the number of operations which can be fully analyzed and for which complete instructions can be given is relatively small. There is no doubt but that it could be followed in many, many cases where it is not followed at the present. But, looking at the world's work in its entirety, the total percentage of the work which can be treated in this manner is small.

CONTRACT BASIS ANTAGONIZED BY ORGANIZED LABOR

31 Whatever attempt has been made up to date to place labor on a contract basis, on a basis of bargaining, has been done entirely by the employer, and has been antagonized by organized labor. And yet it is the writer's opinion that the ultimate solution of labor trouble must be based on some method of bargaining, probably collective.*

32 Labor unions are fighting organizations. The underlying principle of their conception was to find the necessary strength in numbers. The unions were the army with which the laboring man fought the employer. There was no other means of accomplishing the aims of the laboring man except to fight, and fight they did; and if fighting has given them, if not all, at least a large portion of what they desired, it is but natural that they should consider war the best means of reaching their ends. Being a war organization they had to consider the employer as the enemy, and as a consequence every act of the employer was to them an act of the enemy, something intended to defeat the union and its aims. The unions, so far, have not offered any constructive suggestion, and this cannot be expected from a fighting organization. Furthermore, it is to the personal interest of union leaders to hold to this system. It may be taken for granted that there have been, and are, many unselfish union leaders, but this does not offset the general tendency of the system to perpetuate itself as a fighting organization, because this has in its turn a tendency to further the interests of the officers.

33 Another reason why the attempt of the employer to put labor on a contract basis has not been well received by the labor unions, lies in the historical fact that on account of occasional scarcity of work it became one of the principles of the union to take measures to insure that whatever work was available should be sufficient to keep the union men in employment. The three chief means used to accomplish this were: first, to prevent anybody but union men from working at the trade; second, to limit the number of apprentices, or in general, newcomers in the union; third, to limit the output per man. The second item has not been strictly adhered to because it was found that by limiting the number of men in the union the number of men outside the union was increased, thus decreasing the relative fighting strength of the union. The first item has been strictly adhered to, but has often been denied by the union, probably for the purpose of satisfying public opinion.

The third item is still generally adhered to, but is mostly camouflaged. Whether the unions should be condemned or praised for the attitude they have taken, is a matter of standpoint. If the true status of the union is that of a fighting organization, then no objection can be taken to these methods. We should not expect war without guns or accidents. Whether the interests of the world at large are best served by having the unions act as fighting organizations is more than doubtful, and the writer for one believes that it is time for the unions to forget some of the past, turn over a new leaf, and prepare for a future in which there shall be peace, progress and production.

HOW AIMS OF ORGANIZED LABOR CAN BE ATTAINED

34 Taking the aims of organized labor to be: —

1 Proper share of the proceeds of labor
2 Reasonable working conditions and working hours
3 Right to organize
4 Collective bargaining,

the writer believes that these aims could all be accomplished without strife if the last item, collective bargaining, were put into actual practice instead of being a mere catchword.

35 If employers and employees together would put as the first item in their catechism the truth that the world must produce more in order to have more; if employer and employee together would try to find an equitable way of estimating the value of work done; if both would subscribe to the truth that no permanent gain can be made by wearing out a man's capacity for work nor by allowing him to work less than he should for the good of the world at large; if then employers and employees together would organize a bureau of research for defining the conditions under which various classes of work should take place; and if finally an attempt were made by both employers and employees to classify men according to their natural or acquired ability, there would be very little reason left why a union should be a fighting organization.

36 If the writer were the owner of a machine shop and the business agent of a machinist's union should come to him and say that his union does not admit members who are not bona-fide machinists, who are not sober and industrious men, and that the union would see to it that the men would treat the management in a fair and equitable manner, but that, on the other hand, it would want a share in the management in so far as working conditions, wages,

working hours, et cetera, are concerned, and that it would not permit anybody except members of that union to work in that shop, the writer would probably fall around that business agent's neck and hail him as a savior.

ENGINEERS MUST ASSIST LABOR IN ESTABLISHING SATISFACTORY STANDARDS OF VALUE FOR WORK DONE

37 The foregoing may look like a picture of Dreamland. The writer realizes that such conditions cannot be brought about at once, nor in a very short time, but he believes that they *can* be brought about and that now is the time to start the preliminary work. Up to the present time the unions have acted entirely through their business agents, who sometimes had, and more often had not, a clear idea of the problems involved in the trade they represented. Whatever knowledge they might have had was not permitted to come to the foreground. One does not make allowances for the enemy's good points during war. It seems to the writer that the time has come for the unions to take the first step along the lines of considering union activities as a legitimate business, and legitimate business cannot shut its eyes to facts, however disagreeable to contemplate they may be. It is his belief that the crux of the solution lies in the establishment of a proper wage system, and that in order to establish such a system the engineer must come to the assistance of labor in order to find standards of value for work done. Though it is not likely that there will ever be a time when every human activity can be scheduled and analyzed, yet by far the greater part of industrial operations can be treated in this way. The relatively few operations which cannot be classified and treated in this manner would be such a small part of the total that it would be easy to find a way to compromise when such exceptions occurred.

38 In estimating the value of operations we must drop to a large extent the idea that wage is the compensation for time. It should be made a compensation for product delivered. The value of the product changes constantly. Changes may be due to the desirability of the product, or to the means employed to produce the product, or to the law of supply and demand, or to other causes. In other words, the relations existing between various products are ever changing; consequently, whatever estimate is placed on the value of the product of labor should be changed from time to time,

so as never to have too large a difference between the actual and the estimated value.

THE ESTABLISHMENT OF STANDARDS A GIGANTIC TASK

39 It hardly need be pointed out that to set such standards of value of work must necessarily be a gigantic task. In addition to the many technical difficulties in estimating values of products, there are other elements which must not be forgotten. There are many cases where operations are required which call for extreme skill, possessed by only a few, a skill which can never be found in the mass of the people. As a rule, people possessing such skill love to employ their gifts, they love their work and are willing to produce for less than their product is worth. Such skill should be estimated at its proper value, not only as a matter of justice to the workman, but also because he may finally lose his enthusiasm and industry at large would suffer.

40 There are operations where steady nerves are essential, some where physical courage is required, others which cannot be successfully carried out without many years of experience, et cetera. Such points should all be considered. The value of driving a rivet in some dangerous place of a new skyscraper is higher than that of driving a rivet in a boiler shop. The ability to judge the temperature of a piece of steel by its color is only obtained by many years of experience. This ability could not be called skill, though it is closely related to it.

41 Consideration should also be given to what constitutes a proper minimum and a proper maximum wage. In repetition operations it is sometimes possible for the operator to produce large amounts of work but only at the expense of extreme weariness, and in the long run such a man is not employed to the best advantage of the world at large. Recognition of this fact has led to the prescription of rest periods. On the other hand, there may be cases where it is expedient to employ men not skilled in the operation they are supposed to do. This may happen because skilled men are not available at a given time or place, or because special conditions call for an unusual amount of this class of work. Though the value of the product of such men might be low, they should receive not less than a minimum wage. This minimum wage, however, is in itself difficult to determine. A satisfactory minimum wage for an unmarried young man is not at all satisfactory for a married man with a large family.

RESULTS TO BE EXPECTED FROM A SCIENTIFIC WAGE SYSTEM

42 It is really superfluous to point out the many disturbing factors acting against the establishment of wages in proportion to product. What the writer wishes to emphasize is his belief that, notwithstanding all the difficulties that have been mentioned and the great many other difficulties not mentioned at all, it will be found possible to classify a large portion of the work of the world in such a manner that wages can be set to such a degree of scientific accuracy that the variations caused by the disturbing facts will not be so large but that they will lend themselves to compromise. It is further his belief that such a classification cannot be accomplished by the employers alone, nor by the laboring men alone, but that there should be an attempt at a concerted effort of employers and employees to bring about such a classification. He believes further that when these two classes are working together for a constructive purpose they will find so many things in common that they will be more apt to forget their differences. Finally, he wishes to state once more this belief: That the real cause of the present-day unrest lies in the fact that there is no unit of measurement which both employers and employees can use; or, in other words, the fact that our present wage system is not based on knowledge and justice, but only on guesswork and on the fear that the one may "do" the other.

No. 1722

THE FUTURE OF AVIATION

By Col. E. A. Deeds, Dayton, O.
Member of the Society

"THE war itself gave a tremendous impetus to aviation," said Colonel Deeds, "and when we get far enough removed from it so that the public will be able to discern between the patriotism and performance on one side, and propaganda and politics on the other side there will be found a real record of achievement, more potent and more inspiring, probably, than anything the future can give us, at least for some time to come."

He spoke then of the work of developing an organization which was to produce such a tremendous number of planes, balloons, engines and accessories as were produced during the war, and of providing the necessary supplies which the aircraft program involved.

Turning his attention to the work of the enlisted personnel, he gave the record of the achievements of the boys themselves, both in the training camps of this country and the battlefields of Europe.

Looking to the future the speaker predicted that the line of development will take many forms, mentioning the armament, both offensive and defensive, of planes, the ability to reach great altitudes, higher speeds, the use of the aerial torpedo, wireless control, and the effects which all of these developments will have on military tactics.

Speaking of commercial aviation Colonel Deeds said that the risks were too great and expenses too high at present. Government aid must be injected if aviation is to be a commercial success. He also pointed out the need for landing fields, beacon lights or wireless warnings, instruments for aiding navigation, aerial maps and the investigation of meteorological conditions of the atmosphere.

Colonel Deeds believes in the future of the lighter than air

Abstract of an address delivered at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers. A more complete report of the address will be found in Mechanical Engineering for January 1920.

THE history of water-wheel development as known today is practically covered by the developments of the last twenty or thirty years. This means that it has gone hand in hand with electrical development which alone has made possible the utilization of large capacity units at high speeds. In the matter of speed, electrical-generator development has permitted of higher limits of r.p.m. than hydraulic designers have been able to reach under low- or medium-head, large-capacity conditions. On this account and because the limit of efficiency has long been so nearly reached, the greatest endeavors of runner designers have been directed toward increasing speed and, as has been the case more or less with all lines of design, loyalty to precedent has largely interfered with any radical departures that might make for most rapid progress. There has been no radical improvement in efficiency in the entire history of modern hydraulic practice. On the other hand increases in speed characteristics may serve almost as a criterion of progress, and that such is the case will be seen by even a most superficial examination of Fig. 1.

CHARACTERISTIC SPEED

2 In order to bring out the significance of the subject-matter of this paper it is necessary to briefly cover the history of water-wheel development and with such a history available it only remains to determine upon some common basis of comparison to permit of contrasting the new work with prior development and point out the possibilities it presents. Nature was especially inconsiderate in not standardizing water falls, and as a result water-wheel designers have to meet a much larger variety of conditions than do the builders of any other form of prime mover. Comparisons are therefore difficult. In practice, water-wheel runners vary in dimensions, horsepower, speed, and the head under which they operate.

3 To permit of comparison between any two runners, or between the runners of any two periods of time, some common ground is necessary, some characteristic that can be expressed preferably in a single figure. Such a figure is available in what will be hereafter designated as *characteristic speed.* Because of its peculiar value in water-power work, though at the risk of repetition of more extensive articles already written, the original meaning and use of this term will be outlined briefly and explained by example.

4 To compare two water-wheel runners operating under different heads, developing different horsepowers and running at different speeds, the first step would logically be to compute their power and speed performance under the same head. To do so, however, would leave varying horsepower, speed, and diameter, and so the second step would be one of the following:

A Recompute and compare their powers on the basis of their being so changed in dimensions as to have the same speed

B Recompute and compare their speeds on the basis of the same power.

Either method would give a positive indication of character expressed by a single figure.

5 Basis *A* would give a *characteristic power* which serves to give the desired absolute basis of comparison, though on an exaggerated scale. This basis is used by Professor Zowski in *Engineering Record* of December 26, 1914, and as that article presents the most recently published data on high-speed runner development, diagrammatic comparisons given hereafter will be made on the same basis, using the same scales.

6 In general practice the speed comparison (*B*) is made, and to make universal comparison possible the common head basis is taken as 1 ft. and the common power basis as 1 hp. Professor Zowski used a common speed of 50 r.p.m. in place of unity as that value is about the average unit speed of runners built for the Holyoke test and hence incurs the least recomputation or readjustment of mental conception as to size. The speed in r.p.m. resulting from such recomputation is the characteristic speed and is the characteristic which is used generally in hydraulic-turbine practice, going under the above name in Marks' Mechanical Engineer's Handbook. The name "Specific Speed" as used by Dougherty, Taylor and others, is of European origin. "Type Characteristic" was the name given by Zowski. Often, however, merely "Runner Characteristic" is used. This characteristic may be defined as follows:

> The characteristic speed of a runner is the speed in r.p.m. which a model of that runner would have if operated under a head of 1 ft., this model to be reduced proportionally in all dimensions from the original until it will develop 1 horsepower under 1 ft. head.

7 Expressed by symbols, the foregoing statements, together

with the hydraulic laws governing the variation of power and speed of runners, are as follows:

a Under varying head a water-wheel runner maintains the same characteristics of efficiency if the speed varies as the square root of the head: i.e., as the water velocities, which vary as $\sqrt{2gH}$

b Under such conditions the power varies as the three-halves power of the head neglecting the fine points of friction, etc.

c The power of a runner under a given head varies as the square of its dimensions and its speed varies inversely as the dimensions.

These relations are fundamental and check out absolutely in practice and are the basis of the following:

Let HP = brake horsepower of a runner
N = revolutions per minute (r.p.m.)
H = head in feet
HP_1 = brake horsepower of a runner under 1 ft. head, designated unit horsepower
N_1 = r.p.m. or speed of a runner under 1 ft. head, designated unit speed
d = runner diameter in inches
N_c = characteristic speed.

d To permit of the basis of comparison outlined above we reduce power and speed to 1 ft. as follows:

$$HP_1 = \frac{HP}{H\sqrt{H}}$$

since the power of any runner varies both as the pressure with which the water is presented to it and as the quantity or velocity of the water, which depends on $\sqrt{H}$.

e
$$N_1 = \frac{N}{\sqrt{H}}$$

since any part of a runner should move at a speed which bears a constant relation to the velocity of the water or $\sqrt{H}$.

Since at constant head the power varies as the square of dimensions and speed varies inversely as dimensions, *f* HP varies as d^2, which varies as $\frac{1}{N^2}$, and

g N varies as $\frac{1}{\sqrt{HP}}$

To reduce the capacity of a runner to unity so as to permit of direct comparison of speed, we multiply its power of HP_1 by $1/HP_1$, which means that its dimensions are multiplied and its speed is divided by $1/\sqrt{HP_1}$ (The diameter so obtained might be designated its characteristic diameter, d_c), whence

$$N_c = N_1\sqrt{HP_1} = \frac{N}{\sqrt{H}}\sqrt{\frac{HP}{H\sqrt{H}}} = \frac{N\sqrt{HP}}{H^{\frac{5}{4}}}$$

Care should be taken to see that the horsepower figure used is for

TABLE 1 COMPARISON OF OLD- AND MODERN-TYPE HYDRAULIC TURBINES

Item	Old Types		Modern Types			
	Overshot Wheel	Fourneyron (Tremont) Turbine	Nagler	Usual Mixed-Flow or Francis		Impulse (Pelton)
			High Speed Low Head	Medium Speed	Low Speed Dbl. Run.	Twin
Head in feet, H	14	14	14	200	400	2,000
Horsepower, total	50	180	500	40,000	20,000	20,000
Runner horsepower, HP	50	180	500	40,000	10,000	10,000
Speed, r.p.m., N	10	53	200	150	360	375
Runner diam., in., d	144	40	72	130	72	96
Unit horsepower, HP_1	0.95	3.44	9.55	14.14	1.25	0.11
Unit speed, r.p.m. N_1	2.67	14.16	53.50	10.61	18.00	8.39
Characteristic diam, in. d_c	148.00 [1]	21.50	23.30	34.70	64.60	290.00
Characteristic speed, r.p.m., N_c	2.66 [2]	26.30	165.00	40.00	20.00	2.78

[1] Physically an impossibility under 1 ft. head as the diameter of an overshot wheel is fixed by the head and not by power or speed. Similarly the d_c of column 6, while not so positively a physical impossibility, is practically so.

[2] To obtain the corresponding figure or characteristic speed in the metric system multiply by 4.46, which allows for the slight difference in metric horsepower and for head expressed in meters rather than in feet.

one runner; i.e., for a twin turbine divide total turbine horsepower by 2 and for a quadruplex by 4.

8 As an illustration of the universal application of the basis of comparison, Table 1 is given, which contains examples from actual installations; even figures, however, being used throughout.

9 The final figure in each column is a direct indication of the character of the runner and comparison of these figures shows the relative speeds of the various types under any given condition of head and power. The universal application of this characteristic speed is evidenced by the fact that by its means the performance of any extremes of type may be compared. For example, an overshot wheel is shown to have about one-tenth the speed of the Fourneyron wheel, which was one of the types that superseded it. Similarly, modern wheels may attain from three to eight times the speed of the Fourneyron, and for that reason they have in turn displaced it.

10 The last three columns of Table 1 are indicative of the field which exists for medium- or low-speed runners. They cover head conditions where mechanical features of strength, or hydraulic conditions governing wear, limit the desirability of attaining high speed. For example, for the conditions of column 4, higher speed is readily obtainable from the runner end and is very desirable from the standpoint of generator design. The medium-speed type of runner is used from considerations of life and strength of the runner, both of which would be decreased at higher speed according to the present state of the art. Similarly, the conditions of columns 5 and 6 are best met by low-speed types, the limiting feature being generator speed, although runner design would readily permit higher limits. Hereafter in this paper comments on application will be confined to the high-speed, low-head type of runner, primarily applicable to heads under 100 ft.

HISTORICAL

11 Historically, the progress in hydraulic-turbine building may be excellently illustrated by noting the increases in characteristic speed that have been effected. The earliest types of water wheels were the current wheels of the flat paddle type, used for irrigation, of which there are records antedating the Christian era.[1] These developed into the various forms of overshot, undershot and breast wheels prevalent during the first half of the 19th century, very infrequently reaching a capacity of 100 hp. Their characteristic speed varied up to a maximum of possibly 3, and, as a consequence of their application to such small heads, their r.p.m. was very low, averaging probably under 20.

12 Lack of necessity for capacities beyond what could be

[1] Roman and Chinese particularly.

absorbed by a millstone or saw (see Fig. 6) and crudeness in power-transmitting machinery held capacities and speeds down to low limits. These limits began to be raised about 1825 and the turbine types of water wheel soon displaced their cumbersome predecessors. From about 1825 to 1840 two Frenchmen, Fourneyron and Jonval, developed two general types which were the forerunners of the turbine as it is known today. These are respectively the radial (outward)-flow and axial-flow types and are still known by the names of their originators, both of whom, according to records, had exceptional knowledge of hydraulics. These types of wheels were developed both in Europe and America with efficiencies exceeding 80 per cent, but demands for speed and capacity were such as to limit specific speeds to between 20 and 40 and capacities to considerably under 1000 hp.

13 The next radical step is found in our present form of mixed-flow turbine, which resulted from successively increasing the capacity of the radial-inward-flow type of turbine until the buckets or vanes received the water radially and discharged it axially. This is the type of runner used for low-head work up to the present time, it being designated originally under various trade names such as Hercules, Sampson, Success, New American, and by the names of various designers. More generally it has been known as the reaction or Francis type, though strictly the latter name may not be applied to the mixed-flow runner as appropriately as to the pure radial-inward-flow type which Francis brought to a high state of perfection around 1870.

14 The mixed-flow runner which has been used in developing probably well over 90 per cent of all the water power produced in the world from medium to low heads has had most of its development and reached its highest state of perfection in America. This statement is based on the fact that record performances in size, efficiency, capacity and head of hydraulic turbines have been made and held in America throughout the greater part of the period covering modern turbine development. That this is probably little realized even by our own engineers is strikingly illustrated by the following unqualified statement of an editorial writer in one of our foremost technical periodicals:

> "While the best skill and data in this line are found in European practice, it is no small triumph for American engineers that have so applied experimental knowledge as to obtain in working units the highest efficiencies to date for turbines of this character."

[illegible] lies in the fact that the [illegible] plant, there being nothing in [illegible] compares with it.

[illegible] graphically the history [illegible] as outlined above. These [illegible] data available and, while there [illegible] enveloped by the lines

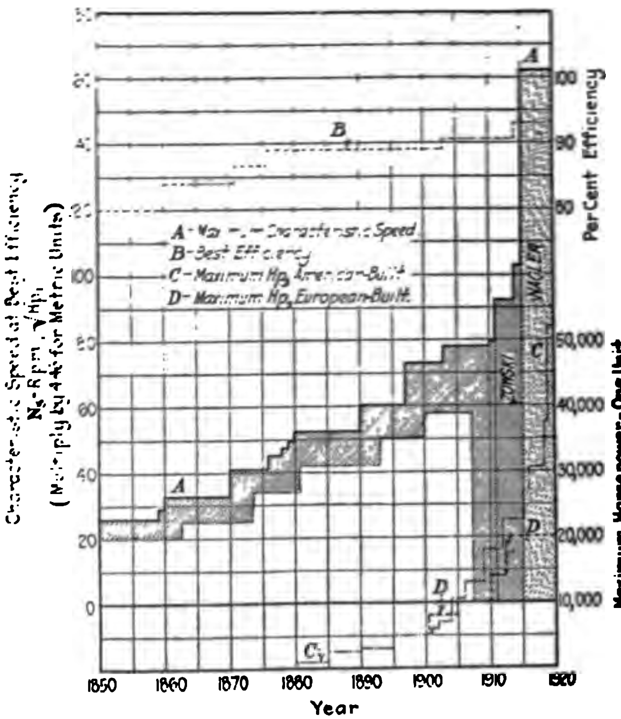


FIG. 1. GRAPHICAL REPRESENTATION OF THE DEVELOPMENT OF THE HYDRAULIC TURBINE

shown, their form would be affected to a negligible extent by their inclusion. Curve *B* illustrates the extent to which practical limits of efficiency have been reached. That most rapid progress has dated from the nineties is very strikingly brought out by curve *A* of characteristic speed and curve *C* of turbine capacity. Both of these criteria of improvement go hand in hand with electrical development. Curve *A* further indicates that characteristic speed was advancing by only moderate amounts, although its improvement had been steady for a considerable period. It is this curve which points out the radical nature of the increases effected by the new type of runner forming the subject-matter of this article, its characteristic speed exceeding previous records by over 50 per cent at the outset. The increases made by Zowski in 1911 and in 1914 when characteristic speeds of approximately 90 and 102, respectively, were attained, were the subject of the most widespread comment among engineers and were the basis of great advance in the design of low-head, direct-connected units.

16 Since the speed of a runner is directly proportional to its

Fig. 2 Typical Group of Modern Mixed-Flow, Reaction (Francis) Runners Contrasted to Impulse-Type Wheel Shown on the Right

These two types are the basis of practically all hydraulic-turbine development since about 1900, with the exception of the type of runner forming the subject-matter of the paper. Characteristic speeds are about 3 for the impulse wheel and from 25 to 95 for the reaction wheels.

characteristic speed, inspection of Curve *A* will indicate that under any given conditions this new type of runner will permit of speeds over 50 per cent in excess of those possible heretofore. The effect of such an increase will be instantly apparent to any one who is familiar with the extensive efforts made to increase electrical generator speed in hydroelectric plants by means of belts, gears, the multiple runners of twin, quadruplex and even octuplex turbines, etc. The high cost, complication, loss of power, and departure from simplicity of such devices are fundamental disadvantages that only elimination will correct, and this elimination has heretofore been attained at the sacrifice of speed. The generator builder and the owner who pays for the unit and the plant to house it are then the individuals most concerned, the former with design difficulties, the latter with increased cost, and both with lower efficiency. The plant at Keokuk is a striking illustration of this feature, direct connection and large capacity being obtained only by using speeds between 50 and 60 r.p.m. as contrasted to the 90 or 100 r.p.m. readily obtainable with the new type of development.

FIG. 3 ONE OF THE FIRST COMMERCIAL APPLICATIONS OF THE NEW "SUCTION" TYPE OF RUNNER DESIGNED IN 1916

Two units were constructed for a rating of 100 hp., 8 ft. head, 225 r.p.m. At the left is shown a high-head Francis wheel having about one-eighth the characteristic speed of the new type of runner, the characteristic speed of the former being about 20.

DEVELOPMENT OF THE NEW RUNNER TYPE

17 In 1907 and 1908 the author was connected with some rather extensive field work comprising erecting, experimentally improving, and testing of some large-size axial-flow or screw pumps. This work concentrated all attention on a single type of hydraulic impeller for over a year and it was only natural that impressions then formed should greatly influence his trend of thought in later work, which has been exclusively along hydraulic-turbine lines. At any event the effect was such that the accepted form of reaction (Francis)

runner, illustrated typically in Fig. 2, then and still the basis of practically all low-head turbine design, seemed unnecessarily complicated and without logical justification from any hydraulic or mechanical standpoint. These ideas crystallized in 1913 when definite application of the axial-flow principle was shown as giving inherently less wetted surface, simpler passages and greater mechanical strength than the corresponding reaction runner having radial inlet and axial discharge.

18 To bring out the comparison most effectively the initial drawings showed the axial-flow runner sketched in on the outline of the reaction runner, using the same runner band and showing the additional advantage of being able to use either the usual radial inlet guide case or the straight axial case. Most but not all of subsequent commercial applications have been along the former lines, but undoubtedly the still greater simplicity effected by the latter, especially in horizontal settings, will bring it into prominence. Models were made with the least possible delay and theories checked out practically with actual runners. New lines of improvement naturally became evident during trials but the original profiles were left fundamentally intact, with the result that the form of turbine runner shown in Fig. 3 was developed. Inspection of this figure reveals the fact that the entire design is based on a straight radial blade, which offers the absolute minimum of wetted surface and of bending moment on the root of the blade. It is by reason of these two fundamental advantages emphasized by the simplicity and inexpensiveness of the design that the author believes the new type of runner will supersede the mixed-flow or Francis type. Runners of practically the axial-flow type but roughly conical in profile may possess certain desirable features of strength or form of passage, but the measure of their advantage is largely indicated by the nearness with which they approach the flat form.

19 Commercial installations of any considerable size were naturally approached with the greatest care as it was difficult to anticipate what effects there might be due to the critical state of water resulting from the high velocities used. The initial small plants designed in 1916 operated without any difficulties with regulation such as might have been expected, nor did commercial operation show any noticeable difference from previous types.

20 Tests on models were verified by Holyoke tests in 1917, the initial design being used partly for commercial reasons and partly for historical purposes. Only one design of blade was tested,

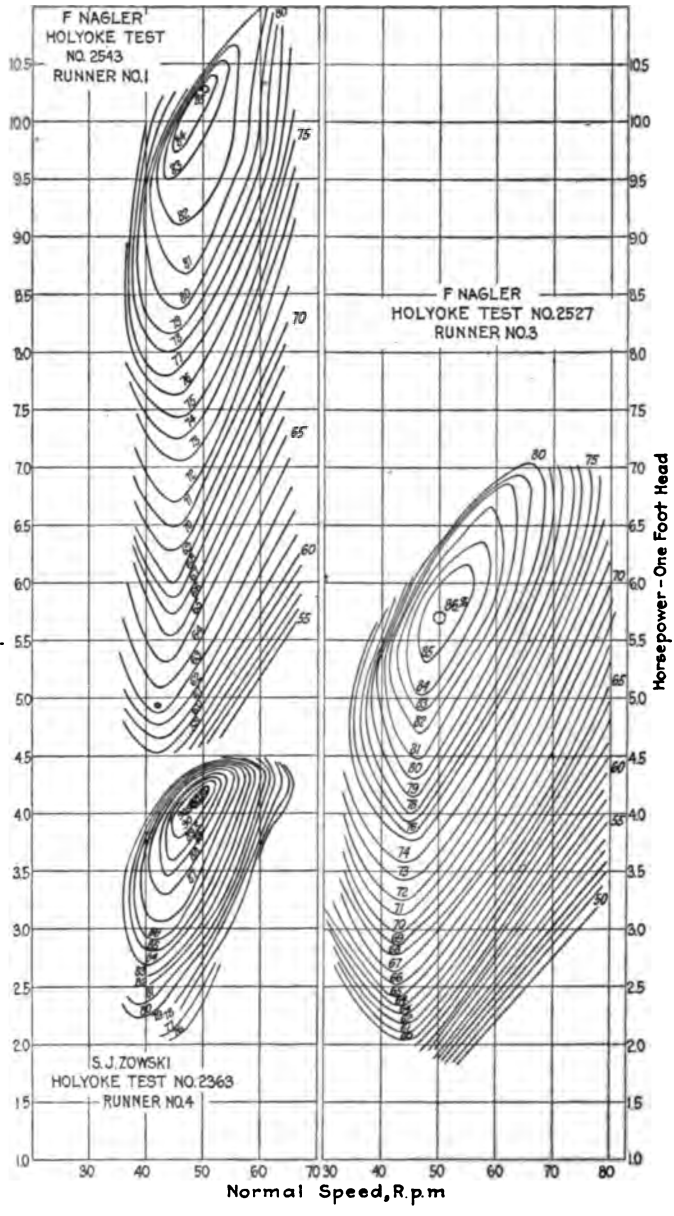


FIG. 5 COMPARISON OF THE ZOWSKI AND NAGLER TESTS

although two runners, one having three and the other four blades, were also given runs. A typical set of results of these initial tests is shown in Table 2, and although they do not equal in efficiency later tests obtained on improved runners, the figures were nevertheless used in plotting some of the curves in Figs. 4 and 5. Comparisons are confined to speed and capacity for the purpose of this description and previous best results are also plotted in Figs. 4 and 5. The basis of these curves is taken directly from Professor Zowski's comparisons in *Engineering Record* of December 26, 1914, and it should be noted that they are plotted to show comparative powers on the basis of constant speed as outlined previously herein, (see basis A, Par. 4), although a common basis of 50 r.p.m. rather than unity is used. This power basis exaggerates the comparison con-

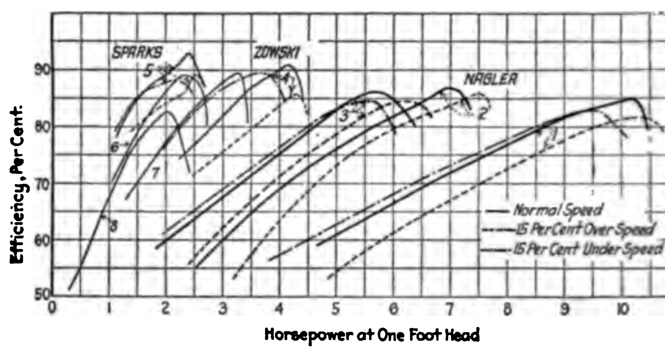


FIG. 4 POWER-EFFICIENCY CURVES FOR WHEELS HAVING A NORMAL SPEED OF 50 R.P.M.

siderably over the characteristic-speed basis, but is retained to permit of direct reproduction of curves showing results typical of the state of runner development prior to the author's work.

21 Referring again to Fig. 4, the highest speed ordinarily offered in commercial work is typified by curve No. 4. Contrasted to this are shown curves Nos. 1, 2 and 3, the second of which is deduced directly from the Holyoke test data of Table 2. The most definite indication of the field of application of the new form of runner is obtained from Fig. 4, on the following basis:

> Under any given conditions of head and with such minimum limit of speed as might be imposed by consideration of cost, efficiency and floor space, runner No. 1 will give 140 per cent and runner No. 2 will give 65 per cent more power than runner types previously available as typified by No. 4.

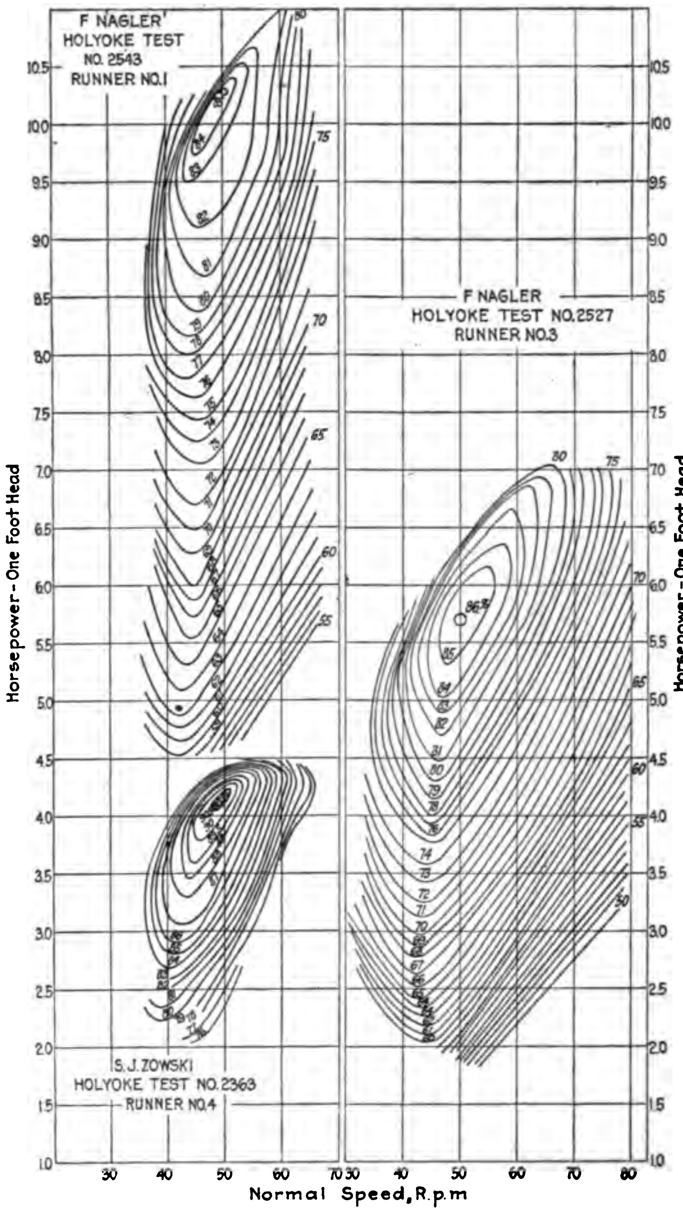


FIG. 5 COMPARISON OF THE ZOWSKI AND NAGLER TESTS

TABLE 2 TYPICAL LOG SHEET OF THE HOLYOKE TESTS

No.	Proportional part of opening of speed gate, in.	Proportional part of full discharge, per cent	Head acting, ft.	Speed, r.p.m.	Relative velocity	Quantity of water discharged, cu. ft. per sec.	Power developed hp.	Efficiency, per cent
21	4.00	0.940	13.35	285.25	1.444	77.95	94.39	79.98
20	"	0.953	13.31	301.00	1.526	78.91	96.39	80.93
18	"	0.964	13.30	316.00	1.603	79.77	97.82	81.30
17	"	0.974	13.26	330.00	1.676	80.52	98.63	81.46
16	"	0.989	13.22	346.60	1.763	81.61	99.89	81.64
15	"	1.006	13.15	363.60	1.855	82.80	100.91	81.72
14	"	1.026	13.08	381.50	1.951	84.23	101.81	81.48
13	"	1.060	12.97	410.25	2.107	86.64	102.91	80.75
12	"	1.097	12.89	438.00	2.257	89.41	102.86	78.70
11	"	1.156	12.68	477.50	2.480	93.45	101.94	75.86
8	3.75	0.910	13.29	292.00	1.482	75.30	93.51	82.39
7	"	0.917	13.27	305.00	1.549	75.82	94.41	82.74
6	"	0.934	13.44	330.50	1.668	77.74	98.78	83.36
5	"	0.948	13.41	347.60	1.756	78.80	100.18	83.60
4	"	0.970	13.36	369.25	1.869	80.52	102.48	84.00
3	"	0.994	13.26	389.67	1.979	82.15	103.99	84.17
9	"	1.002	13.04	393.75	2.017	82.15	101.92	83.90
2	"	1.031	13.16	419.50	2.139	84.88	105.23	83.07
1	"	1.078	13.01	458.25	2.350	88.30	105.17	80.72
10	"	1.096	12.75	471.20	2.441	88.86	100.59	78.29
22	3.50	0.869	13.30	275.00	1.395	71.96	88.06	81.13
23	"	0.879	13.27	292.00	1.483	72.68	90.39	82.64
24	"	0.889	13.24	306.20	1.557	73.41	91.52	83.03
25	"	0.902	13.19	325.00	1.655	74.35	93.67	84.22
26	"	0.922	13.13	346.20	1.767	75.82	96.08	85.12
27	"	0.940	13.06	366.50	1.876	77.10	97.80	85.64
31	"	0.953	12.98	376.50	1.933	77.95	98.46	85.81
28	"	0.967	12.97	388.50	1.995	79.02	99.53	85.63
29	"	0.977	12.91	402.75	2.073	79.66	98.88	84.78
30	"	0.991	12.85	416.40	2.149	80.63	97.79	83.22
32	"	1.032	12.66	449.75	2.338	83.35	96.02	80.23
33	"	1.074	12.47	484.60	2.538	86.09	90.52	74.35
44	3.25	0.840	13.36	280.00	1.417	69.69	88.17	83.50
43	"	0.852	13.33	297.50	1.507	70.62	90.51	84.77
42	"	0.861	13.29	313.50	1.591	71.24	92.03	85.71
45	"	0.872	13.30	325.60	1.651	72.17	93.84	86.20
41	"	0.879	13.22	334.50	1.702	72.58	94.62	86.95
40	"	0.890	13.19	348.20	1.773	73.41	94.78	86.31
39	"	0.900	13.17	363.25	1.852	74.14	95.00	85.79
38	"	0.914	13.12	379.00	1.935	75.19	95.07	84.98
37	"	0.933	13.03	397.50	2.037	76.46	95.47	84.50
36	"	0.947	12.97	412.75	2.120	77.42	94.73	83.18
35	"	0.966	12.91	432.50	2.227	78.80	92.33	80.03
34	"	1.003	12.77	466.50	2.415	81.39	87.14	73.93
56	3.00	0.809	13.56	278.33	1.398	67.65	86.16	82.82
55	"	0.820	13.49	294.20	1.482	68.37	87.93	84.06
54	"	0.832	13.41	312.33	1.578	69.18	90.02	85.56
53	"	0.841	13.40	328.00	1.657	69.89	91.03	85.71

TABLE 2 TYPICAL LOG SHEET OF THE HOLYOKE TESTS (Continued)

No.	Proportional part of opening of speed gate, in.	Proportional part of full discharge, per cent	Head acting, ft.	Speed, r.p.m.	Relative velocity	Quantity of water discharged, cu. ft. per sec.	Power developed, hp.	Efficiency, per cent
52	3.00	0.848	13.38	340.00	1.719	70.41	90.73	84.92
51	"	0.851	13.35	351.25	1.778	70.62	89.98	84.16
50	"	0.875	13.28	372.75	1.892	72.37	91.51	83.96
49	"	0.895	13.20	399.00	2.031	73.83	91.57	82.85
48	"	0.913	13.15	419.50	2.140	75.19	89.56	79.87
47	"	0.944	13.05	448.67	2.297	77.42	86.21	75.24
46	"	0.977	12.94	480.00	2.468	79.77	81.98	70.03
65	2.50	0.731	13.36	270.50	1.369	60.68	73.63	80.08
64	"	0.739	13.34	288.00	1.459	61.27	75.32	81.25
63	"	0.740	13.33	299.75	1.519	61.37	75.19	81.05
62	"	0.744	13.32	312.00	1.581	61.66	74.93	80.45
61	"	0.748	13.33	324.75	1.645	61.96	74.53	79.57
60	"	0.767	13.28	351.00	1.782	63.44	74.93	78.43
59	"	0.789	13.18	384.33	1.958	65.03	75.90	78.08
58	"	0.818	13.08	418.50	2.140	67.14	71.48	71.77
57	"	0.841	12.98	448.00	2.300	68.77	66.95	66.13
73	2.00	0.627	13.55	260.67	1.310	52.38	58.43	72.59
72	"	0.635	13.70	287.25	1.436	53.32	61.32	74.02
71	"	0.645	13.65	308.75	1.546	54.07	62.62	74.81
70	"	0.654	13.64	328.67	1.646	54.82	63.15	74.46
69	"	0.664	13.60	351.50	1.763	55.58	61.91	72.22
68	"	0.676	13.55	376.33	1.891	56.53	60.26	69.36
67	"	0.694	13.50	411.00	2.069	57.87	54.84	61.90
66	"	0.715	13.45	445.25	2.246	59.52	47.53	52.35
81	1.50	0.526	13.74	257.75	1.286	44.23	45.40	65.87
80	"	0.530	13.71	281.75	1.408	44.59	46.62	67.24
79	"	0.539	13.69	311.00	1.555	45.30	46.48	66.08
78	"	0.547	13.74	338.00	1.687	46.00	45.10	62.92
77	"	0.552	13.79	359.25	1.790	46.54	42.18	57.96
76	"	0.559	13.77	381.50	1.902	47.07	38.69	52.63
75	"	0.567	13.76	403.25	2.011	47.71	32.28	43.36
74	"	0.576	13.73	428.50	2.139	48.43	25.16	33.36
82	4.00	1.503	12.66	694.00	3.608	121.40	Dynamometer removed for free runs.	
83	3.75	1.421	12.85	690.00	3.561	115.62		
84	3.50	1.336	13.13	684.00	3.492	109.91		
85	3.25	1.218	13.07	657.00	3.362	99.99		
86	3.00	1.155	13.26	654.67	3.326	95.50		
87	2.00	0.805	13.34	555.00	2.811	66.74		

The turbine runner and shaft, the weight of the dynamometer and of that portion of the shaft which was above the lowest coupling were suspended by ball bearing.

With the flume empty a strain of 5 lb., applied at a distance of 2.8 ft. from the center of the shaft, sufficed to start the wheel.

Comparison on a speed basis using equal power would illustrate the application of the new type as follows:

> Under any given condition of head and power, runner No. 1 will give over 50 per cent and runner No. 2 will give 30 per cent higher speed than runner types previously available as typified by No. 4.

22 Fig. 5 is a reproduction of the "equal efficiency" diagram of the December 26, 1914, *Engineering Record* article. The scale has been extended to permit of showing the relative location of the new point of best efficiency and the great flexibility inherent in the new form of runner. The diagram of runner No. 3 is shown at the right on repeated abscissæ in order to avoid too much overlap of diagram and consequent confusion. The ordinates remain the same. The shape of the "efficiency hills" is somewhat different from that of the mixed-flow runner, although perhaps not as much so as might be expected from the great difference in runner forms. That the guide cases (gates) were of the same design for both forms may be somewhat responsible for the runner not showing greater diversity in characteristics.

23 Fig. 5 presents an excellent basis for showing the contrasting speed characteristics of runners. Obviously their having been recomputed to 50 r.p.m. causes all the highest efficiency areas to lie on the same vertical line. This simplifies comparison in that the horizontal width of the equal-efficiency areas indicates flexibility of the runner or its ability to maintain good efficiency under varying speed or head, the latter being the variation encountered in commercial application. The position of the highest efficiency area on the vertical ordinate indicates whether it is of a high-capacity or low-capacity type or similarly whether it is a high- or low-speed runner. Its suitability for low-head work is determined largely by its vertical position.

24. Naturally after a training of probably forty or fifty years on the basis that the mixed-flow reaction or Francis type of runner is the only type suited to low heads, it may seem rather radical to the engineering profession to propose something different. That positive and skeptical criticism is to be expected is certain, hence it was not until after numerous commercial installations had been installed, tested and subjected to every conceivable adverse condition that a complete showing of the new design could be ventured.[1]

[1] The patent situation had also to be considered, all features connected with the designs of runners shown herein having been covered by patent application.

Bearing on this point it may be stated up to date seventeen commercial runners of this type, varying from 80 to nearly 1000 hp. in capacity, have been built or are building for a total of nine plants. Nine of these have been tested out thoroughly in place with an actual showing of characteristic speeds often over 200 (in the metric system 900) under abnormal low-head conditions when synchronous speed was maintained during high water.

COMMERCIAL ADVANTAGES

25 Numerous and sometimes unexpected advantages have been found to result from the simple and open form of runner. For example, a runner may be taken out and another substituted for capacity variation under flood conditions or for test purposes without removing any other turbine parts except two or three guide vanes. In two plants this was made use of to install a high-capacity runner for obtaining more power during flood periods. Likewise these higher-capacity runners were made by using the original core boxes, which means that blade angles were unchanged, the result being accomplished simply by using a different number of blades.

26 The greatest mechanical advantage arises from the fact that the runner cannot clog up with sticks, blocks, leaves or other debris, a feature which the author's experience with the average low-head installation would indicate should result in several per cent more power the year around.

27 The primary advantages which were anticipated and which have proven out in practice are as follows:

a Lower generator cost due to an increased speed of 50 per cent and over above previous practice. This saving varies from 15 to 35 per cent of the generator cost, depending on its size and speed

b Lower turbine cost due to simpler runner. This averages around 10 per cent, the runner being about one-third the weight of the corresponding mixed-flow type and much easier to build, either solid or with separate blades

c Smaller generator diameter, which means smaller power house

d Higher generator efficiency due to better design possible with the higher speed. This is seldom less than 2 per cent gain and may conservatively be stated to vary from $1\frac{1}{2}$ to 3 per cent *e* Greater turbine flexibility, which permits the plant to give more power under flood conditions when the head is greatly reduced. This runner has an overspeed of about 100 per cent as contrasted to perhaps 60 to 75 per cent for previous types. This means that its efficiency and consequently its power will not become zero until the head has been reduced to about one-quarter normal as contrasted to four-tenths or one-third normal for reaction types. This advantage lies not solely at the extreme limit of minimum head but at all abnormal heads or speeds, as the efficiency is less affected than with the other types. (See Fig. 4 showing abnormal-speed curves.)

28 At the present time efficiencies equaling the records of reaction (Francis) wheels have not been reached, but they are being approached rapidly and with the inherent advantage of better generator efficiency, equivalent combined results for the unit are only a matter of short time. Fundamentally the axial-flow runner should give greater efficiency than the mixed-flow purely from considerations of wetted surface and hydraulic friction, to say nothing of the simpler form permitting of greater accuracy of construction and more correct design.

TYPE OF RUNNER

29 As to the nature of this new type of runner, it may be said that from direction of flow it is undoubtedly a pure Jonval type, although his runner consisted of a narrow row of blades on the periphery of a comparatively large disk, and his characteristic speeds seldom exceeded 20 or 30 as contrasted to the present 100 to 200. Furthermore the Jonval type was a pure reaction wheel, which infers jets issuing from orifices or channels, these being noticeably lacking in the runners of Fig. 3.

30 Professor Baudisch in some of his mathematical studies on the design of high-speed axial-flow runners has introduced a type name which translated literally is "suction jet," the conclusions from his calculations being that in order to produce extraordinarily high characteristic speed it is essential to run into certain underpressure conditions such as result when velocities greater than $\sqrt{2gH}$ are produced in the throat of a diffusing nozzle discharging under a head of H feet. The term "suction turbine" impresses the writer as being quite appropriate, but from another

reason which may be outlined as follows: Underpressure is not essential to high characteristic speed, though it may frequently occur. The primary essential to high characteristic speed (125 to 200) is a reduction of hydraulic friction and harmful centrifugal forces. Neglecting friction and possibly blade thickness, there are no mathematical or hydraulic laws that will prevent doubling or quadrupling any particular characteristic speed by simply flattening the blade angles. A direct analogy to this is the well-known illustration of relative velocities evidenced in the sail of an ice boat. The practical effect of so doing with a given profile is to lengthen the blade so that the friction on the increased wetted surface reduces efficiency and speed or results in constriction of passage. In the author's design these effects are counteracted by cutting out blades, the effect of which is not manifested in the reduction of power that might be expected. On the contrary the discharge is increased *without* reduction of efficiency.

31 Investigation of thrust and power shows that the force on each blade exceeds the product of blade area and the total apparent head, which can only mean that in such cases there is less than atmospheric pressure on the back side of the blade aside from that due to draft head. This is strictly analogous to the distribution of the total force on the wings of an airplane, less than half of which is pressure from below, the remainder being suction on the upper surface. As such suction action is in evidence primarily with these high speeds, the author believes the term "suction" to be peculiarly applicable to this form of turbine runner.

PERIPHERAL COEFFICIENT AND RUNAWAY SPEEDS

32 It may be of interest to engineers who have studied runners to note the high coefficients obtained with this type of design. At normal speed the runner has a peripheral coefficient ranging as high as 2.00 as contrasted to the usual 80 to 85 per cent. Similarly at runaway speed the peripheral coefficient is around 4.00 or a speed four times as fast as the full spouting velocity ($\sqrt{2gH}$) of the driving water. At Holyoke it was very unexpectedly necessary to remove the brake pulley in order to safely measure runaway speed. This high limit is quite at variance with the usual trend of decreasing runaway speed as the characteristic speed increases. The lowest-speed runners or the impulse type have the high overspeed of about 100 per cent, a result which reaction (Francis) runners

approach less and less closely as their characteristic speed increases from 10 to 100.

DISCHARGE LOSSES AND DRAFT TUBES

33 It is difficult in practice to secure these high characteristic speeds without correspondingly high velocities of the water at the runner discharge. Such high exit velocities running from $0.50 \times \sqrt{2gH}$ up to $0.80 \sqrt{2gH}$ would incur tremendous efficiency losses were no draft tube or diffuser used. On low heads and large capacities a long, straight tube is usually uncommercial on account of the excessive excavation involved and some form of a radial-outward-flow type is practically necessary. White's "Hydraucone Regainer" is the most perfect commercial solution of this diffuser problem yet found for general conditions, and practically all of the installations using the author's runner have been furnished with this hydraucone built either of steel or concrete.

CONCLUSIONS

34 In conclusion it is probably well to outline a few of the possibilities resulting from the development of this runner. In pointing these out it is realized fully that such features as the limit of application to higher heads and possibilities of pitting can only be determined by practice. Keokuk has already been referred to as an illustration of an installation where a tremendous advantage would result from the employment of a speed of 90 or 100 r.p.m. as contrasted to one between 50 and 60. A like saving would probably be effected by using a 30,000-lb. runner made without cores and easily transported as contrasted to one weighing 130,000 lb. and ranking among the most complicated castings ever made, in one case having to be split for transportation.

35 In low-head plants where a large power is to be developed it is desirable to use units as large as are feasible. Practical limits, however, are set by generator speed and by size of parts which may cause difficulties in manufacturing, transportation or in placing the turbine structure between available water levels. The first limit is raised by the new design in that for any given generator speed more than double the power can be developed per unit than with the mixed-flow types of reaction wheel.

36 More general though equally significant illustrations of the extent to which the new type of runner may increase possibilities of hydroelectric development are afforded by the consideration

of the following: With any given limit of minimum generator speed and for any given power, the new type of runner will permit turbines to operate under heads one-half as high as those required by the mixed-flow (Francis) runners of prior practice. Similarly, for large powers such units can be developed with low runner cost and without transportation difficulties. As an illustration, this design is the basis of some contemplated units of 800 hp. each under 9 ft. head at 90 r.p.m., a design which incidentally eliminates all gates on the turbine and effects all regulation from gates which form part of the power house on the discharge side of the turbine.

FIG. 6 AN EARLY TYPE OF TURBINE

This turbine was made almost entirely of wood, even to the 18-in. shaft. It was in operation until a few years ago.

37 Probably the most novel application is to horizontal plants where it is necessary to replace old turbines but desirable to retain the electrical equipment. This has been done practically, a single high-speed runner replacing an old quadruple turbine and giving the same speed and more power with the same head.

38 Applying an axial-flow guide case with two 45-deg. bends in the draft tube gives as simple flow as possible with any arrangement and contrasts very favorably with the four 90-deg. bends given to the water with a radial-guide-case setting of a horizontal unit. This arrangement and, when floods are prevalent, the single

vertical setting using a hydraucone regainer are probably the most advantageous forms of hydraulic-turbine settings that can be devised.

39 To a generation of engineers whose practical connection with water power has been based almost exclusively on the use of one type of turbine the author's statements as to the short period of time covering real progress may seem somewhat overdrawn. However, one need only look back a comparatively short time to perceive that modern water wheels are young compared to other forms of prime movers. This was never brought to the author's

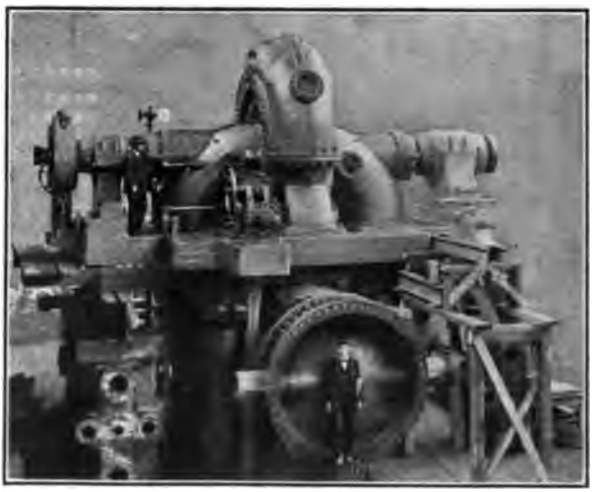


FIG. 7 A 25,000-HP. DOUBLE-DISCHARGE TYPE OF STEEL SPIRAL-CASED FRANCIS TURBINE OPERATING UNDER A HEAD OF 485 FT.

attention more forcefully than by consideration of the views shown in Figs. 6 and 7. The former is reproduced from a photograph presented to the author by Wm. G. Fargo, of Jackson, Mich., and shows a single vertical open-flume turbine having about an 18-in. *wooden* shaft, which was squared at its lower end to receive the four flat paddles forming the runner. Water was admitted through crude "rabbit trap" gates diagonally opposite each other. This wheel was in operation under about 9 ft. head driving a millstone up to a very few years ago, less than fifteen if memory serves correctly. Fig. 7 shows one of the most extreme types using a mixed flow reaction or Francis runner. This turbine, if not actually in operation at the same time as that shown in Fig. 6, missed being so by but a very few years. That two such extremes so nearly overlap in period of time is the best evidence the author can offer to the effect that the field of hydraulic-turbine design still offers tremendous opportunities for improvement.

DISCUSSION

L. F. HARZA[1] (written). The writer had the opportunity, about two years ago, of examining the Holyoke test of one of the author's new runners and since that time has awaited with considerable interest the time when he might feel justified, by results accomplished, in giving publicity to his discoveries and inviting discussion before the engineering societies.

The introduction of the new runner promises to be an important milestone in the development of hydraulic turbine practice. The interesting feature is that the axial-flow turbine was believed to have been definitely and permanently superceded by the inward-flow turbine because of inherently higher speed characteristics of the latter.

But why have we always accepted the inward-flow turbine as inherently higher in speed? It is because former axial-flow turbines, of which the Jonval is the best known, were based upon the reaction principle, as is the Francis type, and the relative speed possibilities of the two could be readily compared because they acted upon the same principle.

In the reaction turbine, whether axial- or inward-flow, the water passages between buckets converge toward the discharge edge and the water is therefore under static pressure, undergoing acceleration until it emerges in a nearly tangential direction at the discharge edge as from an orifice, with a velocity approaching the spouting velocity of water, which, moreover, cannot be exceeded by the buckets.

It is evident then that the turbine will rotate at greatest speed in which the effective radius of bucket discharge is the least; hence the inward-flow turbines apparently inherit speed superiority.

But now enters an axial-flow turbine of entirely new conception. Apparently it operates upon the same principle as the ice boat, which, according to theoretical mechanics, might travel at infinite speed were it not for wind and ice friction. It is, theo-

[1] 115 S. Dearborn St., Chicago, Ill.

retically, merely a matter of the flatness of the exposed surface of the turbine runner vanes or ice boat sail to the direction of the wind.

In practice it appears to the writer that the only limit to the runaway speed of the new turbine is one of blade thickness, mechanical and hydraulic friction, and other hydraulic losses. There is no fixed jet velocity to limit speed, as in other types.

It is the writer's understanding that the new runner requires a much higher velocity at the entrance to the draft tube, thereby increasing the difficulties of securing sufficient length of tube for gradual expansion and for efficient curves.

The use of White's hydraucone to overcome this handicap is ingenious and doubtless hydraulically valuable, but in the only case in which the writer has investigated, it increases the length of the power station about 50 per cent and the cost of the superstructure and substructure in almost as great proportion. This raises the question of the economy of the additional speed and lower cost of machinery, if the increased cost of powerhouse and substructure is necessary to realize the higher speeds of such units.

It is a problem for solution upon the merits of each individual case. The writer would like to know if sufficient experience has yet been gained to justify general comparisons of probable station costs, as between a station equipped with old type runners or with the new runner and the hydraucone.

A. M. Greene, Jr., called attention to the fact that from Fig. 1 the efficiency of the Nagler turbine appeared to be 93 per cent while in Fig. 4 the efficiency was shown to be in the neighborhood of 85 per cent. He also took exception to the author's statement that White's hydraucone offered the most perfect solution of the diffuser problem yet found for general purposes, and pointed out that this had not yet been proved.

Clemens Herschel said that the so-called Francis wheel was in fact wrongfully named. It was the invention of A. M. Swain, and was first manufactured at North Chelmsford, Mass. The *Journal of the Franklin Institute* for April, 1875, contains an account of a test of it, made by James B. Francis, showing up to 83 per cent useful effect. Other tests had previously been made by Hiram F. Mills and others, and are recorded in the *Journal of the Franklin Institute*, vol. 59, no. 3. These tests showed an efficiency of 81.7 per cent at full and $\frac{7}{8}$ gate, and 80.9 per cent at $\frac{3}{4}$ gate, which was exceedingly good for an economically constructed hydraulic turbine.

The wheel described reminded him of a wheel that was considered a freak when it reached the Holyoke Testing Flume, then just opened, about 1881. It was a sort of Jonval wheel, but had no guides; and in despite of its abnormity had surprisingly good efficiency. Wheels in those days were constructed solely by inspiration, not calculation. If not right when first tried, they were tinkered with until they did better. Some were "over-gated" as it was called; others were "under-gated."

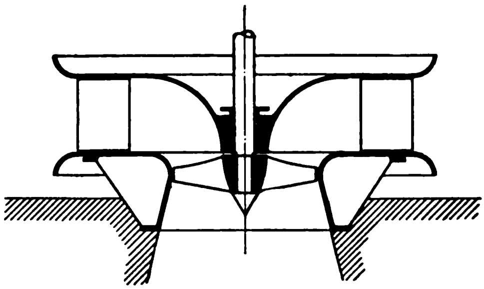


FIG. 8 DIAGRAMMATIC SECTION THROUGH NEW RUNNER AND GUIDES

THE AUTHOR, in answer to a question, said that not all of the efficiencies of the nine plants mentioned in Par. 24 had been measured; most of them had been measured and none of them were below 85 and none above 90 per cent.

Speaking of the part-gate efficiency of the runner, he said that this, in common with the part-gate efficiencies of all high-speed runners, was 6 to 8 per cent lower at half load than the average runner. Hence the real field of the runner was in such an installation as that at Keokuk or that on the St. Lawrence River, where, with a tremendous number of units, the part-gate efficiency would not be a deciding factor in the problem.

No. 1724

THE HVID ENGINE AND ITS RELATION TO THE FUEL PROBLEM

BY E. B. BLAKELY,[1] CHICAGO, ILL.
Non-Member

This paper describes a type of engine, the Hvid, which, it is claimed, has all the advantages and none of the disadvantages of the so-called Diesel type, and which is being produced in units as small as 1½ hp. It can be started cold, has no complicated air-compressor system, and is so economical that it can compete with gasoline engines of the same size. The author outlines the operation af the engine, describing in detail the suction, compression, power and exhaust strokes and presenting a series of indicator cards. Fuel consumption is shown by means of curves obtained as the results of tests performed. The paper concludes with a discussion of the heat balance and torque characteristics of the engine. It is not claimed that this motor has reached its ultimate state of development, but rather that it possesses certain characteristics which, in the author's opinion, ought to attract many internal-combustion engineers by the possibilities they hold out of helping to solve some of the country's fuel problems.

IT has been estimated that internal-combustion engines now furnish approximately two-thirds of all the prime motive power generated in the world. These internal-combustion engines may be roughly divided into three classes:

- *a* Those burning gasoline
- *b* Those burning kerosene, tops and other light distillates, after being started and warmed up on gasoline
- *c* Oil engines of the Diesel, semi-Diesel, hot-bulb and surface-ignition types, which use for fuel crude oils, fuel oils and the cheap grades of kerosene.

Of these three classes the first furnishes about 95 per cent of all prime motive power, and it has been estimated that there are in use now in this country, burning gasoline, nearly 4,000,000 automobiles, 250,000 trucks, 500,000 motor boats, 75,000 tractors and 750,000 farm

[1] Sears, Roebuck and Company.

Presented at the Annual Meeting, December 1919, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

engines. Is it any wonder, therefore, that the demand for gasoline has increased?

2 By ordinary distillation methods, Eastern crude oils yield from 20 to 30 per cent of gasoline, Mid-Continental crudes from 16 to 20 per cent, and Gulf, California and Mexican oils from 2 to 3 per cent. The rough average of these is 15 per cent. The visible supply of crude oil is naturally diminishing, and since the oil is becoming heavier all the time the percentage of gasoline yield is lessening.

3 The supply of gasoline may be increased slightly by making use of the cracking processes, which would necessitate an increased cost of production, and also by blending high-test casing-head gasoline with kerosene, a process bound to be short-lived because our gas wells are rapidly giving out. The supply of gasoline may also be conserved by increasing the thermal efficiency of the engines and by adapting them to burn kerosene and mixtures of gasoline and kerosene. The relief gained, however, would be but temporary at best. We are having trouble enough now in burning properly the present-day gasoline without trying to burn all sorts of mixtures which at the least would necessitate constant changing of carbureting adjustments and methods.

4 Much has been written and said during the past two or three years on the subject of using kerosene as fuel in conventional gasoline engines of both the slow- and high-speed types, and while undoubtedly much has been learned concerning the characteristics of kerosene under certain conditions, the burning of kerosene in gasoline engines, so far as the writer knows, has not been accomplished with complete success up to the present time. By complete success he means starting the engine on kerosene in atmospheric temperatures approximating 0 deg. fahr. and below (for these must be reckoned with) without preliminary heating of any sort and burning the kerosene so as to eliminate troublesome carbonization and complicated and unsightly accessory apparatus, and obtain high economy.

5 In attempting to burn kerosene in modified gasoline engines we are confronted by the following basic difficulties: Kerosene and gasoline are chemically widely different substances, having nothing in common but the base from which they are derived. Their initial boiling points are wide apart, that of commercial gasoline being about 100 deg. fahr., while that of kerosene is about 330 deg. fahr. Their boiling ranges are also totally different, that of gasoline being

340 deg. fahr., while that of kerosene is about 200 deg. fahr. Gasoline-air mixtures will ignite spontaneously at approximately 680 deg. fahr., while similar mixtures of kerosene and air self-ignite at approximately 575 deg. fahr. Mixtures of gasoline and air form a permanent fixed gas, but mixtures of kerosene and air do not. Under these conditions a jet carburetor designed for vaporizing gasoline cannot be expected to vaporize kerosene. The best it can do is to atomize it.

6 In order to vaporize, as well as to prevent precipitation or condensation of the atomized kerosene in the combustion chamber, it is necessary to heat the charge, and since the power output of the engine depends upon the amount of oxygen taken in and burned during each cycle, it is clear that the more the charge is heated the less oxygen we can get into the cylinder and the less power we can obtain. This forces us to a compromise between two conflicting conditions: the maintenance of the incoming charge at the lowest possible temperature which will vaporize the kerosene, and the prevention of precipitation in the combustion chamber. This compromise might be satisfactorily effected in the case of an engine running at a constant speed and load, but in the case of an engine running at varying speeds and loads it is a very different compromise to make, because as the power demands on the engine vary, so must the total amount of heat added to the charge vary.

7 In order to obtain maximum power from any internal-combustion engine, regardless of the kind of fuel used, we must have maximum mean effective pressure, and since mean effective pressure depends largely upon compression pressure, we must use the highest compression pressure possible. This brings us again to a conflicting pair of conditions, because in order to prevent so-called preignition with its attendant disagreeable and harmful pounding, when burning kerosene we are forced to use a relatively low compression pressure, which lowers the mean effective pressure and also the power output.

8 In this connection may also be mentioned the so-called preignition knock which occurs when using too high a compression pressure with kerosene. This knock is not caused by preignition as is generally supposed, but by small detonations after ignition has occurred and the piston has started downward. These detonations are due to the fact that kerosene is of a very complex chemical make-up and that after ignition has started the conditions are most favorable to cracking it. Under these conditions the kerosene

breaks down into simpler combinations, some of which are highly detonating and others less so, and these compounds set one another off successively, according to their stability, but so rapidly as to produce a single knock.

9 That this is so, is clearly shown by comparing the two full-load indicator cards shown in Fig. 1. They were both taken from the same engine but with different cylinder heads and operating with different governors. When *A* was taken the engine was running on kerosene (43 deg. B.) with no water injection, after having been warmed up on gasoline. A throttling governor was used. When *B* was taken the engine was operating on gasoline (62 deg. B.) and with a "hit-and-miss" governor. The compression ratio was 5 to 1.

10 Many engineers believe that if the problem of utilizing kerosene for fuel in these engines now burning gasoline could be solved, the whole fuel problem would be solved. It undoubtedly would help the situation immeasurably, but who can doubt for a moment that the price of kerosene would not soar, once the demand

FIG. 1 FULL-LOAD INDICATOR CARDS OF ENGINE RUNNING (*A*) ON KEROSENE AND (*B*) ON GASOLINE (200-LB. SPRING)

for it began to grow, until finally there would be but little difference between the prices of gasoline and kerosene. The true economic solution of the fuel problem lies not in trying to adapt some particular fraction of the distillation of crude oil to the engine, but in adapting the engine to the available fuel, whether it be crude oil just as it comes out of the ground, or some by-product of its distillation.

11 There have been numerous engines built in the last ten years capable of running consistently on the various crude and fuel oils, as for instance the Diesel and so-called semi-Diesel engines, hot-bulb and surface-ignition engines. These, however, have been used mainly in marine work and in relatively large units. It is out of the question to consider making Diesel engines of much less than 100 hp. per cylinder, because of the complicated fuel-injecting mechanism and the high cost of production. The other types have the disadvantage of requiring external preheating before they can be started, and the torches used for this purpose are a source of constant danger. Electric preheating has been tried, but with little success.

12 There is an engine, however, which has all the advantages of the above-mentioned types and none of the disadvantages. This is the Hvid engine. It can be started cold on any liquid fuel which will flow through a pipe. It has no complicated air-compressor system for injecting the fuel, no hot bulbs or torches, and runs with a fuel economy on a par with the Diesel engine. The Hvid engine can be and is being produced in units as small as $1\frac{1}{2}$ hp., and so economically as to be able to compete with gasoline engines of the same size.

ADVANTAGES OF THE HVID ENGINE

13 A comparison of the Hvid engine of the farm type with a conventional gasoline engine of the same type discloses many factors which show the superiority of the former for this class of work. The Hvid engine has neither electrical devices nor carburetor or mixer. It starts readily on any liquid fuel, even in the coldest weather. On the other hand, the conventional gasoline farm-type engine has electric ignition, which is frequently a source of trouble. It has a carburetor or mixer to be adjusted according to atmospheric conditions and quality of fuel, and finally the gasoline engine is very hard to start in cold weather.

14 Briefly enumerated, the chief advantages of the Hvid engine are:

a Mechanical simplicity
b Low fuel consumption at all loads
c Ability to start and run on any oil which will flow
d Low water-jacket losses
e No lubricating difficulties
f Constant compression
g Remarkable torque characteristics
h Absence of all electrical devices, hot bulbs and torches for ignition purposes
i Absence of all carbureting mechanism
j No carbon troubles.

15 The Hvid engine is of conventional four-cycle type, embodying the usual inlet and exhaust valves, timed to open and close as in any four-cycle engine. The compression pressure is carried to between 425 and 475 lb. per sq. in., which heats the compressed air to between 900 and 1000 deg. fahr. In the cylinder head there is a fuel-admission valve terminating in a small steel cup by means of

which a preliminary explosion is made to force the fuel into the combustion space. Referring to Fig. 2, the Hvid cycle is as follows:

16 *Suction Stroke.* During the suction stroke pure air is ad-

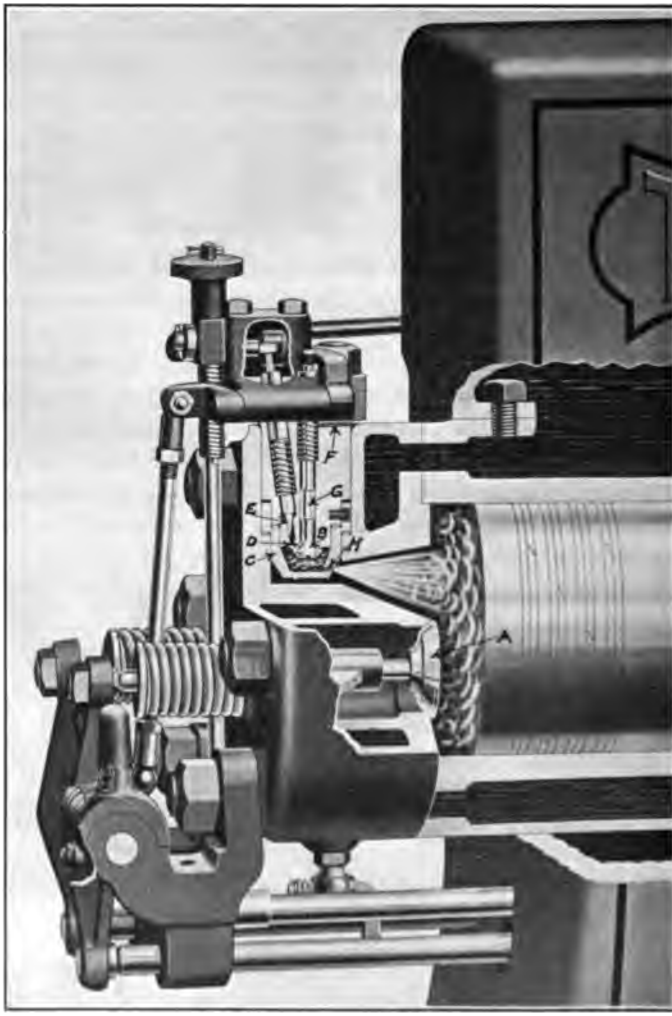


Fig. 2 Cutaway Section of 8-hp. Hvid-type Farm Engine

mitted to the cylinder through intake valve *A*. Fuel valve *B* is opened in synchronism with intake valve *A* and some fuel flows into cup *C* out of hole *D* which is uncovered by the opening of valve *B* (the fuel enters cup C partly by gravity and partly by inhalation). The amount of fuel admitted is controlled by the metering pin E, which in turn is controlled by the governor. At the same time that the fuel is being inhaled into the cup, a small amount of fresh air is also drawn through an auxiliary air hole F, down past a fluted guide G into the cup C. At the end of the suction stroke, fuel valve B and air-intake valve A close, valve B sealing the fuel-admission hole D.

17 *Compression Stroke.* During this stroke all valves are closed and the air admitted to the cylinder on the suction stroke is compressed to about 420 lb. per sq. in., which raises its temperature to between 900 and 1000 deg. fahr. In other words, there is now a

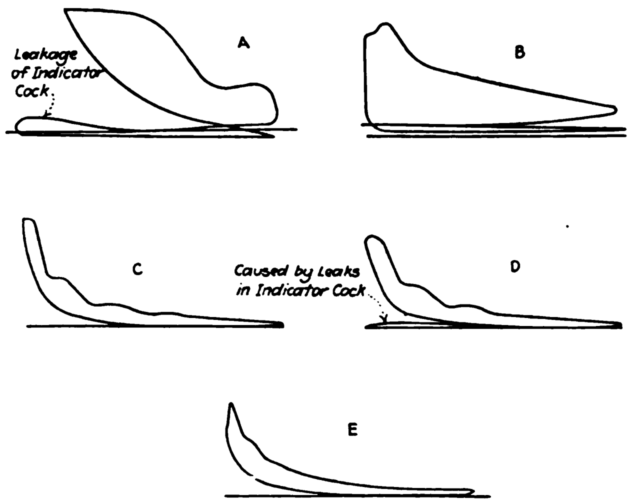


FIG. 3 INDICATOR CARDS TAKEN FROM HVID ENGINES

mass of highly heated air under high pressure in the combustion chamber and this rushes into cup C through small holes H near its bottom until the pressure in the cup is practically equal to the pressure in the combustion chamber. The conditions in the cup are now most favorable to "cracking" the oil, and as the oil cracks the lighter and more volatile components are detonated by the high temperature and the resultant high pressure within the cup forces the rest of the oil out into the air in the cylinder. The amount of fuel consumed in the cup per cycle is infinitesimal because there is only a very small amount of air present in the cup to support combustion.

18 *Power Stroke.* As the fuel in an atomized and vaporous

state comes into contact with the heated air in the combustion space, very rapid combustion takes place and the pressure arising from it drives the piston.

19 *Exhaust Stroke.* As in any four-cycle engine, the exhaust valve opens and the products of combustion are forced out by the piston.

INDICATOR CARDS

20 Fig. 3 shows a series of indicator cards taken from the 8-hp. Hvid engine of Fig. 2, as well as one from a 20-hp. engine of the same type. Descriptions of these cards follow.

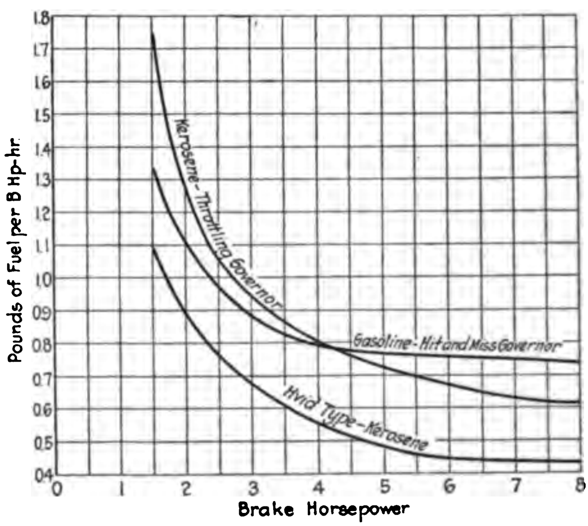


FIG. 4 COMPARATIVE FUEL-CONSUMPTION CURVES OF 5¾ × 9-IN. FARM-TYPE ENGINES OPERATING ON DIFFERENT PRINCIPLES

21 Card *A* is a no-load, stop-spring card which shows a slight vacuum during practically the entire suction stroke and figures approximately 375 ft. per sec. through the inlet orifice; at the same time it shows good volumetric efficiency. This vacuum is maintained intentionally for the reason that there is left in the injector cup, after the expansion and exhaust strokes, a slight residual pressure which interferes with the regular delivery of fuel by gravity. Dropping the suction pressure slightly below atmospheric removes this cup pressure, gives a slight pull on the incoming fuel and makes for close governing.

22 Card *B* is one taken from the inside of the injector cup

with the main air-intake orifice not choked and shows a residual pressure in the cup even during the suction stroke of the engine. One of the greatest difficulties encountered in early Hvid engines of the farm type was consistent governing, and it was not until automatic inlet valves were discarded and the inlet-air velocity was maintained constant that good governing was possible. Card *C* is a full-load card and card *D* is a 25 per cent overload card.

23 Card *E* is a full-load card from an 8½-in. by 10-in. 20-hp. Hvid engine in which the ignition point is much more clearly defined than in the cards taken from the 8-hp. engine. There is no doubt that ignition takes place at relatively the same point on the smaller

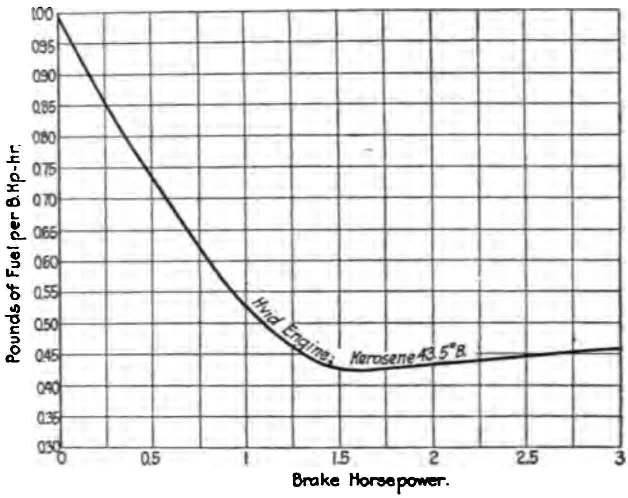


FIG. 5 FUEL-CONSUMPTION CURVE OF A 3 × 4½-IN. HVID ENGINE

engines, but the speed of the smaller engines is so much higher that the ordinary indicator fails to show it.

24 It is interesting to note the waves in the expansion line of these cards. These are typical of all cards taken from Hvid engines and are caused by the introduction of fuel into the combustion chamber in waves. At the moment when the preliminary combustion takes place in the cup, we have in the cup a pressure of approximately 800 lb., while in the combustion chamber there is a compression pressure of only 425 lb. Some fuel is consequently sprayed out of the cup into the combustion chamber by the attempt at pressure equalization. When the fuel comes in contact with the highly heated air in the combustion chamber, the pressure rises

in the combustion chamber and falls in the cup until equalized; then no more fuel can get out of the cup until the piston moves forward and the pressure in the cylinder drops below that in the cup, when some more fuel is ejected from the cup. This is repeated until there is no more fuel left in the cup.

FUEL CONSUMPTION

25 The fuel consumption of small Hvid-type engines is very good, being in general on a par with Diesel engines of large size. If Diesel engines could be economically constructed in units as small as Hvid engines can, it is doubtful whether they would compare at all favorably in thermal efficiency with the small Hvid units on account of the mechanical inefficiency of the air compressors necessary to inject the fuel. In the comparative fuel-consumption curves shown in Fig. 4 the fuel economy of the Hvid engines as compared with two other types of the same size stands out very plainly, particularly at the lower fractional loads. The "hit-and-miss" gasoline test in Fig. 4 was made by Professor Dickinson of the University of Illinois, the throttling-governor kerosene test by Mr. MacGregor of the Hercules Gas Engine Company, under the supervision of Government experts, and the Hvid-engine test by the writer under the supervision of Professor Roesch of Armour Institute. The engine used in each case was a $5\frac{3}{4}$-in. by 9-in., running 450 r.p.m. and rated at 8 hp.

26 The fuel-consumption curve of the small 3-in. by $4\frac{1}{2}$-in. Hvid engine shown in Fig. 5 is particularly interesting because this engine, running at 1100 r.p.m. normally, is the first relatively high-speed engine of this type built. When it was first designed the writer was very skeptical as to the results to be expected from it, because, owing to the high speed, it was natural to suppose that trouble would be encountered with the time element necessary for the introduction of fuel into and ejection out of the cup; but it was found that this little engine could be run at speeds as high as 1500 r.p.m. without any apparent interference with the perfect operation of the Hvid principle. In fact, the faster it was run the better the results. Based on its performance several engineers connected with the production of Hvid engines raised the normal r.p.m. of their engines with very beneficial results.

27 Professor Roesch also supervised other tests made to determine the entropy diagram and the heat balance, while Mr. H. C.

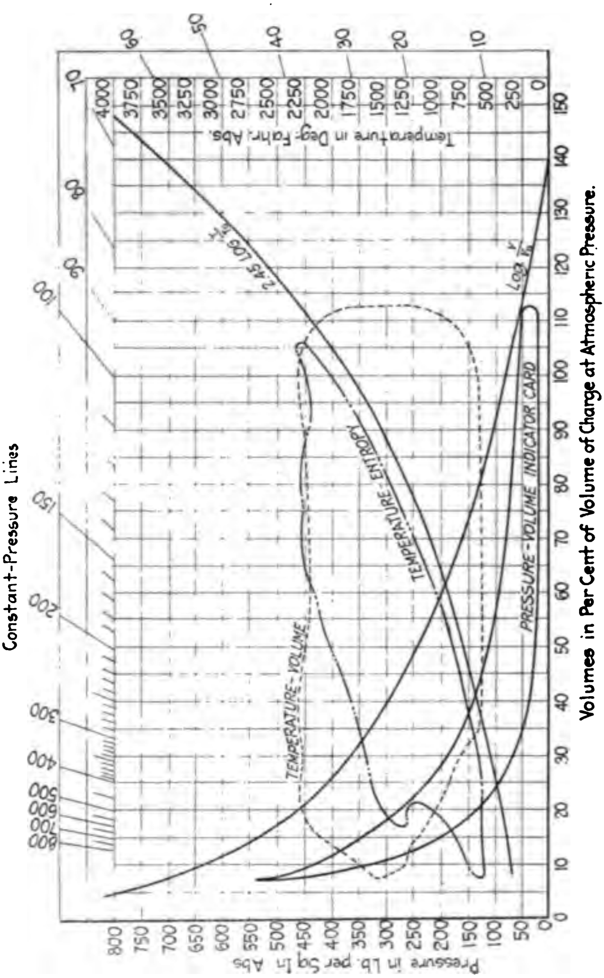


FIG. 6 VOLUME-TEMPERATURE DIAGRAM FOR A $5\frac{1}{4} \times 9$-IN. SINGLE-CYLINDER HVID KEROSENE ENGINE

Knudsen made tests for torque. It is worth while to record here the results of these tests.

28 *Entropy Diagram.* The entropy diagram, shown in Fig. 6, was plotted from a pressure-volume indicator card taken from a $5\frac{3}{4}$-in. by 9-in. single-cylinder Hvid-type engine running at 450 r.p.m. and using kerosene as fuel. This diagram is submitted because it shows the general temperature characteristics, which are quite different from those in an explosive gasoline engine. It is interesting to note the low maximum temperatures, 2300 deg. fahr. abs., as compared with the maximum temperatures for gasoline engines, which frequently run as high as 3000 to 3500 deg. fahr. abs., and also the sustained temperature in the Hvid engine throughout the working or expansion stroke. At first glance it might be

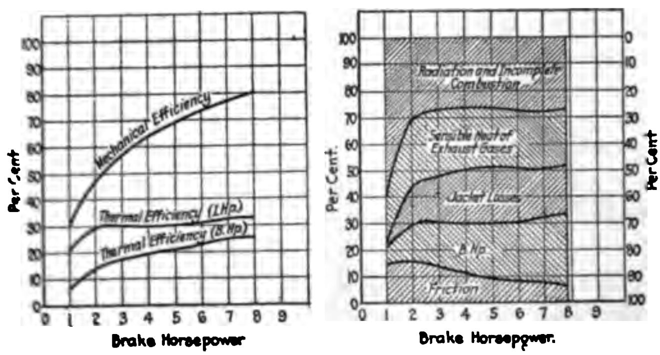


FIG. 7 HEAT BALANCE AND EFFICIENCY CURVES FOR $5\frac{3}{4} \times 9$-IN. SINGLE-CYLINDER HVID ENGINE

argued that this sustained temperature would be harmful because of undue heat losses to the water jackets, but since the water-jacket losses are low and the thermal efficiency of the engine is remarkably good, the writer believes that combustion in the Hvid engines takes place in the form of zone burning and that the cylinder walls are more or less insulated from the high temperatures of combustion [illegible] of air which is not burnt until near the end of the stroke.

29 *Heat Balance.* This test gives the mechanical efficiency [illegible] efficiency for both brake and indicated horsepower. [illegible] $5\frac{3}{4}$-in. by 9-in. single-cylinder Hvid engine [illegible] connected to a Sprague electric cradle dynamometer [illegible] "Spicer" universal joints. Engine speeds [illegible] by means of electrically operated [illegible] were also made for determining the

jacket-water loss and the sensible heat of the exhaust gases (calorimeter method). The developed and friction horsepower were determined by means of the dynamometer. Indicator cards were also taken, but because of the probable errors due to the high pressure involved and the comparatively high speed of the engine, these cards were used merely to study the valve settings and general events of the cycle, and not for indicated power measurements. The engine was operated under various loads and speeds with various adjustments of fuel supply, compression and cup design. The final setting was made with a compression of 390 lb. per sq. in.

30 Test runs, curves for which are shown in Fig. 7, were conducted at various loads from a maximum to about one-eighth of

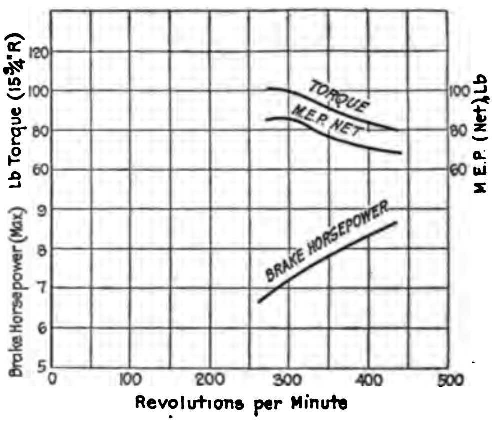


FIG. 8 BRAKE-HORSEPOWER AND TORQUE CURVES FOR A 5¾ × 9-IN. SINGLE-CYLINDER HVID ENGINE

maximum load, and readings were taken to determine the following:

1 Friction horsepower (electric-dynamometer method)
2 Brake horsepower (torque and speed)
3 Jacket-water loss
4 Sensible heating in the exhaust (calorimeter method)
5 Loss due to radiation and incomplete combustion (by difference)
6 Fuel consumption (lb. per hour).

Items 1 and 2 are determined directly from dynamometer readings; items 3 and 4 are calculated from observed temperatures and weights; item 5 is determined by difference; and item 6 is obtained from direct measurements. The heat value of the fuel expressed in B.t.u. per lb. of kerosene is calculated from the following accepted formula:

$$\text{B.t.u.} = 18{,}440 + 40 \times (\text{deg. Baumé} - 10)$$

and is 19,740 B.t.u. for the quality of fuel used in the test runs.

31 *Torque.* In Fig. 8 are shown some of the torque characteristics of the Hvid engine. When a gasoline engine of conventional design is overloaded so that the speed drops beyond a certain point its torque drops rapidly, because a certain velocity of air must be maintained through the carburetor to pick up and vaporize

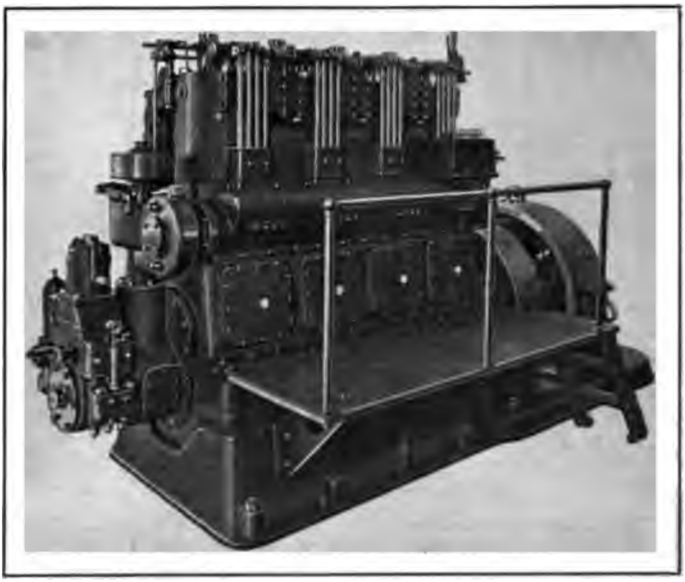


Fig. 9 100-hp. Hvid-Type Direct-Connected Generator Set

the fuel and carry it into the cylinder; but in a Hvid engine, since the introduction of fuel into the cup and into the cylinder is not dependent upon the velocity of the air taken in, as the speed drops, due to overload, more fuel is admitted than at normal speed, because the time element for the introduction of fuel is lengthened and the engine consequently shows remarkable "hanging-on" characteristics. Under these conditions it is very wasteful of fuel without a doubt, but there are certain conditions where this "bull dog" characteristic is desirable even at the expense of fuel economy.

32 Referring to Fig. 10, Mr. Knudsen states that these values

were actually obtained on the test block but that he is of the opinion that under actual operating conditions the torque should not be allowed to climb quite so high for the lower speeds, the reason being that the engine smokes badly and takes too much fuel at speeds below 450 r.p.m. It would seem to be better practice to vary the opening of the fuel valve with the speed of the engine, so that a curve could

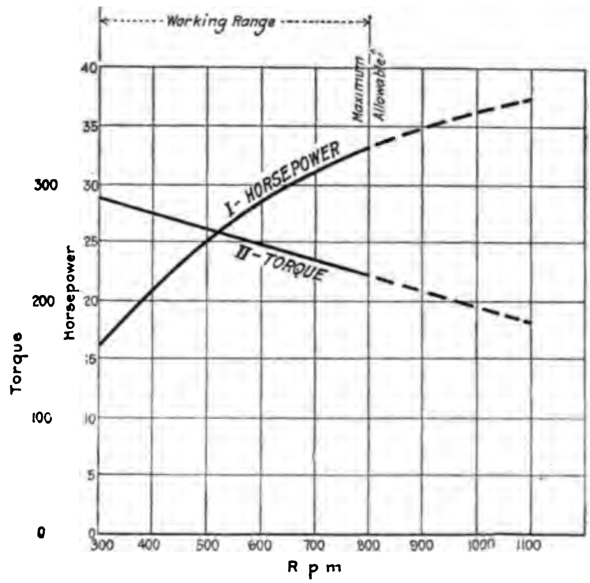


FIG. 10 HORSEPOWER AND TORQUE CURVES

The curves in this figure represent the absolute maximum horsepower and torque. The engine will develop continuously from 75 per cent to 80 per cent of these values.

be plotted, giving maximum torque at different speeds with a uniformly clean exhaust.

33 In conclusion, the writer would say that while the Hvid engine has by no means reached its ultimate state of development, it nevertheless possesses a number of wonderful characteristics which ought to attract many internal-combustion engineers by the possibilities they hold out of helping to solve some of our fuel problems.

DISCUSSION

H. SCHRECK. In the synopsis at the beginning of his paper the author states that the Hvid engine "has all of the advantages and none of the disadvantages of the so-called Diesel engine." This is a broad statement. Superiority over the Diesel engine is claimed by the author, but comparisons are made with gasoline and kerosene engines.

In Par. 14, the advantages of the Hvid engine, of which the author speaks are enumerated. However, all these points apply in the same measure to the Diesel engine, although points *b* and *g* are still more favorable to the Diesel than to the Hvid engine.

The statement in Par. 11 that the Diesel engine cannot be built in units of less than 100 hp. is disputed by the fact that European builders before the war were exporting Diesel engines of an output of as low as 8.5 hp.

A disadvantage of the Hvid engine compared with the Diesel is the preliminary explosion in the cup, with its uncertainty of operation, carbonizing of holes, etc.

In Par. 32 it is stated that below 450 r.p.m. the torque should not be too great as fuel consumption is increased for such lower speeds and the engine smokes badly. The high speed would preclude its use for marine purposes. A further disadvantage of the Hvid engine is that its regulation is more difficult than that of the Diesel engine. A Pacific Coast concern had attempted to build an engine of this type of 25-hp. per cylinder but had been unable to secure proper regulation.

Charles E. Lucke said that he had been familiar with the engine for some time as he had worked on it at one time with the author. He recognized that it was necessary to adopt a standard mean effective pressure and compression in a commercial engine, but that these pressures were not necessarily the best for all conditions. As a matter of interest to engineers he would like to have the author submit some figures showing the available mean effective pressures with various grades of oil at the most economical or rated load, and the most desirable compression pressures and the effect of load and grade of oil.

He said that the engine was of the Diesel type and that there was no reason for calling it anything else. It ignites by compression alone, without other aid, the combustion is more nearly non-explosive than explosive, the combustion lasts for an appreciable portion of the outward stroke, to judge from indicator cards, so that the engine is evidently of the solid-injection Diesel type.

The fact that this engine would ignite at lower temperatures than the air-injection Diesel type was perfectly reasonable. If air, compressed to 1000 lb., is allowed to expand to half that pressure, the temperature will decrease so that it is to be expected that compression in the engine must be carried much higher in order to allow for the cooling effect of the expanding injection air.

HENRY H. SUPLEE said that he was glad to hear Dr. Lucke pronounce the engine one of the Diesel type. He called attention

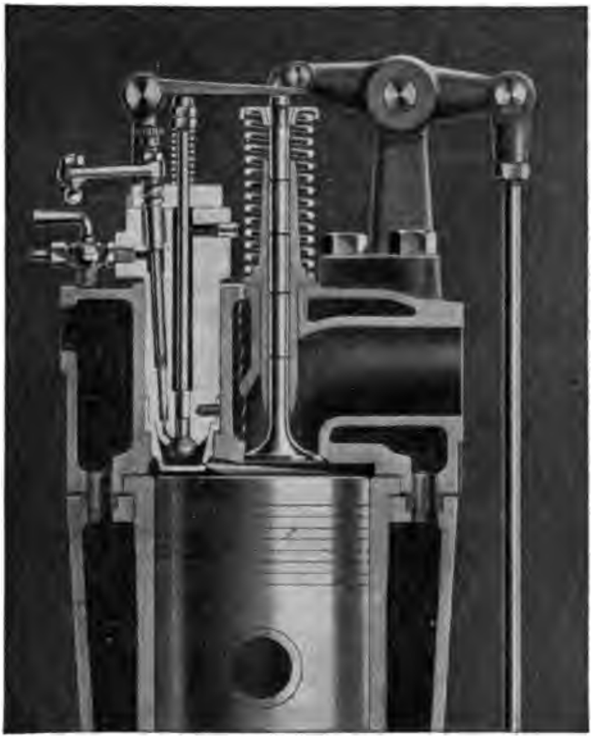


FIG. 11 CYLINDER HEAD SECTION SHOWING HVID SYSTEM IN VERTICAL ENGINE

to the number of engines of the Diesel type which were being built and experimented with in England and Europe.

THE AUTHOR, in his closure, and in answer to Mr. Schreck, again emphasized the point that the Hvid engine was much cheaper to manufacture than the Diesel with its air compressor. He had not intended to convey the impression that it was impossible to build small sizes of Diesel engines but that the costs were excessive. He further explained that his comparison of the Hvid engine with the Diesel had been made because both operate on the same principle, and that comparison with gasoline and kerosene engines had been made because it is with engines of these types that the Hvid engine is designed to compete. The difficulties of regulation were great, but not insuperable, as Hvid engines were now running with as good regulation as any engine with which it might compete.

In answer to a question by Henry H. Suplee regarding the necessity of supplying cups for different grades of fuel, the author replied that the volume of the cup and the size and number of the holes in it affect the operation of the engine with various grades of fuel. A single cup serves for various grades of fuel. The same cup, for instance, can be used for fuels of from 43 to 35 deg. B. With fuels as low as 33 deg. B this same cup can be used but the results will not be as satisfactory, while from 30 down to 26 deg. B. it is necessary to change the cup and the size of the holes. The heavier the oil, the smaller should be the size of the holes and the larger the volume of the cup. The heating surface is also smaller in cups for heavier fuels.

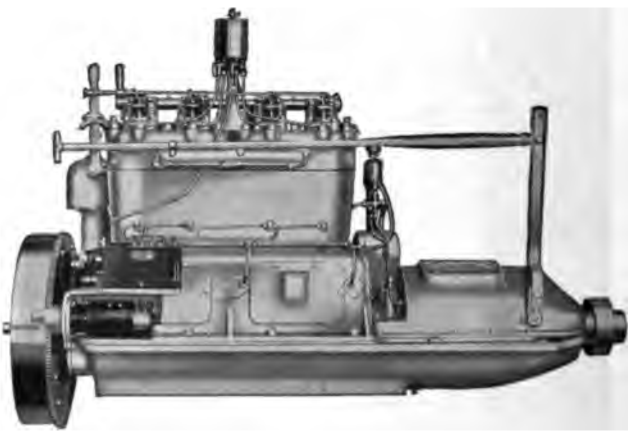


FIG. 12 A FOUR-CYLINDER ENGINE OF CONVENTIONAL DESIGN (CYLINDERS CAST "EN BLOC")

In answer to a question by George N. Van Derhoef about the flexibility of the engine under varying speeds and torque the author referred to the performance of a 30-hp. four-cylinder unit in which the torque at low speeds was remarkable. The only disadvantage compared with the conventional type of engine was that acceleration was not as rapid. It would pick up under load more evenly, however. The reason was obvious, — that in the conventional type of gasoline or kerosene engine the velocity through the carburetor was reduced at low speeds, while with the Hvid engine, running slowly and with metering pin open, more fuel was admitted and greater torque resulted.

Compression pressure had been run as low as 280 lb. with kerosene in the Hvid engine, the author stated in answer to a question by John R. Du Priest. By experiment the compression pressure for commercial engines using kerosene had been set at 425 lb. and at 470 lb. for heavier fuels. The engines are built with adjustable connecting rods so that the purchaser may increase the compression of his engine, and they are also supplied with an extra cup so that he may use heavier fuels if he desires.

Asked by H. P. Porter about the sizes of Hvid engines, the author replied that the standard farm types were 1.5, 3, 6 and 8 hp. Some larger single-cylinder horizontal units developed 15, 20, 30 and 60 hp. per cylinder. One marine engine concern was building the engine in 2-, 4- and 6-cylinder units, all with 8.5 by 12 cylinders. Another marine engine concern puts out a four-cylinder, 25-hp. unit with electric starter. Still another company was making four-cylinder 100-hp. units.

With reference to the starting of small Hvid engines up to 12 hp. the author said that the inlet valve was held open to relieve compression, the fuel turned on and the engine cranked rapidly by hand. The mechanism holding open the inlet valve was suddenly tripped, and the inertia of the flywheel was sufficient to carry the engine over one compression which was sufficient to start it.

In answer to a question by E. D. Thurston, Jr., who asked if there was trouble when starting the engine of a sudden explosion of the entire charge, the author stated that such trouble had been experienced only when the fuel had been allowed to escape from the cup into the cylinder before starting. It was impossible to use an automatic inlet valve on the engine because if it should stick, fuel would be pulled through the cup into the cylinder with a resulting detonation which might wreck the engine.

The author replied to Henry E. Whitaker that the regulation of the farm type of engine was within 5 per cent.

To Dean Benjamin, who asked about the stoppage of the holes in the cup due to carbonization, the author replied that this would happen if heavy oil was used in a cup designed for a lighter oil or if there was incomplete scavenging, when a small amount of fuel might remain in the cup and coke. The location of the holes in the cup with relation to the bottom and with respect to each other had a very marked effect upon keeping the cup clean. It was discovered that with holes parallel to each other and to the bottom, and located just above the bottom of the cup, scavenging of the cup would be complete and there would be no building up of carbon deposit.

No. 1725

COMBUSTION OF HEAVIER FUELS IN CONSTANT-VOLUME ENGINES AND IN ENGINES OF THE SUPER-INDUCTIVE TYPE

By Leon Cammen,[1] New York, N. Y.
Non-Member

Well-recognized liquid-fuel conditions make it imperative to prepare for the demand for engines capable of running on fuel heavier than gasoline, such as paraffin gas oil and similar products. The cardinal element governing the design of such engines is the rate of combustion of the fuel-air mixture, and since oil-air mixtures have a much lower basic rate of combustion than gasoline, conditions have to be created in the cylinder which will accelerate the combustion. One method of doing this, suggested in the paper, is by the use of svperinduction, and an installation is described showing its application and formulæ for power output are deduced. The design of kerosene engines and the carburation of kerosene and similar fuels are discussed in some detail.

IN considering the fuel supplies of the world it must be borne in mind that the Russian fields are definitely out of operation and may stay out for a number of years. The Rumanian fields have been very materially weakened by the destruction wrought by British engineers during the retreat of the Allied forces from Rumania, and the Galician fields, although injured less than the other two, suffered quite extensively during the many changes of the first half of the war. The loss from these three sources constitutes roughly, on the basis of pre-war statistics, one-third of the world's production of oil.

2 On the other hand, the demand for liquid fuel for automotive purposes has grown, and is still growing, by leaps and bounds. In addition to the millions of motor cars and trucks crowding the highways, two great new consumers of fuel have appeared, aeroplanes and tractors, and it is a fact but little known outside of the trade

[1] Associate Editor, Mechanical Engineering.

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

that this year more tractors will be built in the United States than trucks.

3 The world is therefore between the nether millstone of decreasing fuel supplies and the upper millstone of a rapidly-growing consumption and apparently the only way out is to improve our engine in one, or both, of two directions: first, making it more economical as regards fuel consumption; and second, by enabling it to use lower grades of fuel, and especially such grades as can be produced in larger quantities than gasoline.

STATUS OF CONVENTIONAL ENGINE

4 The conventional motor-car or aeroplane engine belongs to the constant-volume type, and in the vast majority of cases is of the four-stroke cycle. With very few exceptions it is of the stationary, vertical, V, or radial-cylinder type, water-cooled, of fairly low compression, and with speeds varying from about 1000 to close to 3000 r.p.m., and with constant-stroke pistons.

5 This engine has reached such a state of development that when operating on gasoline or fuels vaporizing at low temperatures (under 180 deg.) only slight improvement may be expected. In flexibility, volumetric efficiency, power output and fuel economy the modern high-grade automobile or aeroplane engine has nearly reached the limit of perfection for the given type; and it is doubtful if the work necessary to develop them would be well spent in view of the fact that what the world needs today is not a better gasoline engine, but an engine which will run better on a fuel costing about one-half to one-third of what gasoline costs or is likely to cost in two or three years.

KEROSENE ENGINES

6 The first effort to meet this situation was to adapt the conventional engine to the use of kerosene. The problem of running a motor-car engine on kerosene seemed to be comparatively simple. Marine engines have been operating on that fuel for years with good success. But the pioneers in this field did not always realise that the marine engine runs at comparatively low speed and with fully open throttle and constant load, while the motor-car engine has to meet far more difficult and varied conditions of operation. The struggle was a hard one and it is still somewhat doubtful whether the aim has been definitely attained.

7 In order to understand the difficulties encountered one must consider the processes in an engine of the constant-volume type such as now used in automotive equipment. Practically all the mixing of fuel and air is done outside of the engine cylinder and the charge is ignited by an electric spark at one or (at most) two spots. The compression is moderate, even at the maximum filling of the cylinder, and at low throttle it is very small, so there is but little chance for the kerosene to vaporize in the cylinder. All the vaporization that occurs must take effect *and be maintained* previous to the mixture reaching the cylinder. It must also be borne in mind that explosive combustion, the only kind which generates power in an engine cylinder, is produced only by an extremely intimate mixture of fuel and air, in which there are no layers or large drops of liquid fuel.

8 With these facts in mind, and the further fact that in a conventional engine comparatively low temperatures prevail in the carburetor, air intake to carburetor and engine manifold (not in excess of 150 deg. fahr.), we may divide all liquid fuels into two groups. In the first will be such fuels as 68- to 73-gravity gasoline, benzol and products of hydrogenation like cyclohexane, fuels which vaporize at moderate temperatures, and which, once vaporized, do not condense until subjected to comparatively low temperatures, say, 60 deg. and under. These are ideal fuels for the conventional engine. To the next class belong fuels which do not vaporize at these temperatures, such as kerosene, or vaporize only to a very small extent, as paraffin gas oil. It is obvious that with an ordinary carburetor these fuels will not form an intimate mixture with air, since the conventional carburetor relies for this on the ability of the fuel to vaporize.

PREHEATING THE FUEL

9 The designers of kerosene engines have therefore resorted to two methods, either singly or in various combinations. The first and most obvious has been preheating the fuel. Kerosene is a product of distillation, and was once a gas. It seemed reasonable to expect that if it should again be made a gas, as by the application of sufficient heat, it would mix well with air and give perfect conditions of combustion.

10 This would be correct if it were not for the particular conditions which exist. In the first place the velocity of reaction in

the case of distillation of kerosene in a still and its vaporization in a carburetor, hot-spotted manifold or vaporizer are vastly different, and in the case of complicated hydrocarbons, such as the higher-oil series, the velocity of reaction is of great importance. From the work of Kharitchkoff, and to a certain extent that of the writer, it is safe to say that kerosene vâpor from a still and the vapor from a vaporizer are two very different products.

11 It should also be noted that vaporization in a still occurs without the presence of air, while in the hot-spot manifold or carburetor with vaporizer the air is not only present, but is brought to the temperature of the fuel vapor. It is known, however, that the higher hydrocarbons, such as octanes, nonanes, and up, when hot and in the presence of air, are apt to break up into the lower series, depositing some of their carbon as coke, and liberating hydrogen. There is also a suspicion (although, as far as the writer is aware it has never been directly proved) that products of the acetylene series are formed, especially when catalyzers of the proper character are present (in this case iron of the manifold may act as such a catalyzer).

DETONATION IN KEROSENE ENGINES

12 The presence of free hydrogen and possibly acetylene in the products of vaporization of kerosene is of great importance as it lies at the bottom of one of the troubles with kerosene engines, that is, the so-called phenomenon of detonation, sometimes confused with preignition.

13 Under certain conditions, especially when operated at full power, kerosene engines develop a characteristic knock, somewhat similar to that due to preignition in gasoline engines, and leading to a rapid falling off in power developed by the engine. At first it was assumed that it was preignition, possibly due to overheating of spark-plug terminals; but recent investigations, especially those of C. F. Kettering and H. L. Horning in this country, have established that the rapid rise of pressure, which produces the knock, occurs several crankshaft degrees *after* ignition, which of course makes it impossible to believe that it is due to preignition.

14 The writer carried out a rather interesting test apparently pointing to the cause of the knock. He added to trade gasoline about 3 per cent of water and passed the whole, previous to admission to the carburetor, through a small container filled with calcium carbide. This added to the mixture in the cylinder a small amount

of acetylene, and developed a knock which occurred in the same manner as in some kerosene engines.

15 There is an indication, that the "kerosene knock" is produced principally by hydrogen. If this is the case, the knock could be eliminated by introducing into the fuel a material that would easily dissolve in kerosene, vaporize at about the same temperature as kerosene, and, what is most important, combine directly with hydrogen. Such a material would then form a combination with hydrogen, either inert to oxygen or burning at a much slower rate than the kerosene-air mixture.

16 It so happens that iodine satisfies all these conditions, and, as a matter of fact, a small addition (1.5 to 2 per cent) of iodine to kerosene does away entirely with the knock. This would support the theory that the kerosene knock is due to the excessive rate of combustion, which in its turn is due to the presence of such materials in the fuel as hydrogen.

17 To understand it, one must clearly visualize the process of combustion in an engine of the constant-volume type. At the end of the compression stroke (assuming maximum volumetric efficiency) the cylinder is filled with a combustible charge highly-compressed and therefore heated to a temperature fairly close to the point of self-ignition (which is higher for gasoline — about 630 deg. — than for kerosene — 570 deg.). The spark ignites the charge at one spot and the combustion thus started acts in two ways: first, the gases generated occupy a greater volume than before, and thus raise the pressure and hence temperature in the rest of the cylinder; and second, the heat of combustion of the gases tends to emphasize this process by raising the temperature of the gases still further.

18 Combustion in an engine cylinder usually takes place in a fairly leisurely manner, and is completed only by about the time when the piston reaches its lowermost position. (Essentially, therefore, there is no difference between the nature of combustion in engines of the constant-pressure and of the constant-volume type.) In fact, even in gasoline engines, with very lean mixtures, the rate of combustion may be so slow that it is still proceeding when the inlet valves are opened for the next cycle, which produces what is known as "popping into the carburetor."

19 This, however, according to Hofsaess, is only the process with slow-combustion mixtures, such as gasoline-air or kerosene-air.[1]

[1] Thesis at the Technical School, Karlsruhe, 1913, quoted by Engler in *Erdoel*, vol. iv, p. 540, in which it is stated that the velocity of flame propagation for methane-air mixture is 27.5 cm. per sec.

But if in the same cylinder there are present gases forming with air mixtures of higher rate of flame propagation, they will ignite at their own rate. According to the same source as above, the rate for hydrogen is 200 cm. per sec. and for acetylene 113.5, or several times that of methane-air mixtures. Hence, when either hydrogen or acetylene are present in the mixtures, they ignite much more rapidly, and ignite the rest of the mixture at the same rate as their own ignition. In other words, they serve as a primer for the rest of the mixture, or as hundreds of spark plugs scattered through the mixture. It is extremely important to understand this process very clearly, as it explains some of the troubles encountered in operating kerosene engines.

20 It appears, therefore, that at best high preheating of kerosene does not promise to give a complete solution of the problem, even assuming that the other difficulties connected therewith have been solved, one of the most important of which is the decrease of the volumetric efficiency of the engine due to increase in temperature of the incoming charge.

FINE VAPORIZATION OF THE FUEL

21 The other way along which a solution of the problem of burning kerosene has been attempted, has been by means of extremely fine vaporization of the fuel. There have been two considerations underlying this idea.

22 The first was to burn kerosene with air, not as a mixture of gases, but somewhat in the same manner in which explosions of coal and grain dust take place; that is, by converting the kerosene into such a fine powder, or fog, as to insure a very intimate contact between it and the air. From what we know of the mechanism of dust combustion, one can easily believe that such a fog mixture will explode when raised to the proper temperature. In that case preliminary heating of the fuel outside of the cylinder becomes unnecessary, the main heating of the charge being secured through its compression in the engine cylinder, although the air may usefully be preheated to a certain extent. This system, where a thorough mixing of the fuel and air is produced by the mechanical atomization of the fuel exclusively, would obviously be inefficient unless the atomized fuel were delivered by the most direct path into the cylinder, as the best distributing manifold has enough corners, loci of turbulence and loss of velocity head to produce coalescence of the

kerosene globules. It requires, therefore, an atomizer in each engine cylinder which, although correct theoretically, may prove to be too complicated to find a universal commercial application. Nevertheless the principle of the construction is sound, as has been proved by the tests carried out under the auspices of the Automobile Club of France by Bellem and Brégéras in the past year.

23 The other consideration underlying the application of atomization to the production of explosive mixtures with kerosene as fuel is based on the principles of vaporization. It has been found that if kerosene in a very fine state of atomization is maintained even for a very short time at a temperature of from 225 to 240 deg. fahr., it will pass to a state of gas, even though its final point of distillation be in the neighborhood of 530 deg. fahr., and even though its initial point of distillation be at 275 deg.

24 Tests by the writer at an early date, however, indicated that to secure this result the atomization of the kerosene must be very fine indeed as obvious from the fact that the time available for the process is extremely small. In an engine running at, say, 1800 r.p.m., the entire process of suction and compression takes less than $\frac{1}{30}$ sec., in which time the drop of kerosene has to vaporize.

ATOMIZATION OF KEROSENE IN ORDINARY CARBURETORS

25 When it comes to the practical atomization of kerosene, it is found to be by no means an easy matter. Tests made for viscosity on a number of specimens of gasolines and kerosenes of American origin have shown an average viscosity for gasoline of 0.7, and for kerosene of 1.5. In addition to this, kerosene has shown a surface skin at least four times as tough as gasoline.[1]

26 In ordinary carburetors atomization can be produced only by the difference in pressure existing between the mixing chamber of the carburetor and the float chamber, and to a certain extent by the entrainment produced by a column of air rapidly flowing past the carburetor nozzle. The energy thus available is too small to produce perfect atomization unless extremely fine nozzles are used, which led to a series of tests on very fine nozzles.

27. A number of carburetors were built. The first had fine orifices (jewelers' drill No. 10 and even smaller) drilled in copper plates $\frac{1}{64}$ in. thick, this size having been selected as the drills would

[1] Tests by the writer; cp. Engler-Hoefer, *Erdoel*, vol. iv, p. 51.

not work in thicker plates. A very poor atomization followed, and it was found that instead of being broken up into a fine spray as was expected, the kerosene simply passed through the orifices and spread over the face of the plate. It was then that the tests previously referred to were started to determine the surface tension of kerosene as compared with gasoline, and its high values were discovered. Obviously, the thin plate did not offer enough resistance to the flow of the liquid to break the skin.

28 A number of new carburetors were then constructed, with the nozzles formed between adjacent plates, as in Fig. 1, the passages being formed by making scratches on the face of the plate. This permitted the size of the passages to be varied as desired, both as to length and depth, and resulted in an immediate improvement in the operation of the engine to which the carburetor was attached.

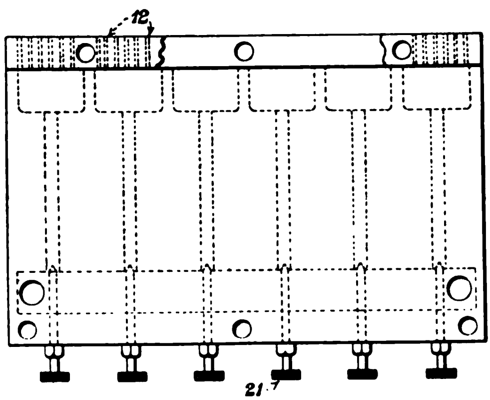


FIG. 1 CARBURETOR NOZZLES BETWEEN FACES OF ADJACENT PLATES

29. The tests indicated that good atomization is secured only when a large number of nozzles are used, the number in the tests on a four-cylinder, 3.75-in. by 5-in. engine having varied from 74 to 400. About 300 nozzles gave the best results.

30 As regards the dimensions of the nozzles, it was also found that there are certain critical dimensions giving the best results. Here again a number of tests were made with plates of different thicknesses and nozzles of different depths, and it was found that on the whole the best results were secured with nozzles about $\frac{1}{8}$ in. deep, and shaped as a semicircle 0.003 to 0.004 in. in diameter.

Later tests have brought out a rather interesting fact, namely, that the arrangement of 300 nozzles of the above dimensions which gave good results on kerosene proved to be very poor for use with gasoline, while six to ten larger orifices, say, 0.1 in. in diameter, gave good results with gasoline and very poor results with kerosene. Just why gasoline operates so poorly with the very small nozzles seems to be a matter far more interesting than would appear at first sight, and study of the problem will apparently lead to the establishment of some rather novel relations in connection with capillary flow. Tests are under way on this subject.

ATOMIZING KEROSENE NOT ENOUGH

31 But atomizing kerosene at the nozzle of the carburetor, even very finely, seems to be only half the battle. Other sources of trouble must be overcome before the mixture can be properly burned in the cylinder.

32 The first question is that of delivering the atomized mixture to the engine cylinder, which implies the design of proper manifolding. For gasoline engines the question of good manifolding may be considered as solved, but it is somewhat more involved when it comes to kerosene engines.

33 A gasoline-air mixture, when above the critical temperature, is a perfect gas and is not affected as to state by the shape of the manifold, presence of bends or factors creating turbulence. A poor manifold may affect the volumetric efficiency of an engine, but not the condition of the mixture. With kerosene, however, a poor manifold affects the condition of the mixture. If rapidly moving fine drops strike against a bend in the manifold, the tendency is for them to adhere to the wall of the manifold against which they impinge. To determine this factor, a number of tests were made on the same engine with various manifolds, all the other conditions being equal; in some cases long and crooked manifolds were used, while in other tests the carburetor was placed right against the body of the engine. In order to maintain all the conditions as uniform as possible, the manifolds were in all cases carefully heat-jacketed and lagged with asbestos so as to avoid excessive cooling of the mixture with the long manifolds. The running of the engine, power curve, and especially the condition of the interior of the manifold and the cylinder after a run clearly showed the great amount of condensa-

tion that takes place with manifolds which offer obstructions to the flow of the mixture. For example, a straight, long manifold has been found to be far preferable to a short one with one or two bends. The best was one which permitted the location of the carburetor flush against the cylinder block.

34 There still remained, however, a number of crooks and bends in that part of the manifold comprised in the cylinder-block casting itself, and it soon became apparent that that section of the manifold had to be redesigned, as well as the exterior portion, in order to secure the nearest approach to a straight-line flow. In other words, kerosene-engine manifolds have to be designed from

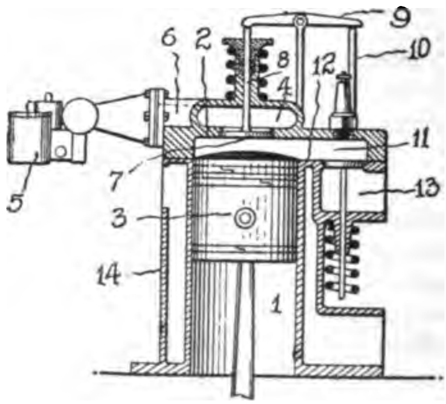


Fig. 2 Engine Cylinder with Inlet Manifold Cast into the Head

end to end with an eye to some relation like the Fanning formula. The designs shown in Figs. 2 and 3 have been adapted to take care of these conditions. The carburetor sits right at the cylinder head, the manifold is cast into the cylinder head, and the mixture flows in a straight line from the mixing chamber of the carburetor to the valves, the only turn being made at the valves where the temperature conditions are such that condensation need not be feared.

TEMPERATURE RELATIONS IN THE MANIFOLD

35 As has been stated above, kerosene will vaporize if heated to about 225 deg. fahr. when in the state of fine atomization (this figure is still somewhat uncertain, and possibly depends on the kind of kerosene used). On the other hand, if finely atomized and heated

to a very much higher temperature, the kerosene easily breaks up and causes a particularly vicious knock, as found in many runs made under this condition.

36 The problem is therefore to bring the kerosene-air mixture to a temperature of approximately 225 deg., or as close to it from below (i.e., without exceeding it) as possible. Preheating the mixture by exhaust gases was tried, but did not prove successful for two reasons: in the first place, notwithstanding numerous attempts, the writer failed to develop a design of an exhaust-gas heater which would not involve a very large frictional resistance to the flow of the mixture with attendant condensation. In fact, it was found on several occasions that with the walls of the preheater in excess of 300 deg., considerable kerosene settled in the tube. So far no means had been found to control the gas temperature in the preheaters

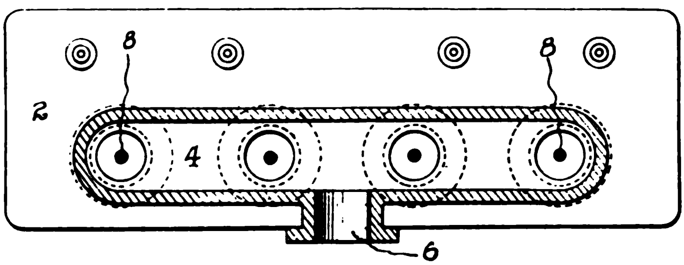


FIG. 3 INLET MANIFOLD FOR FOUR-CYLINDER ENGINE OF FIG. 2

which made for an irregular operation of the engine. Because of this, the entire preheating of the kerosene-air mixture was thrown on the manifold itself, this being done in the following manner:

37 As shown in Fig. 2, the walls of the cylinder were liquid-cooled. The head had no jacket and was cooled partly by radiation, partly by the flow of gases through the manifold, and to a certain extent by the flow of heat through the gaskets into the cylinder walls. It was found that with kerosene as fuel and with the temperature of the outer jacket liquid at about 210 deg. fahr., the temperature of the head was in the neighborhood of 350 deg. fahr. which is just right to raise the temperature of the mixture flowing therethrough to about 225 deg. This gave a good mixture in good condition for use in the cylinder and all that remained was to properly burn it there, a problem easier stated than solved.

TEMPERATURE RELATIONS IN THE CYLINDER

38 A large number of tests were made, both on the block and on the road, to determine the best temperature of the cooling liquid for burning kerosene, as it was felt that this was one of the most important governing conditions. It was found that while the engine would operate after a fashion with a fairly wide range of temperatures, really good results, comparable with those secured with good gasoline, were obtained only when the cylinder temperatures were kept within pretty close limits.

39 In particular, it has been found that the temperature of the cylinder walls affects quite materially the state of the oil in the crankcase and the oil consumption. The following are some of the details of these tests.

TESTS ON CRANKCASE DILUTION

40 Possibly the greatest difficulty in operating with kerosene lies in the tendency to crankcase dilution. Because of this, an extensive series of tests were made to determine if possible the laws governing it. The amount of crankcase dilution in some of the tests was determined by the measurement of viscosity and flash point, and by distillation of the oil taken out of the crankcase. In other tests benzol was used as an indicator.

41 It was found (and these results have been confirmed by similar tests made by the office of the Director of Military Aeronautics, War Department) that the amount of crankcase dilution is directly related to the temperature of the engine, and is much greater in a cold engine than in a warm one.

42 An apparently new fact brought out by these tests is that, if the engine is properly operated, not only need there be no additional dilution when running on kerosene, but that part of the gasoline which may have slipped through into the crankcase when running on gasoline will actually be eliminated when later running on kerosene. In other words, operation on kerosene when the engine is under proper temperature control, does not necessarily lead to excessive crankcase dilution. This fact has been borne out in some of the road tests and appears to be of considerable importance.

43 A test run of 758 miles in seven days was made in Ohio, in the summer of 1919, the engine being started each morning on gasoline and shifted to kerosene as soon as the water in the radiator

was sufficiently hot (200 deg. fahr. for start, and at boiling point during running). The total oil consumption during this run was under half a gallon, the oil at the end of the run being in perfect condition.

44 This very low consumption of oil may be explained as follows: The engine was run on trade gasoline when cold, which caused a certain amount of crankcase dilution. Had the engine continued to run on gasoline, the fuel which got into the crankcase would have stayed there, gradually thinned the oil, and caused a loss, directly, by increased consumption because of the lower viscosity, and also possibly through the breather. But when the engine was shifted to kerosene it ran so hot that any gasoline that may have slipped into the crankcase at starting, evaporated, which brought the viscosity of the oil back to its original figure. The fact that kerosene is to a slight degree itself a lubricant, may also have contributed to the low oil consumption.

45 It might be said in passing that to a man accustomed only to gasoline engines the operation of an engine on kerosene will prove full of surprises. Thus, a gasoline-driven car will not run without water in the radiator. With kerosene we have on several occasions driven the car for distances as great as ten miles at from 20 to 25 miles an hour without water in the radiator — of course only on good level roads, as the engine would rapidly overheat if an attempt were made to drive it up hill without water.

TEMPERATURE RELATIONS IN THE CYLINDER (*continued*)

46 All these factors indicate that the cylinder-wall temperature in kerosene engines has to be somewhat higher than for gasoline engines, due, probably, to two causes. First, it is very likely that only a part of the kerosene "dust" is vaporized in the manifold, the time being too short to produce complete vaporization. The "dust," which is really a very fine spray, is then vaporized in the cylinder. (The possibility of this, as far as the writer is aware, was first indicated, indirectly, by tests by J. B. Replogle, of the Remy Ignition Company.) But such a vaporization involves the passage of a large amount of heat into a latent state, and therefore a higher temperature is necessary to maintain the mixture in an ignitable state.

47 The second and possibly more important reason for the necessity of higher cylinder-wall temperature in kerosene engines

is that the rate of combustion of kerosene-air mixtures, all else being equal, is a good deal slower than that of gasoline-air mixtures. This may be also expressed by saying that kerosene-air mixtures have a lower *basic* rate of combustion than gasoline-air mixtures. But the rate of combustion of a gaseous mixture depends on the temperature of the mixture, since obviously the nearer the temperature of the mixture to the point of self-ignition, the faster the mixture will ignite and the faster it will burn. A fairly cold gasoline mixture may still burn sufficiently fast to give good results, while a kerosene-air mixture at the same temperature will fail entirely.

RATE OF COMBUSTION A FACTOR OF CARDINAL IMPORTANCE

48 The entire design of the engine should be built around the rate of combustion of the fuel-air mixture used in driving the engine, a fact but dimly realized until we came to design engines for fuels other than gasoline. A good deal of trouble with kerosene engines has been due to the fact that, in the vast majority of cases, they have been designed for gasoline and not for kerosene at all. This in particular has been the case with those, fortunately less numerous today than, say, two or three years ago, who expected to be able to operate any engine on kerosene by attaching to it a "special carburetor."

49 It therefore becomes of primary importance to determine by what factors the rate of combustion in an engine cylinder is governed, apart from the physical properties of the explosive mixture itself.

TEMPERATURE OF CYLINDER WALLS

50 The most important of these factors is the temperature of the mixture, which is materially affected by the temperature of the cylinder walls, which, in turn, means temperature of the cooling fluid. Numerous tests under all sorts of conditions, both on the road and on the block, have shown that for kerosene the temperature of the cooling liquid has to be a good deal higher than for gasoline. Tests made at temperatures at the exit from the jacket ranging from 125 to 255 deg. fahr. have shown that the best results have been obtained with temperatures around 210–215 deg., which indicates the necessity of using oil for cooling kerosene engines; or, if water be used in engines other than of the stationary type with

hopper cooling, of resorting to means for the condensation of steam generated. This, by the way, explains the comparatively good results in recent public tests obtained with kerosene in tractor engines by several makers of tractors who use oil cooling and can therefore carry the outlet temperatures on their cooling jackets to higher levels than can be done with water in conventional radiators.

51 The question of the efficiency of the conventional radiators for oil cooling led to an elaborate series of tests on heat transmission from oil to water and oil to air, carried out for the writer at the Mechanical Engineering Laboratory of the Brooklyn Polytechnic Institute by two senior students, Ed. F. O'Reilly and A. R. J. Wiedmann, under the direction of Prof. Wm. J. Moore. These tests have shown that with oil inside and air outside the efficiency of a conventional radiator is only about 70 per cent of that of the same radiator with water inside, which means either a larger radiator or more complicated methods of cooling.

COMPRESSION AND RATE OF COMBUSTION

52 The next factor affecting the rate of combustion of the mixture in the cylinder is the compression available, which means a good deal more than the "compression ratio" so often taken alone into consideration.

53 From gasoline-engine practice we have learned that, within certain limits, the higher the compression ratio the higher the maximum power output of an engine, the limits being imposed mainly by the tendency of the mixture to preignite. But judging an engine solely by the compression ratio, like judging it by its volumetric efficiency, is apt to lead to error.

54 Compression of the charge is necessary because it raises the temperature of the mixture to a point so near its temperature of self-ignition that, once started by the electric spark, combustion proceeds very rapidly. This determines quite definitely the upper limit of compression in an engine as a function of the chemical properties of the fuel-air mixture and temperature and pressure at which it enters the cylinder. It determines also the most suitable compression pressure which an engine should have when designed to work at a given load and may be usefully employed in the design of marine, racing-car and to a certain extent aeroplane engines. It should be understood, however (which is not always the case),

that this determines only the upper limit of compression, a fact which has a good deal to do with the failure of some kerosene engines.

55 The temperature of self-ignition of optimum kerosene-air mixtures is about 570 deg. fahr., as compared with about 630 deg. for optimum gasoline-air mixtures. The obvious tendency, therefore, is to make the compression in kerosene engines lower than in gasoline engines, which is the proper course to take in the case of a marine engine or stationary engine operating on constant load. But in our tests on automobile engines where the compression, by means of inserts, was varied from 50 to 85 lb., and especially in the block tests, we have found that while low compression makes it possible to drive the engine somewhat harder before preignition starts, it also deprives the engine of its flexibility.

56 This is only natural. The maximum compression temperature is obtained only with the maximum filling of the cylinder, or at full throttle. At partly open throttle we have a partial filling of the cylinder, and therefore, while the compression ratio is the same, the final compression pressure is different and hence the final temperature of the charge is also different. In other words, *the final compression pressure depends not only on the compression ratio,* which is a constant for a given engine of conventional design (it is not so in a variable-stroke engine), *but also on the volumetric efficiency of the engine during the given cycle.*

57 Therefore, if a kerosene-air mixture ignites well at, say, 500 deg. fahr., and if this temperature is produced by the compression in a given engine at maximum filling of the cylinder, at half-filling, or half-throttle, the compression pressure will be roughly one-half, and the temperature of the mixture will be lower, in accordance with well-known laws.

58 But if combustion in a kerosene-air mixture proceeds at a certain suitable rate when this mixture is at an initial temperature of 500 deg., a much lower rate of combustion will be maintained when the temperature is, say, 375 deg., and the point may be, and actually is, reached when explosive combustion will not take place at all.

59 In fact, while the writer has not yet been able to prove it positively, all his observations lead him to believe that a moment is reached, especially with fuels of a grade still lower than kerosene (such as paraffin gas oil), when combustion continues until the opening of the inlet valves in the next cycle. With gasoline this would lead to "popping into the carburetor." With kerosene and

fuel oil the combustion at partial filling of the cylinder does not proceed with the same vigor, and the following little-suspected process takes place:

60 As an inlet valve opens, the incoming charge meets a certain amount of inert gases of combustion at the top of the cylinder; but when it reaches the bottom of the cylinder it encounters a naked flame resulting from continued combustion of the preceding charge. As the pressure in the cylinder is very low, the incoming fresh charge is comparatively cold, flows in gradually, and is mixed to a certain extent with inert exhaust gases — it burns up as it comes in, but not explosively.

61 In other words, during the induction stroke we have in the cylinder something akin to what happens in the combustion stroke in the Akroyd cycle. Only the trouble is that here the fuel is burned up during the induction stroke when it can do no good; and when the power stroke comes there is no more fuel mixture in the cylinder to drive the engine. The latter therefore dies out. In common parlance this is described by saying that the engine "does not idle well," or "lacks flexibility."

62 This trouble is fundamental and does not depend on the design of the carburetor. There are only two ways to overcome it. One is by varying the temperature of the induced charge, making it colder at full throttle and warming it up at low throttle, which to a certain extent is done, for example, in the cylinder construction shown in Fig. 2, where the charge at full throttle flows faster and does not get heated up as much as the slower-flowing charge at low throttle. In a different manner the same result is secured in the Dray kerosene shunt. This, however, is only a crude solution of the problem and does not meet all the conditions that are encountered.

63 The other method of meeting the conditions consists in using a variable-stroke engine of such a construction that the full stroke is used in connection with wide-open throttle and a reduced stroke when the throttle is partly closed. The writer is not aware that such an engine has ever been built and doubts very much the feasibility of such a construction, but its equivalent may be attained in a somewhat roundabout way.

64 After all, what we need is the same final compression of gases in the cylinder at all positions of the throttle in order to obtain the same heats of compression, which might be secured by filling the bottom of the cylinder when operating at low throttle with a cushion of either air or inert gas. This in effect would be equivalent

to a variable-stroke engine, as to all practical purposes the inert gas may be considered as a fluid part of the engine piston (subject, of course, to the maintenance of a perfectly stratified filling of the cylinder). If we could be sure that the exhaust gases admitted to the bottom of the cylinder do lie in a layer separate from the main charge, this would present a good solution of the problem.

65 For example, Fig. 4 gives one of the possible constructions meeting the above-established conditions. In this construction, exhaust gases cooled in the coil 13 are admitted to the bottom of the cylinder at 14 toward the end of the induction stroke, the opening of the passage being governed by the slide valve 10 and cam 16.

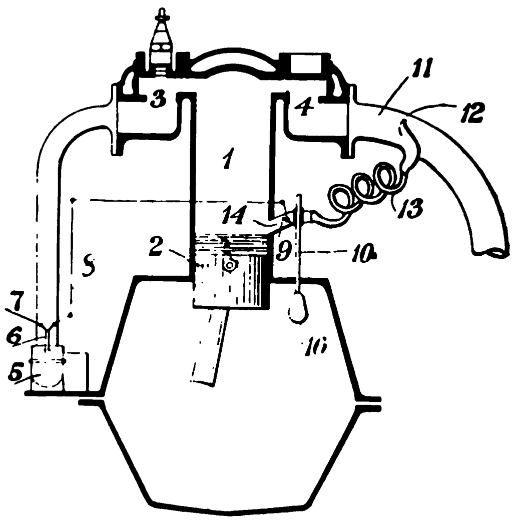


FIG. 4 SUPERINDUCTION ENGINE, MODIFIED DUGALD CLERK DESIGN

In addition to this there is in the passage a throttle 9 so interconnected with the carburetor throttle 6 that when the latter is fully open the admission of exhaust gases is cut off; and when the carburetor throttle is closed the exhaust gases are freely admitted. It is important to note that the end of the pipe at 14 is directed downward, which presumably will help the gases to spread over the piston head like a blanket rather than rise up and mix with the cylinder charge. The design shown is an evolution of that which Sir Dugald Clerk experimentally employed in 1912. If desired, coil 13 may open into the air instead of into the exhaust manifold. There is no material difference between the two, provided always that the charge remains stratified and the bottom charge separate from the fuel-air mixture. There are indications that this is so, or rather that it is possible to work out a design where it will be so. Tests have shown, for example, that a spark plug located at the bottom of the compression space fails to ignite the charge at low throttle.

66 The construction suggested recently by Harry R. Ricardo in a paper before the Institution of Automobile Engineers,[1] for preventing preignition of kerosene, namely, the admission of pre-cooled exhaust gases to the cylinder to dilute the charge, belongs to the same class of expedients, though less fully elaborated. The writer has made several tests of the Ricardo scheme, but found that while it kills preignition, it also materially reduces the power output of the engine, as was to be expected, and therefore is not helpful. The difference between the Ricardo scheme and the one shown here lies in the fact that, first, Ricardo admits exhaust gases as an integral part of the mixture instead of as a stratum in the cylinder, and second, he admits the exhaust gases at full throttle, while in the scheme shown here they are admitted only at low throttle.

RATE OF COMBUSTION AND EXPANSION PRESSURE

67 But compression pressure determines not so much the actual rate of combustion of the charge, as initiation of the combustion. As pointed out before, the combustion of the first layer of mixture, in immediate proximity to the spark plugs, creates a rise of pressure and temperature in the rest of the cylinder far in excess of that existing at the end of the compression stroke, and it is this greater temperature which makes possible the rapid combustion of the charge in the cylinder.

68 It is usually assumed that in constant-volume-type engines the expansion pressure depends on the final compression pressure, and it is sometimes expressed as being so many times the compression pressure. This, however, is true only for the conventional type of engine running at sea level on gasoline. Actually it depends exclusively on the heat content of fuel contained in the charge, or, more correctly, on the rate of evolution of heat energy during the process of combustion. Conversely, the rate of combustion at each instant depends on the expansion pressure obtaining at the preceding instant during the process of combustion.

[1] *The Automobile Engineer*, Jan. 1919, p. 2.

69 The following calculation may make the meaning of the above statement clearer:

70 Assume two single-cylinder engines of the same construction, each with a swept volume of 75 cu. units; and that one engine operates in the open air at sea level, while the other operates in a compressed-air chamber in which there is a gage pressure of one atmosphere, or two atmospheres absolute. Both engines run at full load.

71 Let us now see how they will operate. Assume that the open-air engine works at 100 per cent volumetric efficiency, that the combustion period lasts 10 time units, and that the combustion proceeds uniformly (the latter assumption, while not absolutely correct, does not involve any very appreciable error). Furthermore, we may assume that the volume of gases developed from combustion is equal to seven times the volume of the original mixture, which is fairly close to actual conditions.

72 Then at the instant of ignition we have, say, 25 cu. units of compression space wherein are forced 100 cu. units of mixture. At the end of the first time unit 10 cu. units of mixture have been consumed, developing 70 cu. units of new gases, and raising the total volume of gases to 160 cu. units. At the same time the piston traveled down one-tenth of its stroke (not strictly true, of course), increasing the space to 32.5 cu. units. The compression ratio is therefore not 100/25, but 160/32.5. A similar calculation will show that at the end of the fifth time unit the volume of gases has increased to 400 cu. units, the space to 62.5 cu. units, and the compression ratio to 400/62.5, or to 6.4. This may be taken as the "expansion pressure index" for the open-air engine of given dimensions and operating on a given fuel. This is the conventional motor of today.

73 In the case of the engine placed in the compressed-air chamber, all conditions, including the final compression pressure, are assumed to be the same, the volumetric efficiency is the same, but the weight of the charge is double, in proportion to the greater density of the air. This also permits putting into the cylinder a double weight of fuel (the question of economy is not yet here considered).

74 Therefore, at the instant of ignition we have in the cylinder 200 cu. units of free charge, and as we assume the same compression pressure as in the open-air engine, we must have a double compression space, or 50 cu. units.

75 Carrying out the same calculation as before, we find that at the end of the first time unit of combustion there will be in the cylinder 320 cu. units of (free) charge occupying 57.5 units of space, and at the end of the fifth time unit 800 cu. units of (free) gases occupying 87.5 cu. units of space. Therefore, in the case of the compressed-air-chamber engine the compression ratio at the end of the fifth time unit is 800/87.5 = 9.1, as compared to 6.4 for the open-air engine. Actually, the difference in expansion pressures between the two types of engines will be greater than shown here because of the difference between the temperature of the gases, but not enough experimental data are available to determine the influence of this factor quantitatively.

RATE OF COMBUSTION AND EXPANSION PRESSURE

76 We thus see that it is possible to raise the expansion pressure very materially by forcing into the cylinder a greater charge than it will take when supplied with air at atmospheric pressure. What has been referred to as a "compressed-air-chamber" engine is generally known as a superinduced engine ("supercharged" is also used), and will be referred by this name in what follows.

77 Since the temperature of the mixture in the cylinder varies in proportion to the pressure to which it is submitted (apart from the direct action of the hot gases), it must be very much higher in a superinduced engine than in one of the conventional type (referred to above as the open-air engine). And while at the beginning of the power stroke approximately the same expansion pressures prevail in both cases if the final compression pressures are the same, the expansion pressure rises much faster and to greater final values in the case of the superinduced engine than in the conventional motor. This applies, of course, to the case when the same fuel is used on both engines.

78 We have established, however, that the rate of combustion of a gas depends on its temperature, and therefore it is obvious that by raising the pressure in the superinduced engine as compared with the conventional motor, we increase the rate of combustion of the fuel. Because of this, the rule as to rate of combustion given above may be also expressed by stating that the rate of combustion in a given space in a given interval of time during the process of combustion is a function of the number of heat units evolved therein

during the preceding similar interval of time, provided the volumes wherein the gases are contained have not changed materially during the two intervals. This definition, though somewhat more involved than the preceding one, is of advantage as it permits one to express the same relation in terms of molecules for the case of two engines having different terms of induction, but working with the same fuel.

BURNING LOW-GRADE FUEL

79 Those who have had experience with the burning of low-grade fuels (kerosene, distillate, paraffin gas oil, etc.) in engines of the constant-volume type know that the main difficulty lies not so much in igniting the charge as in properly burning it. With modern magnetos and battery coils there is no difficulty in producing a spark, and if necessary an arc, hot enough to ignite a properly compressed charge of practically any fuel that has been sufficiently atomized; but *running* an engine on the heavier fuels is a very different proposition, indeed.

80 This is due, as shown above, to the slow rate of combustion of the heavier fuels, especially at partial cylinder filling, which is confirmed by two facts: first, it is much easier to run a kerosene or distillate engine on full throttle than on low throttle or idling; and second, it is easier to run a kerosene engine at full throttle at moderate speeds than at high speeds (e.g., at 1200 to 1400 r.p.m. there is no trouble in running either on kerosene or distillate an engine with compressions ranging from 65 to 85 lb. and with proper carbureting, manifolding and temperature-control devices; but all attempts to run engines on these fuels at 3000 r.p.m. have failed so far as the writer is aware).

PRESENT STATUS OF ENGINE DEVELOPMENT

81 Summarizing the conclusions from what has been given above we are confronted with a situation in respect to engine development which may be expressed in the following few paragraphs:

a There is an impending shortage of the better grades of fuel, such as are now used in automotive equipment;

b There are still large quantities of lower grades of fuel available if only the engines can be made to handle them;

c The adaptation of the engines to the new types of fuel involves practically a basic redesigning of the essential features of the engine, such as manifolding, cooling, etc.

d Because of the lower basic rate of combustion of the lower grades of fuel, it is hardly possible that the conventional motor may be made to handle them with the same flexibility as it does gasoline;

e Also, because of the lower basic rate of combustion of the lower grades of fuel, the engines, if of the present conventional design, will needs be of lower speed and power output when operating on these lower grades of fuel than those at which they are now operated on gasoline;

f If lower grades of fuel are to be burned in automotive engines, the engines must undergo a much more thorough change in design than involved in a makeshift adaptation, namely, a change that will bring about a high rate of combustion of a charge consisting of a fuel having a low basic rate of combustion.

g The condition stated in the preceding paragraph can be accomplished by superinduction. So far no other means have been indicated to produce the same result.

h Since superinduction is also likely to produce a greater power output from an engine of the same volumetric displacement, and possibly an economy in fuel consumption, one may confidently look forward to greatly increased attention being given to it in the near future, both as a means of generally improving engines, and as a way of burning the heavier fuels.

SUPERINDUCTION

82 So little attention has so far been paid to superinduction that in the majority of books on internal-combustion engines this term is not even mentioned. Sir Dugald Clerk may be considered as the first who has carefully investigated the subject, but it was really the demand for aeroplane engines that would deliver good power at extremely high altitudes that brought it seriously to the fore.

83 Briefly, superinduction may be defined as a method of operating an engine of the constant-volume type so that the pres-

sure in the cylinder at the beginning of compression is higher than atmospheric pressure.

84 Various ways have been tried to produce superinduction. The simplest and best-known is that employed in two-stroke cycle engines with crankcase compression, where the pressure at the beginning of the compression stroke is usually from two to three pounds higher than atmospheric. The conditions of operation of two-stroke cycle engines, however, are such that the benefits of superinduction have been there largely balanced off by other factors.

85 Recently in this country Kessler has adapted the same scheme to four-stroke-cycle engines. He also uses crankcase compression and admits at the end of the induction stroke a small amount of air to the bottom of the cylinder, this raising the initial pressure to about 15 lb. absolute, instead of the 12.5 lb. that it would probably have been otherwise. The writer has not seen any logs of tests of the Kessler engine, but statements have been made in the technical press [1] to the effect that the method adopted has increased the power output of the engine enormously (it is stated that a six-cylinder 3 x 3 engine delivers as high as 100 hp.).

86 Sir Dugald Clerk himself has employed superinduction in a manner somewhat similar to that shown in Fig. 4, with the exception that, as far as the writer is aware, he did not provide any throttle in the exhaust-gas passage leading to the bottom of the cylinder.

87 A very different scheme has been suggested by E. Schimanek.[2] He arranges the engine in such a manner that it makes several suction strokes for one power stroke, the cycle being of six or more strokes. A receiver has to be used for storing the air compressed after the first suction stroke. Because of this, at the second suction stroke the volumetric efficiency can be good only when the compression space is comparatively small, which limits the application of the cycle to engines of very high compression, and makes it of more interest in connection with engines of constant-pressure than of constant-volume type.

88 As previously stated, superinduction took a new lease of life when war conditions created a demand for high-altitude flying. At high altitudes the air is so rarified that the weight of oxygen in the cylinder charge is reduced materially, which also reduces the weight of fuel that can be burned. The compression pressures are also materially reduced and there are good reasons to believe that

[1] Cf. *Automotive Industries*, June 12, 1919, p. 1293.

[2] *Zeitschrift des Vereines deutscher Ingenieure*, January 25, 1913.

the volumetric efficiency is also less than on the ground. All this cuts down the power output of the engine to a very appreciable extent.

89 It early became obvious that the engine power could be easily raised by increasing the density of the air delivered to the cylinder, which could be accomplished by ramming an extra charge of air. The question was only as to the type of the pump that could be used, since the pump had to be very light and still capable of handling a large volume of air. This problem has been solved by Professor Rateau in France and by Sanford A. Moss and E. H. Sherbondy in this country by providing a small fan, driven by an exhaust-gas turbine running at the terrific speed of some 30,000 to 40,000 r.p.m. It may be stated, in this connection, however, that the general results of the entire scheme have not been uniformly favorable. Superinduction did to the engine what was expected, and increased its power very materially, but the efficiency of the propeller (of the constant-pitch type) did not keep pace with the new conditions, and very nearly all that was gained at the power plant end was lost at the airscrew end. Until a variable-pitch propeller has been developed to the practical stage, superinduction for high-altitude flying will probably continue to be of doubtful advantage.

90 It is significant that in all these schemes, with the possible exception of the Schimanek arrangement, the supercharge consisted of pure air, although in the Rateau-Moss-Sherbondy method an increased amount of fuel was delivered by the carburetor.

91 The superinduction system shown in Fig. 5 belongs to another type, where the supercharge is made up of a fuel-air mixture. In this case two carburetors are provided, the main carburetor 18, and an auxiliary carburetor 13. The reason for using two carburetors is the desire to reduce the dimensions of the air compressor as far as possible.

92 The main carburetor is supplied with atmospheric air and the auxiliary carburetor with air from tank 14, to which air is delivered at a pressure of, say, 30 lb. absolute from a compressor (not shown). Valve 19 governs the admission of the additional mixture to the extent of limiting it to the induction stroke, the actual timing of admission being constant for a given engine, and governed by the position of port 12 with respect to the piston stroke.

93 The operation is as follows (cf. the timing diagrams which have been found best for an engine running at about 2000 r.p.m.,

Fig. 6). The main inlet valve *I* opens 8 deg. past top center (*T.C.*) and closes at 180 deg. Additional admission *S* begins 30 deg. ahead of bottom center (*B.C.*) and ends 30 deg. back of it.

94 The earliest tests with this system have shown what might have been expected, namely, that unless the pressure is the same at the inlet to the carburetor, in the carburetor float chamber and in the fuel tank, the operation of the carburetor becomes extremely irregular; in fact, at low throttle there is a tendency for the fuel to flow out of, instead of into, the mixing chamber, i.e., backward through the nozzle supply tubing. Because of this, air is supplied

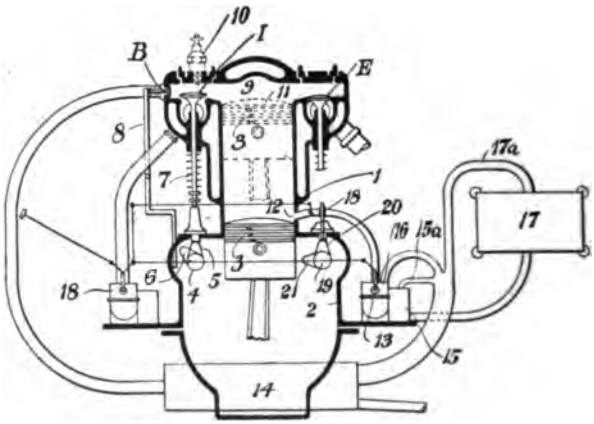


Fig. 5 Superinduction Engine with Two Carburetors and Balanced Operation of the High-Pressure Carburetor

to the float chamber (pipe 15^a) and to the fuel tank (pipe 17^a) at the same pressure as to the carburetor inlet.

95 It has been found to be easy to bring up the pressure in the cylinder at the beginning of the compression period (not compression stroke, which is about 30 deg. ahead of the actual compression period) to about 25 lb. absolute, with 30 lb. absolute in tank 14. The question is only whether the additional complication and power consumption of the air compressor will be sufficiently compensated by the gain in power and economy. As no power output calculations of superinduced engines are available in engineering literature, as far as the writer is aware, the following may be of interest:

POWER-OUTPUT CALCULATION OF SUPERINDUCED ENGINES

96 The formulæ used here are the same as those given in Heldt's Gasoline Automobile, Vol. I, edition of 1916, pp. 28 and 30.

$$W_e = \text{work of expansion}$$

$$= \frac{\pi \cdot b^2 \cdot a \cdot P_c}{14.4} \times \left(\frac{s}{r-1}\right) \times \left[1 - \left(\frac{1}{r}\right)^{0.32}\right]$$

where b is diameter of cylinder, a ratio of explosion pressure to compression pressure, P_c compression pressure, s stroke, and r compression ratio.

97 P. M. Heldt, in the book above referred to, gives a numerical calculation of the power output of a good automobile engine of conventional design. For the sake of comparison a numerical example has been here worked out for the same engine with super-

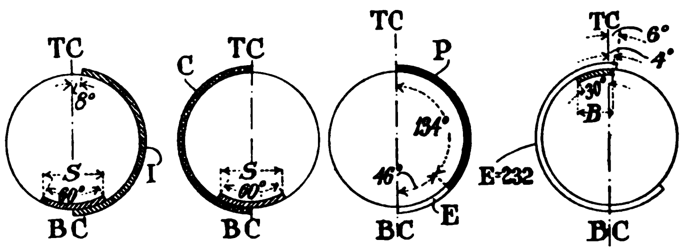


FIG. 6 VALVE-TIMING DIAGRAMS FOR THE SIX-STROKE SUPERINDUCTION CYCLE WITH PISTON HEAD COOLING

induction to 25 lb. absolute pressure. The engine under consideration is a four-cylinder, 3.5 x 5, running at 1800 r.p.m.

98 For a superinduction engine of the automobile type, P_c is assumed to be the same as without superinduction. The compression ratio r is of course much less, as the initial air pressure is twice as great, and is taken to be 2.5, as compared with 4.5 for good automobile engines. However, a is very much higher, as has been shown above, and is taken to be 5.5 as compared with 3.5 for conventional engines.

99 This gives for the work of expansion of a superinduced engine:

$$W_e = \frac{3.14 \times (3.5)^2 \times 5.5 \times 89}{14.4} \times \left(\frac{5}{2.5-1}\right) \times \left[1 - \left(\frac{1}{2.5}\right)^{0.32}\right]$$

$$= 1077 \text{ ft-lb.}$$

100 The work of compression is determined from the formula:

$$W_c = \frac{\pi \cdot b^2 \cdot P_i}{14.4} \times \left(\frac{rs}{r-1}\right) \times (r^{0.3} - 1)$$

where P_i is the initial pressure in the cylinder at the beginning of the compression stroke. In an ordinary engine P_i is estimated at 12.5 lb. It is taken here to be 25 lb. so that

$$W_c = \frac{3.14 \times (3.5)^2 \times 25}{14.4} \times \left(\frac{2.5 \times 5}{2.5 - 1}\right) \times (2.5^{0.3} - 1)$$
$$= 172.5 \text{ ft-lb.}$$

101 The useful output is equal to

$$W_u = W_e - W_c = 1077 - 172.5 = 904.5 \text{ ft-lb.}$$

or, in round figures, 900 ft-lb.

102 As the engine runs at 1800 r.p.m., there are 900 power strokes; also, the engine has four cylinders. The horsepower output is therefore

$$\text{Hp.} = \frac{4 \cdot 900 \cdot 900}{33000} = 98.9 \text{ hp.}$$

103 This does not, however, take into consideration the internal losses and the consumption of power to drive the air compressor, which may be roughly estimated at 15 to 20 per cent, the latter figure being here taken for the sake of safety. This gives 78.6 hp. from a four-cylinder, 3.5 x 5, engine running at 1800 r.p.m., as compared with 32.5 hp. for an engine of the conventional type. Somewhat lower figures have been obtained in actual tests, due mainly to the fact that the engine under test decidedly would not stand the higher power output.

POWER AND TEMPERATURE RELATIONS IN THE SUPERINDUCED ENGINE

104 The writer has been asked on several occasions to explain in physical terms the greater power output of the superinduced engine. It is due exclusively to the higher rate of combustion of the fuel and the resultant greater expansion pressure on the piston. After all, the formula *PLAN* holds good for all reciprocating engines no matter what may be the nature of the acting fluid. Now, in a superinduced engine P is very much higher than in the conventional motor, and it is therefore natural to expect also a higher power output.

105 The temperature relations in the superinduced engine are of peculiar interest.

106 First comes the question of losses to the water jacket. The usual assumption is that heat flow is a function of two variables for a metal wall of the same material and thickness: area of the surface transmitting the heat, and difference in temperature between the cold and warm side. Tests carried out by the writer appear to indicate, however, that the latter is true only to a limited extent, and holds good only when applied to the case of flow of heat during fairly long time periods. While sufficient work has not yet been done to establish a definite formula, it appears that when the difference of temperature between the hot and cold side is very great, and the time for the flow of heat short (measured in fractions of a second) the flow depends almost exclusively on the area of the cylinder wall. It would appear likely, therefore, that the loss of heat in superinduced engines will be only slightly greater than in engines of the same size of the conventional type, and hence considerably lower in proportion to the fuel burned. As a matter of fact, experimental work has so far confirmed this, though not enough has been done to establish a dependable rule.

107 But in another direction an important source of trouble has developed. In superinduced engines not only the rate of combustion is higher, but also the temperature at which the combustion proceeds, which is in accordance with what has been established elsewhere in this paper. Furthermore, the temperature of the exhaust gases is also higher than in conventional motors. All this imposes a severe thermal strain on the piston head, which, as a matter of fact, is none too well off in this respect even in conventional motors. This statement is limited to the case where the same fuel is used in superinduced engines as in conventional engines. The temperature of combustion depends on the rate of combustion of the fuel-air mixture.

108 There are two ways to help this situation. The first and most obvious is to run the engine at a lower speed, with better cooling of the cylinder walls, and a slightly earlier opening of the exhaust valve. In this way, there would be a smaller gain in power, but as the main aim of superinduction is the burning of lower-grade fuels, this would not matter so much.

109 Another way out would be to provide means for cooling the piston head. To do this by means of a flow of liquid, as has been done in Diesel-engine practice, does not seem promising with the

small, high-speed piston of motor-car and aeroplane engines. But air cooling of the piston head appears both easy and sufficient to the extent of keeping the temperature below the dangerous level.

110 In a superinduction engine we must have an air compressor of some kind, and to provide a blast of air for cooling the piston head involves only a small enlargement of the compressor. Fig. 5 shows how this has been carried out in the present case. Valve *B* is provided, operated from the same camshaft as the inlet valve, and connected with the compressed air tank 14. As shown in Fig. 6, this valve opens 30 deg. (it was found later that 20 deg. may be sufficient) ahead of the top center (*T.C.*) on the exhaust stroke, and closes 4 deg. after top center, or 2 deg. ahead of the closing of the exhaust valve, and 4 deg. ahead of the opening of the inlet valve. In this way a blast of air is acting on the piston head all the time that it is at or near top center on the compression stroke. In addition to direct cooling of the piston, the air blast through valve *B* has a scavenging action and in this way reduces the temperature of the incoming charge, which again helps to cool the piston. It may be remarked in this connection that the scavenging blast has been tried a good many times and has contributed to a better operation of the engine, but thus far not to an extent sufficient to compensate for the additional complications involved. In connection with superinduced engines, however, the complications are less material, as the air compressor is there anyway, and blast has to be used to cool the piston head.

111 It should be added that still another method may be employed, namely, the introduction of a spray of water directed on to the piston head at or near the end of the exhaust stroke. This has been tried by several experimenters in England and Germany in connection with constant-pressure-type engines, and has given satisfactory results under certain conditions.

112 From what has been said above, it appears that the cycle of the superinduction engine may be defined as follows (cf. diagrams in Fig. 6):

First stroke: main admission (*I*)
Second stroke: supplementary admission (*S*)
Third stroke: compression (*C*)
Fourth stroke: expansion (*P*)
Fifth stroke: exhaust (*E*)
Sixth stroke: cooling blast and scavenging (*B*)

all the six strokes to occur within two revolutions of the crankshaft.

LOW-GRADE FUELS IN SUPERINDUCTION ENGINES

113 We have seen above that one of the difficulties in burning low-grade fuels in conventional engines of the constant-volume type, namely, their low rate of combustion, is overcome by the greater expansion pressure in superinduction engines, which materially accelerates the rate of combustion by raising the temperature of the mixture. There is another difficulty, however, which has not yet been touched upon sufficiently.

114 It has been shown above that low-grade fuels have to be well atomized before they can be properly burned in the engine cylinder. But the lower the grade of the fuel, the greater its viscosity and the stronger its surface skin. For atomization of the fuel we have to depend upon the difference of pressure between that in the carburetor float and in the engine cylinder on the induction stroke, and between the carburetor air inlet and engine cylinder on the same stroke. Because of this, all else being equal, the degree of atomization depends primarily on the above differences of pressure. Tests have shown that while these differences of pressure, with atmospheric pressure at sea level at the inlet to the carburetor and in the float chamber, are sufficient in the case of kerosene and better grades of distillate, they are not high enough to permit us to handle even the best grades of fuel oil, such as paraffin gas oil, with any degree of satisfaction. The lower the grade of fuel the greater its viscosity and the stronger its surface skin.

115 In the case of the auxiliary carburetor 13, Fig. 5, the differences of pressure are about 15 lb. higher than in the conventional type of engine (and than in the carburetor 18), which makes it possible to handle the lighter grades of fuel oil without much trouble, thus permitting the carburetor 18 to run on gasoline, and carburetor 13 on a light grade of fuel oil.

116 This is particularly facilitated by the fact that, while complete stratification of the charge is hardly present, the top portion of the charge is practically pure gasoline mixture, which ignites first and creates sufficient pressure and temperature in the cylinder to burn the heavier oil in the lower part of the cylinder.

117 The commercial features of this construction and the question of costs have been deliberately left out of consideration, the use of the two fuels being mentioned here mainly with a view to giving an illustration of the vast possibilities of the application of the superinduction principle in constant-volume-type engines, such as used for automotive equipment.

118 The construction shown in Fig. 5 appears rather complicated. The reason for this is that poppet valves are shown in order to present the relation of the parts more clearly. Actually, however, a construction which requires five poppet valves per cylinder, while not impossible, is not good engineering. The situation is, however, entirely different with slide-valves, especially such as the Knight type. There all that is necessary is to cut a few holes in the valve cylinders and the walls of the engine cylinder, which does not involve any particular difficulties or sensibly affect the cost of production.

CONCLUSIONS

119 The shortage of the higher grades of fuel makes it necessary to review our engine design with a view to adapting it to the lower grades. The controlling element in this revision has to be the low basic rate of combustion of the new fuels, necessitating a design of engine where means are available to burn the heavier fuels at a faster rate than is possible in the conventional engine. Superinduction affords such a possibility, besides increasing the power output of an engine of given size. It offers, therefore, the most natural solution of the present difficulty, and, as a matter of fact, is already strongly under consideration by the engineering fraternity.

120 This is all the more so as it is already apparent that, if we want to secure an alleviation of the present fuel situation of a permanent nature, we must find ways of utilizing fuels that are more easily obtainable than kerosene, and that means refined natural oils, such as paraffin gas oil. This obviously implies a basic redesign of our engines along the lines of higher pressures and possibly temperatures.

121 One thing seems to be certain: While we have attained a very high degree of achievement in the design of the gasoline engine, the high-speed high-efficiency constant-volume type kerosene or fuel-oil engine that will take its place will have to be designed with a far clearer understanding of the processes of combustion than is evidenced by the engine of today. In other words, the good kerosene or fuel-oil engine will have to be first and last a better engine intrinsically than anything we have today.

No. 1726

RELIABILITY OF MATERIALS AND MECHANISM OF FRACTURES

By Charles de Freminville,[1] Paris, France
Honorary Member of the Society

It is a common experience that the behavior of materials in practice does not always agree with their behavior under test; designs worked out from data secured from test pieces have not always proved satisfactory, and actual machines have had to be modified from the experience of practice. The differences between test predictions and actual results are set forth in the following paper, and it is shown that the reasons for the discrepancy between test and actual conditions can be determined by an analysis of the mechanism of fractures of the test pieces and of actual specimens. Such an analysis reveals the fact that the mechanism of the fracture follows certain laws which can be formulated. The paper also deals with methods of designing members so that stresses in them do not conflict with the laws of mechanism of fracture. Pieces can be so designed and so assembled that the stresses in them largely avoid the conditions of fracture, and it is significant that when such design and assembling is carried out, the behavior of the actual machines is much more similar to the behavior of the test pieces themselves.

WHEN the author was invited to read a technical paper before The American Society of Mechanical Engineers the first subject which came to his mind was, The Reliability of Materials, chiefly because of his previous experience in the use of metals and with the testing of metals. If that word "reliable" impressed itself upon his mind, it was certainly because it is a word of which he has realized the full meaning in America, where the people are always looking for something reliable: reliable information, reliable products, reliable men; a country in which he found so many reliable friends, and which has proved herself so prominently and so thoroughly reliable in her immense effort to save the mutual ideal of civilization of France and America, which cannot conceive Science associated with an ideal other than Liberty.

2 Reliability is the quality we desire in the metals we use, but

[1] Mechanical and Industrial Engineer. Consulting Engineer, Schneider Works.

Presented at the Annual Meeting, New York, December 1919, of The American Society of Mechanical Engineers.

it is very difficult to ascertain it through a short test. The reliability of materials is like the reliability of friends. Whatever brilliant qualities they may exhibit, their reliability is known only in the long run, so that in the case of an apparent failure, the qualities of a metal, like those of a friend, should not be questioned too quickly. It is not enough to know the intrinsic qualities of the metal; a very careful study of the circumstances under which the failure occurred must also be made, and it is this fact which has made the study of fractures and of the mechanism of fractures one of great interest to the author.

3 During the early days of the automobile industry the problem of resistance of materials began to be considered in a new light. The first motor cars that were built were designed with factors of safety then generally accepted for the resistance of medium-carbon steel. But the engineer soon came into contact with the sportsman and was tempted to put into the gearing a stress much greater than the one for which it had been made. One of the problems to be solved, therefore, was to secure a material which could safely be used for gears.

4 At that time steel makers had produced, among certain other new products, ternary steels, which showed brilliant promises of new qualities if the results of the laboratory tests to which they had been submitted could be taken as a proof of their reliability.

5 Viewing the question of material for motor cars in a general manner, the constituent parts appeared to belong to two distinct classes. The first included such parts as gears, crankshafts, connecting rods, etc., which had to withstand severe normal work. Also only a small amount of wear was allowable and the class of accidents to which they were liable was limited but very serious, being chiefly those of seizing or an encounter with stray nuts or bolts.

6 The parts of the second class did not have much stress imposed upon them in ordinary service, but they did have to meet with what can be termed "normal accidents," and were therefore expected to bend to a certain extent without ceasing to fulfill their duty. To this class belonged front axles and, more generally, all the parts connected with the steering mechanism. For these, the "normal accident" can happen very frequently. It is almost impossible to say that the wheel of a car will not come rather abruptly into contact with a curbstone, or even with the wheel of another car. In such case something must yield or break. It is of course but natural to hope that something will yield and that this yielding has been foreseen; also that it will take place on a piece easily seen and

easily replaced, and that the material of which the piece is made will not cease to be reliable.

7 The impact test, to which attention was directed at that time in France by M. Considère and later by M. Frémont, seemed to be a very convenient method for ascertaining the characteristics of the parts of each class. The impact test, however, was devised primarily for soft steel and had to be adapted for very hard steel. But by making use of the impact test it was possible to select for the parts of the first class a grade of steel which only fractured after a very small amount of deformation, and this could be considered as a very excellent quality for the parts of this class.

8 The parts of the second class were expected to bend under the impact test to a very great extent, and it was easy to find wrought iron or mild steel possessing this characteristic, although the reliability of these materials for the use considered is a matter open to discussion. But here again the problem became complicated, because there were those who seemed reluctant to admit that some of the parts had to bend easily under certain circumstances and that a limited amount of toughness was not desirable, inasmuch as the steel maker pretended to have grades of steel uniting in a rare degree ductility with toughness.

9 The tensile tests of these steels showed a high ultimate strength and a great elongation. The impact test showed that a considerable amount of energy had been absorbed by the test piece, been possible great amount of distortion. It is evident that if it had which gave a to have a diagram of the impact test, showing at every moment what was the amount of stress corresponding to a given deflection, the matter would have been made very much clearer. But since there was no machine that would give such a diagram, a substitute was found in a static bending machine registering the amount of energy taken in the deformation process, for it was possible to ascertain that in all the cases which had to be investigated, the number of kilogram-meters absorbed was practically the same as in a quick bend or in a drop test. So far as the appearance of the fracture was concerned there was no perceptible difference in either case, which showed that the metal which seemed to possess ductility and toughness did not possess these characteristics to the extent believed.[1]

[1] The author has recently learned that the same idea of interpreting a drop test by means of a diagram given by a static bend test has been successfully used in this country for testing rails, and he believes that it can be used in many instances.

10 Figs. 1 and 2 are diagrams for comparing the results obtained by drop tests and quick-bend tests on notched test pieces. In these diagrams the ordinates of curves A are proportional to loads corresponding to deflections shown on the abscissa scale. In curves B (which are the integrals of curves A — ordinates $y_1y'_1$, $y_2y'_2$, etc.,

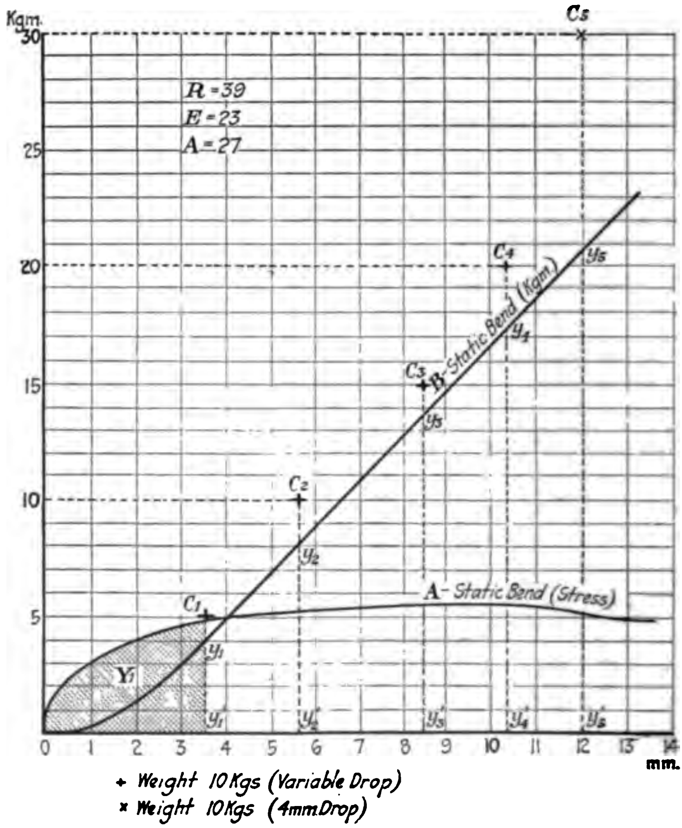


FIG. 1 DIAGRAM FOR COMPARING THE RESULTS OF STATIC BEND AND DROP BEND TESTS

being proportional to surface Y and so on) the amount of energy consumed in the bending test is given for each value of the deflection. The ordinates y'_1C_1, y'_2C_2, etc., are proportional to the amount of energy absorbed by the drop test for the same deflection, as measured in the bend test, and can be easily compared with $y_1y'_1$, $y_2y'_2$. The energy absorbed in the drop test appears to be only slightly greater than that absorbed in the quick-bend test, but the process has followed the same course in both cases, and the difference may be attributed to errors in the experimental work.

11 Fig. 1 is a diagram of a test of wrought iron or mild steel in which the amount of stress necessary to obtain a given deflection remains constant over a long range. This is the type of material which can be expected to give good results for parts of the second class.

12 Fig. 2 is a diagram of a test of a steel with a certain amount of ductility, in which the energy consumed for a given deflection

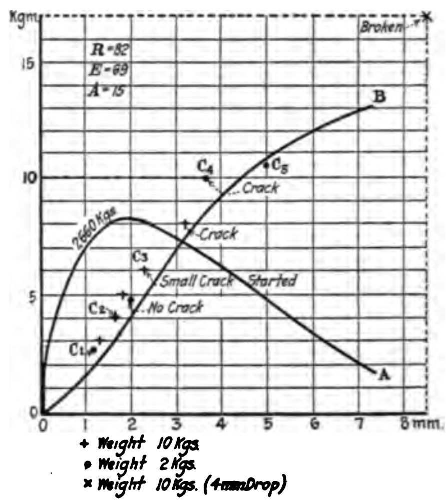


FIG. 2 DIAGRAM FOR COMPARING THE RESULTS OF STATIC BEND AND DROP BEND TESTS

appears to be greater than in the case of Fig. 1, and which was offered for parts of the second class. It should be noted that in Fig. 2 curve *A* is absolutely different from curve *A* of Fig. 1, as it shows that a maximum amount of the energy is absorbed after a small deflection has been obtained. This maximum is the point where the piece begins to show a crack, which of course did not exist in the test piece whose diagram is shown in Fig. 1.

13 Parts of the second class made of this metal proved a decided failure. Even without experiencing the "normal accidents" which they were expected to withstand, they showed a great tendency to crack without the slightest deformation, which method of fracture

FIG. 3 RESULT OF SHARP SHOCK ON A GLASS VESSEL

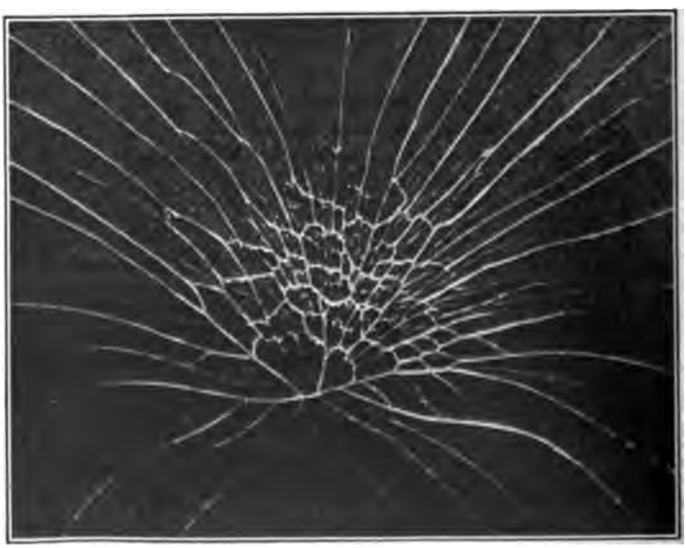


FIG. 4 DEVELOPMENT OF THE FRACTURE RESULTING FROM A SHOCK ON A THIN SHEET OF GLASS

has been called *fissillité* by the late M. Brustlein, of the Jacob Holtzer firm, Unieux, France.

14 By looking closely into the fracture of the broken test piece, which was supposed to possess ductility and toughness, it could be seen that most of the distortion, or flowing of the metal, had taken place not before the fracture was started, but during the process of extension of the fracture, and this gave to the amount of energy recorded an altogether different meaning. In this case the failure of the material could be traced to a bad interpretation of the test, but other failures were encountered which could not be traced to the quality of the metal used. Typical among these was the sudden snapping of levers, under conditions which could be specified but

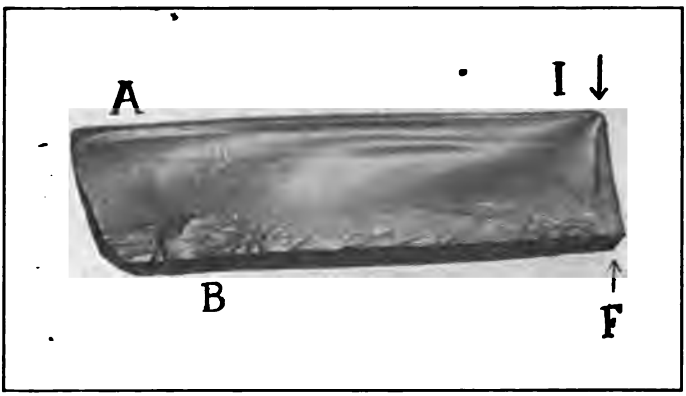


Fig. 5 Fracture Surface in a Slab of Glass

which were difficult to explain. The impact test, like others, could not give a clue as to reliability in certain circumstances.

THE STUDY OF FRACTURES

15 The uncertainties in the behavior of material, the propagation of a crack without any distortion of the piece, the sudden snapping of a rod without any perceptible alteration in the neighborhood, even the changes which were produced in brittleness under heat treatment, appeared to be difficult to account for, and it seemed to the author to be worth while to add to the theoretical views of the question by making, as far as possible, a thorough study of the fracture itself, which would tell, in some cases, the story of the failure.

FIG. 6 SMALL SPLINTERING FOCUS GLASS

FIG. 7 SPLINTERING FOCUS GLASS

16 It is frequently believed that fracture is a form of failure of such peculiar character as to escape all rules. However, the attention of the author was first attracted by the very regular fractures of sandstone, which is used extensively in France for pavements, and which so resemble one another that they seem to be real diagrams of what has happened during the rupture. Moreover, a certain number of peculiarities of the fractures were found to be common

FIG. 8 FRACTURE IN JUDEA BITUMEN

to both sandstone and steel. It was also noticed that these same features were found in glass fractures, in which they could be observed with great accuracy; also that they were still easier to observe in Judea bitumen, where the breakage was produced very easily. A further study has shown that the fractures developed in an identical manner in these various materials.

17 It is not possible in the limited time at the disposal of the author to go thoroughly into the subject of the study of fractures,

and as a result attention can only be called to some of the main features. Fig. 3 shows a fracture developed in a glass vessel by a sharp shock. In this should be noted the symmetry of the surfaces

FIG. 9 FRACTURE IN GLASS

originated and the peculiar disposition of the starting points of these surfaces from both ends of a small line to which the surfaces are tangent. Fig. 4 shows the development of the fracture resulting from a similar shock in a thin sheet of glass. Here great symmetry is seen, the network also originates from a small line, and some of the surfaces extend entirely through the glass, some only partially.

18 Fig. 5 is taken from a slab of glass broken in the same

FIG. 10 FRACTURE IN JUDEA BITUMEN

manner. It shows one of the surfaces dividing the slab into pieces, which was also partially cut by other surfaces. In this particular case the blow was given at *I* on the top of the slab, and it will be seen

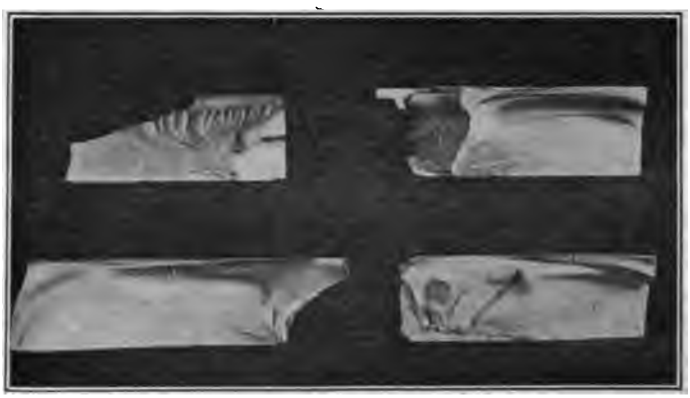


FIG. 12 FRACTURES IN TOOL STEEL

that the small line, noted in the previous figures as the origin of all the cracks, is situated at *F* at the opposite side of the surface. It presents a very remarkable appearance and the general disposition of the surface induces us to believe that the fracture originated in that region.

19 Figs. 6 and 7 show in great detail other specimens of that very peculiar surface, which illustrate very clearly that there is a starting point of the fracture. These examples enable us to see that one of the main surfaces cutting the test sample has not expanded at one stretch, but is formed by surfaces overlapping each other like the blades of a fan. This is made still more evident by the fracture

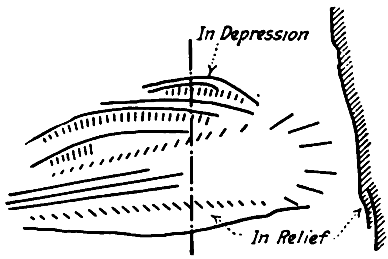


FIG. 11 SCHEME OF FRACTURE IN JUDEA BITUMEN

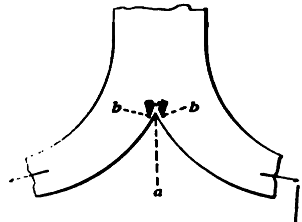


FIG. 14 SCHEME OF TEARING, WITH CRACKS THAT ACCOMPANY IT

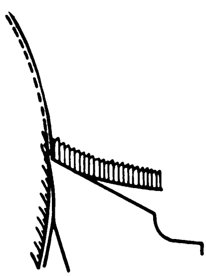


FIG. 16 SCHEME OF STEEL CHIP

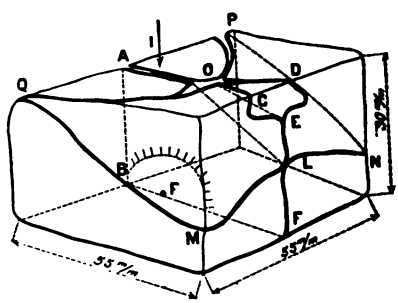


FIG. 18 OUTLINE OF THE FRACTURE OF A BITUMEN BLOCK

shown in Figs. 8 and 9. These surfaces overlap each other at distances often smaller than a hundredth of an inch, and by making use of the method called *recoupement* (recutting) by the geologists, it is possible to follow the successive order in which the various elements of the fractures have developed.

20 Fig. 11 shows in greater detail one of these surfaces, and enables us to point out some of its special features. Part of the surface is bright and part dull. A study of the bright part can be made from Fig. 10, which shows a standard fracture in Judea

bitumen. It is very easy to follow the surfaces overlapping each other, as is shown by the scheme in Fig. 11. The general arrangement of the surfaces bears a very close resemblance to the surface of water as it runs in a culvert, passes over a dam, or expands in a shallow pond.

21 The same arrangements of fractures, overlapping surfaces, foci, etc., which are easily observed in glass and Judea bitumen are also met with in tool steel, as shown in Fig. 12.

22 We find that the dull part gives the impression that it has been rent or torn away, while the bright surface seems to have been carefully cut. Fig. 13 shows how the dull surface is originated by

Fig. 13 Rents Formed under the Main Surface in Glass

small rents coming from under the surface. If now we come back to the focus of Figs. 6, 7 and 8, and look for the point where it has been originated we find that it also lies under the surface.

23 Fig. 14 shows what takes place in the tearing of a tough substance such as gelatine. The cracking *bb* travels ahead of the parting at *a* and prepares the way for its advance.

24 Fig. 15 is another and striking example of the tearing of a tough material. The parting at or near the edge of the tool has been made possible by a "cracking" similar to that shown in Fig. 14. The sketch in Fig. 16 shows the scheme of the formation of the steel chip.

Fig. 15 Steel Chip

Fig. 17 Fracture of a Bitumen Block

25 But how can a bright surface be produced? How can the cutting process be carried on without the sub-surface cracking? How can surfaces be cut which run parallel within such small distance as the ones we have noted? The only explanation which seems to be satisfactory is that the cutting power is a vibratory motion of the surfaces of the fracture, so long as they do not join each other.

26 Fig. 17 shows a very typical arrangement of the fracture surfaces in a bitumen block and Fig. 18 gives an outline of the different parts. The same arrangement is constantly found in steel

Fig. 19 Piece of Steel Tire Broken in a Drop Test

fractures, Fig. 19 showing a similar fracture in a piece of a steel tire broken under a drop test.

STRESS AND STRAIN

27 What are the conclusions to be drawn from this brief study of the mechanism of fractures? How can it pretend to be a contribution to the science of resistance of materials? In a general way, by showing the succession of the phenomena which have taken place during the spreading of the fracture, it affords an experimental proof that the straining effects are more detrimental to the material

than mere stress. This is in accordance with what has been shown by a purely mathematical study of the question.

28 In case of an impact fracture, be it in a piece of glass or in a piece of steel, we find the starting point of the fracture under the surface, although the maximum stress must certainly be located on the surface itself.

29 In the case of a piece subjected to tension only if a comparison is made with the fractures already considered it will show that fracture has originated along the axis of the test piece (see Fig. 20). The surface of the test piece itself sustains a very important stress, but without being subjected to the strains resulting from

FIG. 20 TEST PIECE RUPTURED UNDER TENSION; FRACTURE ORIGINATED ALONG THE AXIS

the contact of layers undergoing uneven stresses, flowing, so to speak, to give the reduction of area.

30 The predominance of the internal or straining action is no less striking in the case of a rent where the straining effect travels in advance of the stress which could be originated at the separating point. (See Fig. 14.) This condition should induce the metallurgist to pay more attention to the causes which produce straining, in other words, to the tensions which may exist in the pieces after being cast or forged. This is illustrated by the changes in brittleness which occur in glass undergoing heat treatment. The heat treatment does not bring about any change in its molecular arrangement, but the glass coming out of the furnace is very brittle; after being properly annealed it is far less brittle and after being "chilled," or submitted to a proper heat treatment, is so much less so that it is in the so-called unbreakable state; that is to say, it can sustain shocks and bending to a much greater extent than ordinary glass.

31 The first condition is one of very uneven tension; the second, one without tension; and the third, one which has a beneficial tension. This unevenness of tension also exists in steel, irrespective of the chemical or other changes in structure. An example of the effect of these internal tensions is given by Fig. 21, which shows a fracture in a rail. It is evident that the fracture was started in the neighborhood of the axis of the head, a fact which could not be accounted for if the piece had been under normal conditions; therefore the causes of a fracture of that kind must be traced to a tension most

Fig. 21 Tension Fracture in a Steel Rail

likely produced in the rolling mill. This case illustrates the mode of cutting which produces bright surfaces in glass, and which can only be explained by the presence of a vibratory motion of the surfaces when they are apart from one another.

32 In conclusion the author trusts he has created the impression that a methodical study of fractures is of the greatest importance, especially as it leads to a knowledge of the causes of failures which cannot always be attributed to the intrinsic qualities of the metal itself.

No. 1727

MOTOR-TRANSPORT VEHICLES FOR THE UNITED STATES ARMY

By John Younger, Pittsburgh, Pa.
Member of the Society

While the title of this paper might indicate that the mechanical features of the various types of motor-transport vehicles would be discussed, such is not the case, for to do so, as the writer points out, would merely be to recapitulate catalog specifications. The paper accordingly contains a discussion of the types of vehicle best adapted to various military needs. Motor-transport vehicles have been classified by the Army under ten broad heads, ranging from the motorcycle to the 5-ton truck, and such a classification has been found to be a most satisfactory one. The ultimate ideal is of course to standardize a single type of vehicle for each class, and with that point in view the writer discusses the various vehicles used. Numerous photographs are presented as illustrative of the American design and construction, and tables giving data on the various types used during the war are also included.

IN considering the subject of motor-transport vehicles used by the U. S. Army the writer believes it is unnecessary to go into any detail regarding the history of motor-vehicle transportation in military use. It is of interest in passing, however, to note that steam vehicles were successfully used years ago, reaching their highest point of development in the British-South African war. Motor road transportation does not come into great prominence again until the Great War of 1914, when its use exceeded the estimates of even the most radical. During the war, motor vehicles were used by the tens of thousands. They paralleled the railroads, they crossed the railroads, they radiated in all directions; they were even employed in munitions supply work in England; motor trucks loaded with shells in the Midlands were driven straight to the channel ferry and run right up to the front line in France, without any changing of load, the saving in time being of course enormous. It is a matter of record that the fight for Verdun was fought and won largely through aid furnished by motor-vehicle equipment. The railroad was at the mercy of the enemy, and a military highway accommodating four streams of trucks was built, thus saving the city.

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

2 These are matters of history pointed out principally to illustrate a new mechanical feature which has arisen in warfare. The successful army, as of old, is in large measure the one which possesses the characteristic of great mobility. The internal-combustion-engine motor vehicle has in the past decade accomplished a revolution in military mobility and it is proper, therefore, that attention should be drawn to the technical features underlying this great change.

3 It may be of interest to note, before discussing the various types of motor vehicles which were used during the great conflict, that in England every mechanical vehicle that was in existence was requisitioned for some form or other of military service, and that during the first few months of the fall of 1914, England commandeered every vehicle of any kind, no matter how old, to rush troops and supplies to the front. In this country we were fortunate in not having our ordinary commercial transportation dislocated to any large extent. To the best of the writer's knowledge there were no second-hand vehicles commandeered for Government use, but a large number of vehicles produced in this country were at some time or other requisitioned for service with our forces.

4 To go into detail on their mechanical points would be merely to recapitulate their catalog specifications. It is felt advisable, therefore, to try to form some measure of classification so that the whole broad array of motor equipment, ranging from the small motorcycle to the huge 5-ton cargo-carrying truck, can be studied intelligently and with a view to seeing in what direction we should plan in the future to take care of our military needs.

CLASSIFICATION OF VEHICLES

5 Such a classification would be as follows below, and as the war progressed it was found more and more that this classification was a scientific one into which the various types of vehicles tended to concentrate, so that in the ultimate ideal there would be one standard-type vehicle in each classification. This particular point is one, however, which will be studied later:

1 Motorcycles
2 Light Passenger Cars, suitable for carrying five passengers
3 Heavy Passenger Cars (enclosed or limousine type) for carrying staff personnel
4 Ambulances
5 ¾-Ton Trucks, for carrying cargo or personnel

6 $1\frac{1}{2}$-Ton Trucks (either pneumatic or solid tires) for carrying cargo or personnel
7 3–5-Ton Trucks on solid tires
8 Trucks of the 4-wheel-drive type
9 Cargo-carrying track-laying type of vehicle
10 Trailers
$\frac{3}{4}$-ton
$1\frac{1}{2}$-ton
3–5-ton
10-ton

SPECIAL ARMY REQUIREMENTS

6 Before proceeding with a discussion in detail of each of the types in the foregoing classification, it would be well to consider whether the Army has or has not special requirements which render necessary the special design in their equipment. The motor vehicle of whatever type, with the exception of the track-laying type and four-wheel-drive type, is designed primarily to operate over roads. The inability of the motor vehicle to work economically on poor roads has called forth a general demand for improved highways which has resulted in the spending of millions of dollars on state highways this year alone. These state highways are chosen in location just as a railroad is — because of the economic purpose the road will serve, either in our commercial life or in our recreation. The modern passenger car is rarely driven over poor roads. One has only to contrast the congested highway on a Sunday with the absolute tranquility of a nearby dirt road to demonstrate the truth of this statement. It is a fact that designers and manufacturers have taken note of this and have intended their vehicles to operate on average good roads. The very small minority that operate pleasure cars on poor roads have to pay the penalty, and even more so with motor trucks. A great volume of business lies in the cities where exceptionally good roads are met, whereas the number of trucks operating outside of the cities is very small. The amount of power necessary to drive motor vehicles on good roads is only a fraction of that required to drive them over heavy roads.

7 The problem of the Army, however, is different, for rarely, if ever, does it operate in a big city, its operations being confined to the open country. It has to take roads as it finds them and sometimes it has even to make its own roads. As pointed out above, it must be mobile, and mobility in an army means ability to go any-

where. It is a matter of interest to examine the roads on the borderlands between this country and Mexico and to observe that they actually barely exist. The needs of the Army must therefore be considered from the military standpoint and neither political nor commercial considerations should be allowed to block the main issue, namely, that the Army must have a proper motor transportation suitable to insure its mobility under any possible circumstances in which it will be called to fight. Stress is laid on this point as there have been arguments pro and con as to the advisability of the army using commercial vehicles. The war in France brought little light on this subject because for years the battle line was practically stationary and the country behind the lines had innumerable highways of high-grade construction of an antiquity reaching back far beyond the days of Napoleon — actually to the days of Julius Caesar. It is true that in other parts of the world, in Africa, Russia and in Southern Europe, battles were fought in countries with few roads, but there it was found that the average commercial type of vehicle had definite limitations. The motor vehicle, therefore, for military use should be considered from the military aspect and we should not necessarily try to foist on the Army vehicles which are of proven value only in the commercial field.

DISCUSSION OF TYPES — MOTORCYCLES

8. The motorcycle proved of tremendous use in warfare. Where despatches had to be carried over exceedingly rough country, where high speed was essential, where transportation under fire was necessary, the motorcycle proved its value. Two of the commercial makes of motorcycles manufactured in this country proved that in general they were satisfactory from a military standpoint. Military use, it is true, developed certain weaknesses which were ultimately remedied, but the experience of the war indicated that the present day motorcycle of standard construction suitable for [illegible] use would survive as a standard of equipment, and that a light weight machine — not over 250 lb. in weight — should be developed for "solo" work.

LIGHT PASSENGER CARS

9. Two American machines were used in great quantities, the Ford and the Dodge. Both gave good satisfaction but it is felt that [illegible] could be obtained by concentration

on one passenger machine of a somewhat more rugged design. There are, however, commercial aspects in this case as machines such as the above have a tremendous market off the beaten highway and their economic and satisfactory use by farmers and the like over poor roads, or over no roads at all, furnishes sufficient test to warrant the army considering very seriously in this case the commercial vehicle. On the whole, it was found that the car of the Dodge type gave the most satisfactory results. This type of vehicle was used for officers in their work involving the covering of territory, in the transportation of light supplies, as an emergency repair wagon and in all kinds of ways which will suggest themselves to any student of military affairs. It is the writer's personal opinion that there is no necessity for two types of cars, one type alone will be ample. Only in cases of sudden emergency, where a tremendous increase in the army would be required, would it be advisable to increase the number of types.

HEAVY PASSENGER CARS

10 For staff use it is essential that a more powerful car, capable of maintaining a high speed over long distances, should be provided. This car should be preferably of the sedan or limousine type to insure comfort to the occupant. The staff officer in directing troop movements is often required to make long journeys at night in such a car, and all facilities should be provided to insure rest. This car should be provided with facilities for carrying maps, reports, etc., and with necessary lights so that such could be studied during the night. This car will only have a limited use and will be used almost exclusively along the highways, so that a commercial type of car would be quite suitable. During the war the Cadillac was used exclusively for this purpose by the American Army and its specifications would stand today as being typical of what would be required for future use. (See Table 1.)

AMBULANCES

11 The ambulance is a peculiar type of vehicle in the sense that while its load consists of only four or five passengers, the shape of the load (the patients, of course, being recumbent) necessitates almost a box-van construction for the body. Furthermore the fact that the patients have to be carried with great care necessitates special attention being paid to the springing and the comfort generally of these vehicles, particularly when it is realized that they

must operate as closely as possible to the firing line and in all probability some distance away from the good roads, and subject to shell fire. The writer feels that for this purpose the chassis used for passenger carrying is one which will be found most desirable for this use, with a possible change in the direction of lengthening the wheelbase to insure better adaptation to the body.

12 During the war there were two types of ambulances used, one a simple 4-patient ambulance mounted on a long-wheelbase Ford chassis; the other, a more elaborate one mounted on the $\frac{3}{4}$-ton G.M.C. truck. Both ambulances had their distinct use, the Ford type being used close to the firing line, while the G.M.C. was used more behind the lines from the rest stations to the base hospitals. In view, however, of the limitation in number of our standing army, it is believed in the interest of concentration of equipment that it would be advisable to have only the one chassis used for this purpose. It is felt unnecessary to go into the details of this chassis.

PASSENGER-CARRYING HEAVY VEHICLE

13 The passenger-carrying heavy vehicle calls for special mention as it differs from the cargo-carrying type. In connection with artillery work it is necessary to carry a band of men for special work. Reconnaissance also calls for a crew of men to go out with range finders, various binoculars, instruments and tools for marking so that a proper site can be selected for the placement of field guns. The vehicle for this purpose must be provided not only with seats carrying the necessary personnel, but also with proper compartments for carrying the various tools required. An example of such a car is shown in Fig. 1. The vehicle for this purpose should be capable of traveling over rough ground and should be, as are the others, of sturdy construction. Another vehicle is the machine-gun truck in which a crew of men will carry one or more machine guns with the necessary equipment and supplies. The same type of chassis is also desirable and the body is very similar to that of the reconnaissance type.

14 A third type of passenger-carrying vehicle is what is usually known as a staff-observation car. This is a large car — sometimes of the omnibus type in which as many as nine or more passengers can comfortably be carried. This is used on occasions when the staff, particularly in the artillery, wishes to make an observation run or for the purpose of visiting different posts. The $1\frac{1}{2}$-ton cargo-carrying chassis with pneumatic tires is quite suitable for this work.

LIGHT PNEUMATIC-TIRE TRUCKS

15 The $\frac{3}{4}$- and $1\frac{1}{2}$-ton pneumatic-tire trucks can be grouped together as their work is very similar. A high state of development was found in these capacities of commercial trucks from the Army standpoint, this being most certainly a result of the fact that both these capacities are used in large numbers in country districts where bridges and other limitations do not interfere with their use. These vehicles are used for carrying moderate loads, the smaller size being used for emergency repair work, carrying emergency stores, and for final distribution of supplies from the main dump or from one or other of the advanced stations. They are also used in large num-

Fig. 1 Artillery Reconnaissance Body on $\frac{3}{4}$-Ton Chassis

bers in connection with the Air Service, furnishing supplies of all kinds not only for the airplanes but for the personnel. Airplanes, of course, had their stations off the road in the fields, and naturally the lighter trucks were found necessary.

16 Special attention is called to the use of pneumatic tires. It was found in trucks up to $1\frac{1}{2}$ tons capacity that pneumatic tires allowed running on very poor roads and over comparatively soft surfaces, such as a grass field, under conditions that would have prohibited the use of solid tires. It was also found that high speeds could be maintained for long periods with practically no damage to the vehicle. For example, a $1\frac{1}{2}$-ton capacity cargo truck could be driven at a speed of 30 miles per hour over give-and-take roads for long periods. This, of course, has a tremendous value in mobility

and also in emergency cases where supplies can be rushed forward. Under certain conditions, such as operating over sandy roads (as are found in many parts of the country), these moderate-capacity trucks with large pneumatic tires would be practically the only trucks that would be of any value.

17 Much discussion has raged over the use of pneumatic tires, whether they should be single or dual on the rear. European practice has been to use dual tires on the rear, the front and rear size being the same, thus eliminating the necessity of carrying two spare tires and tending toward interchangeability. American practice has been to use one size of tire on the front, such as, for example, 38 in. × 7 in. and a larger 40 in. × 8 in. single tire on the rear. Both types have worked out successfully during the war and it is felt that a little longer period of development is necessary before a decision could be arrived at as to which is likely to be the surviving type, if not both.

THE 1½-TON SOLID-TIRE TRUCK

18 The 1½-ton cargo-carrying truck with solid tires can be looked upon in the Army as the maid of all work. Cargo of all kinds, men, supplies, food, and munitions are carried on this steady-going vehicle. This is the vehicle which in motorized equipment has taken the place of the standard mule escort in the old Quartermaster Department. Able to run at 16 miles per hour, of sufficiently light weight that it can go over any average bridge, thoroughly worked out over a period of years in commercial life, the Army found but little change necessary in adapting it for their purposes. In the selection of one type for standardization it was found very difficult to distinguish which among a number was the best, and it is no slur on any of the types not approved when, for various reasons not always connected with their technical worth, they were rejected.

THE 3–5-TON CARGO-CARRYING TRUCK

19 The 3–5-ton truck was not used during the Mexican War and it remained for the Great War to establish its status as one of the standard vehicles of the Army. It was around this capacity of truck that tremendous discussion raged during the early stages of the war. The Army, in considering the various heavy-capacity trucks on the market, came to the conclusion that they were unsuitable for military use. The 5-ton truck of commerce is used almost solely around the cities. Bridge and load limitations have

been, of course, partially preventive of its use in outlying districts, but the big obstacle has been the fact that the industry, requiring a truck of this capacity, is almost invariably located in the larger cities and requires this purely for city use. The commercial vehicle of this capacity has therefore been developed, consciously or unconsciously, along good-road conditions. It was found that, in operation over poor roads, it had insufficient ability to proceed at a regular speed through heavy mud holes or heavy winter conditions. This point will of course be argued by those responsible for the average heavy-capacity truck, and while it is true that such

Fig. 2 Standard "B" Truck Fully Equipped for Cargo Purposes

trucks could operate under such conditions, the penalty paid was nevertheless tremendous.

20 For this reason, the Government saw fit, in 1917, to design and develop a truck of its own, this being known variously as the "Liberty" truck or "B" Type truck, the latter being its official title. Fig. 2 shows this truck fully equipped for carrying cargo. It was developed by the coöperative efforts of the engineers of the industry and the Government and was put into production, approximately 18,000 being manufactured to meet the demands. In this vehicle the special points that were required by the military use were built into the truck and were not added as an afterthought. In addition, a comparison of Tables 3, 4, and 5 and Figs. 12 to 14 at the end of the paper, will show that the ability of this truck (piston displacement per 1000 lb. moved 1 ft., or the torque at the rear wheels' circumference) is approximately 50 per cent greater than that

of similar capacity in commercial trucks. This is more noticeable on the low or first gear drive. In driving around on the various trucks, watching their performance under heavy conditions, one could not but be impressed with the ease with which this truck operated over extremely heavy going.

21 This capacity of truck was developed as a larger brother to the $1\frac{1}{2}$-ton truck. Its military advantages were not at first seen but later they became obvious. Its overall length was less than 10 per cent greater than that of the $1\frac{1}{2}$-ton truck and similarly with its width. Carrying two or three times as much cargo as the smaller truck, it offered virtually the same target on the road and in convoy work was almost 66 per cent less in length. From the standpoint of what might be called the direct labor problem of the Army, the number of men required to operate the heavy truck was exactly the same as that for the lighter truck; thus, for given amount of cargo it released more men for the firing line. The work of repairing, the number of spare parts necessary, maintenance and cost of running were also almost identical with those of the lighter truck, and it is not surprising, therefore, to find that although this class of truck was introduced in the Army at a later date, it made tremendous strides, outnumbering any other capacity of truck. Incidentally, the French and British armies found the same thing, the average truck capacity developed during the war as being suitable being somewhere about four tons. Another advantage that showed itself, due to capacity and ability to put on large platforms, was that this vehicle offered means of using trucks for all kinds of work where mobility was essential. This will be treated of later under the heading of "Special Types."

22 It is felt that this "B" type truck will be the standard heavy cargo-carrying truck for the army. It will undergo normal developments with time, and in peace conditions, prove itself of tremendous value in military operations. During war times, if there be any kind of highway system, it will also demonstrate its value. It is believed that with the proper organization and with the proper help from the Engineer Corps in improving the existing highways and bridges, it will be able to operate almost anywhere that the army operates.

THE 4-WHEEL-DRIVE-TYPE TRUCK

23 The 4-wheel-drive truck shown in Fig. 3 is that type of vehicle in which the driving power is conveyed to all the four wheels

and in which the full weight of the truck is available for traction purposes. Sometimes this type of truck steers on all four wheels, and sometimes only on the two front ones. Almost invariably, however, there is some type of differential locking or partial locking device which insures power being delivered to each wheel, whether or not that wheel is on a slippery or hard surface.

24 This type of truck is also subject to great argument; either its users are very strongly for it or they are very much against it

Fig. 3 Ordnance Design of 4-Wheel-Drive "Tractor-Truck"

Note the kind of ground this truck was expected to traverse

— there seems to be very little middle ground in the discussion. On analysis, the reason for this shows itself. In military work and in emergencies it is essential that certain supplies be carried from one point to another, no matter what the terrain may be like. Sometimes the conditions are so bad that men must carry the supplies themselves or by mules; sometimes the conditions, while severe and absolutely prohibitive of the use of the ordinary vehicle, will yet allow of the use of the 4-wheel-drive truck. With driving chains on all four wheels and with an extremely low gear reduction, as will be seen from Table 2, every ounce of power is available for traction and the 4-wheel-drive truck accomplishes its mission. Feats like these of regular daily occurrence in the Army have made military people, particularly in the Ordnance Department and Artillery Corps whose work is mostly "off-road," staunch adherents of the 4-wheel-drive truck.

25 It must be realized, however, that during this work the strain on the truck is enormous. Only the heaviest jobs are given to this type of vehicle and under these severe conditions, it is not difficult to understand that the maintenance problem is tremendous. It is from this last viewpoint, in comparison with the 2-wheel-drive vehicles, that the 4-wheel-drive type has suffered, and suffered, it must be said, unfairly. Over good or reasonably good roads there is no necessity for a 4-wheel-drive type of truck, but over almost impassable roads the 4-wheel-drive truck will justify its existence. In the service of the Marine Corps, for example, where weight limitations are necessary and where the ability to go anywhere is essential, the $1\frac{1}{2}$-2-ton-capacity 4-wheel-drive truck is standard.

26 During the war a demand for a heavier-cargo 4-wheel-drive truck than that developed in this country arose, and as a result the Ordnance Department designed and produced a few trucks whose characteristics are tabulated in Table 2 under the heading of "Ordnance Tractor Truck." A comparison of this truck with the other two trucks and a further comparison with the 2-wheel-drive trucks used will give an idea of the tremendous ability of this type of vehicle.

27 It should also be noted that the 4-wheel-drive type of truck proved its value as a tractor for hauling guns over roads where the track-laying type of vehicle was almost prohibited. In hauling trailers and in the general haulage work required, this 4-wheel-drive type of vehicle became a necessary part of the equipment.

TRACK-LAYING CARGO-CARRYING TRUCKS

28 First of all, this type was developed for agricultural-tractor purposes, the idea then being seized upon and used by the British in the design of their tank. It was natural that the caterpillar or track-laying type of vehicle should eventually be discussed in its relation to carrying cargo over "off-road" conditions. As practically all production capacity was used in the manufacture of tanks and what spare capacity there was was used for agricultural tractors it was almost impossible to develop the cargo-carrying track-

laying type of vehicle during the war, but all indications pointed to the necessity for considering this design very seriously for the future in order to still further attain the ideal of perfect mobility. Small progress was made however, but still enough to show that the attention of military designers should be directed largely to this type. For winter conditions, for example, this may be found necessary; and for shell-torn fields, otherwise impossible to traverse, a track-laying vehicle may give the solution of the problem of transportation. One drawback to its use at present is the fact that over hard roads the wear and tear on the vehicle is greatly exaggerated

FIG. 4 EXPERIMENTAL TYPE OF TRACK-LAYING AMBULANCE USING FORD CHASSIS

and the maintenance problem is very severe. There is no question, however, but that these mechanical problems will be solved in time. An interesting application of the track-laying principle is shown in Fig. 4.

TRAILERS

29 Trailers, strictly speaking, are not automotive vehicles, but their use is so linked with the others that it is necessary to consider them all together. Broadly their capacities fall under the same headings as do the cargo-carrying trucks, but experience has demonstrated the necessity for a much heavier type of trailer of the

capacity of 10 tons. Apart from the commercial use of trailers they were first used by the Army as a means of getting around the shortage of engine-transmission and motive-axle production — in other words, to furnish some means of transportation. It was later found that apart from this they were of great value.

30 Their use is indicated where extreme mobility is not required, as certain functions of the Army can be carried out, not on the firing line but at a more or less permanent base. For example advance repair shops need not be as mobile as the fighting army,

Fig. 5 Army Rolling Kitchen Ready for Moving

but rather sufficiently immobile so as to allow of the work being done. There should be, however, sufficient mobility to allow of eventually catching up with the advance forces.

31 A simple illustration will demonstrate the great value of trailers by citing the installation of a tire press on a 5-ton-capacity trailer. This installation included the necessary press, hydraulically operated by hand lever, capable of pressing off the largest solid tire from the rear wheel; the necessary crane equipment to lift the wheel up to the bed of the press; and conveniences for carrying the

necessary tools, etc. This tire press is conveniently mounted on a trailer which can be instantly attached by means of a standard artillery pintle hook to an automotive truck and moved to any desired location. It can then be left at this station and the work of changing over tires proceeded with while the automotive truck can proceed to other work where its power plant is of more use. Such equipment as small repair shops, camp kitchens (see Fig. 5), partially mobile offices, X-ray surgical laboratories, cranes (see Fig 6), etc., can be conveniently mounted on trailers.

32 Other uses of trailers are indicated where for special reasons the truck or tractor is insufficient; for example, a field gun mounted

FIG. 6 CRANE FOR ORDNANCE AND ARTILLERY PURPOSES MOUNTED ON 10-TON TRAILER

on its own wheels is not able to traverse mile after mile of hard road at high speed. It is, therefore, advisable to draw this gun on the trailer platform such as shown in Fig. 7 and tow it behind the truck. Similarly, tanks and caterpillar tractors were prohibited from traveling along the French roads, due to the damage they were said to create, and furthermore, it was found advisable, on account of their slow speed, to have them proceed at a much faster pace. They were therefore also driven under their own power on to the platform of a trailer and towed along the roads at a comparatively high speed.

In addition to the foregoing there is also a certain use in connection with ordinary cargo-carrying trucks, where a small number of trailers have been found advantageous in helping out the trucks in the carrying of supplies.

33 It is believed that standardization of trailers for military use will have to proceed along somewhat different lines than the standardization of trucks. The trailer is especially for emergency uses and all kinds of special work, whereas the platform will vary with the requirements. There is no reason, however, to prevent the standardization of the running gear, such as, for example, the structural framework, the axles, springs, wheels, etc., and the draft gear for towing.

34 There will undoubtedly be discussion as to whether 2-wheel-drive trailer or 4-wheel-drive type trailer will be advisable. The

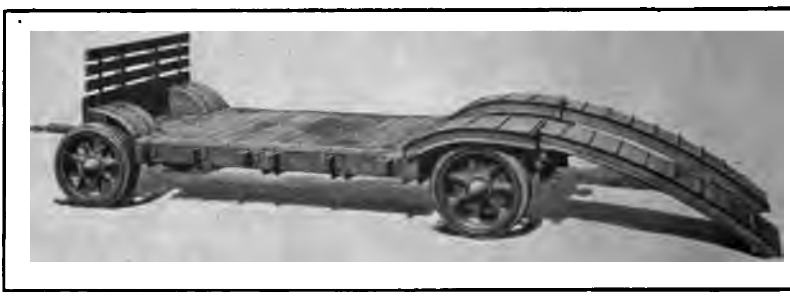


FIG. 7 ARTILLERY 10-TON TRAILER FOR TRANSPORTING TRACK-LAYING TRACTORS, SMALL TANKS, OR GUNS

solution of this is again limited by lack of experience and information which will have to be digested, analyzed and the result derived therefrom. It would appear as if the 2-wheel type of trailer will be of value in the smaller sizes and the 4-wheel type in the larger sizes from 2 tons upward.

SPECIAL TYPES

35 In the military use of motor vehicles, first of all for cargo purposes it became obvious to inventive minds that here was not only an idea which would enable functions previously carried out in a slow way to be speeded up, but an idea which would establish new functions. Special vehicles were asked for and were produced (mobile repair shop (see Fig. 8), vehicles carrying searchlights driven from the engine of the truck as generator, searchlights carried on the

platform of the truck operated from their own engines (see Fig. 9), chemical laboratories, vehicles for the special purification of the water supply, photographic laboratories, lithographic laboratories, vehicles in which the power of the engine was geared to a large winch for the maneuvering of large observation balloons or to a crane thus creating a wrecking car (see Fig. 10), vehicles with integral wireless apparatus, with X-ray apparatus, with special tanks and pumping equipment and heating equipment for degassing and delousing soldiers, disinfecting vehicles, laundry vehicles, etc. Where

Fig. 8 Ordnance Mobile Repair Shop

action was necessary these thoroughly mobile vehicles could be brought forward at once.

36 An interesting case is that of the "degassing outfit," shown in Fig. 11. It was found that the effects of mustard gas could be greatly minimized if not altogether prevented by treating those exposed to a heavy shower of warm water shortly after contact. The soldier would be stripped, given a hot shower, under pressure, for about fifteen seconds, then sprayed with liquid soap, then lathered, then washed off; the whole operation taking about one minute.

He would then receive new disinfected and deloused clothing and a new gas mask and be fit for immediate service again. A large tank truck carrying 1200 gal. of water was fitted with a heating device taken from a steam car, enabling the water to be brought to a good temperature. A centrifugal pump at the front end of the tank, geared to the engine of the truck, pumped this hot water under pressure through hose pipe to a system of gas piping with spray nozzles allowing twenty-four soldiers to have shower baths simultaneously. Attached to this truck was a trailer or sometimes another truck carrying the cleaned clothing. Its mobility allowed it to be kept in close contact with the front line, where full service could be

FIG. 9 SEARCHLIGHT EQUIPMENT USED BY ENGINEER CORPS, MOUNTED ON SPECIAL MACK 6-TON CHASSIS

The searchlight power plant is independent of the truck power plant

given. Incidentally, this was similarly used for delousing purposes and ordinary shower baths.

SPECIAL FEATURES FOR MILITARY WORK

37 Military operation demands special equipment. Trucks operated in convoy over long periods of time tend to make the drivers "dopy"; the short space separating the trucks is not always sufficient to prevent one truck running into another either backward or forward, therefore ample rear and fore bumpers or guards must be provided. Proper towing hooks must also be provided front and rear to enable a disabled truck to continue its journey, for supplies must not be left behind. The Standard Artillery type of pintle

hook is now universal at the rear of American military trucks and at the front two hanging towing hooks are provided, one on each side of the frame.

38 In running in convoy a tremendous amount of dust is raised and often drivers in the middle and rear are almost unable to see through this fog. This dust finds its way through the air intake into the carburetor, into the engine, and causes rapid wear of the piston, cylinders, and other parts. This is a problem peculiar not only to military operation, but also to agricultural-tractor operations. The remedy is found in the provision of an air-cleaning

Fig. 10 A Special Type of Wrecking Apparatus Mounted on a "B" Truck

Note the steadying feet, under the pillars, to ease the load on the frame

device to filter out the dust and grit, allowing only clean air to enter the engine.

39 Extra wide drivers' seats must also be provided. Provision must be made for two soldiers to sit alongside the driver, and soldiers with their heavy overcoats and various accouterment are necessarily bulky. A reserve supply of gasoline must likewise be provided for, and this was done in some of the military trucks by the provision of receptacles like milk cans. It is desirable to have special lamp equipment, on which there should be considerable research done in order to obtain sufficient light to operate but insufficient for the

enemy to notice. At the advanced front even a lit cigarette was not tolerated and driving was done either looking at the stars between the trees or by sound and, in some cases, by instinct. The motor transport driver's job, contrary to general opinion, was among the most hazardous at the front and certainly was one of the hardest, and honor should be given to these men who kept motor transportation operating faithfully through periods of extreme stress in complete darkness, men who were as equal to their task as the machines beneath them.

CONCLUSIONS

40 At various points throughout this paper the question of the Army designing its own vehicles and the question of standardiza-

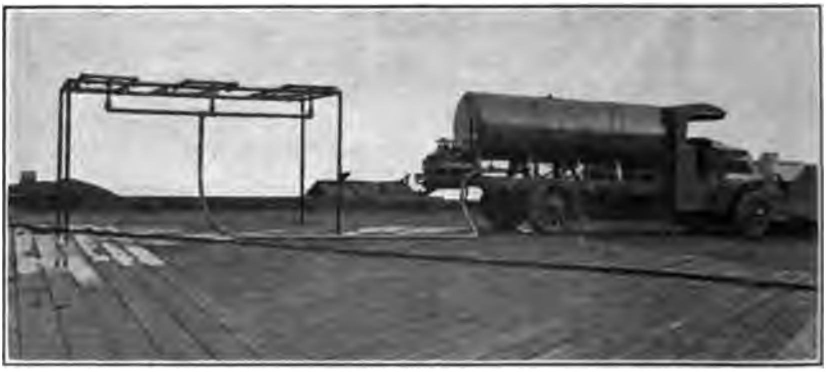


FIG. 11 MOBILE SHOWER BATH, DEGASSING OR DELOUSING OUTFIT

Note the heater at the rear of the tank.

tion have been brought out. In concluding the writer feels that it is insufficient merely to state the classification of vehicles used without drawing attention to the fact that they do not always remain new, but have to be repaired by Army help. In other words, this means using the help at one's disposal without the ability to obtain any other. Furthermore, the vehicles in order to be of economic use must be maintained just as the Fifth Avenue Coach Company, for instance, maintains its omnibuses. That company operates over a definite route, has definite schedules and can maintain a repair shop built of brick or stone with the knowledge that this system will continue for many months, if not years, to come. The active army, on the other hand, in time of warfare does not know from one day to another where it will be a few days later. The maintenance problem must therefore be simplified down to its very elements, and in this it is obvious that with the fewer the types of trucks the fewer will be the number of parts necessarily carried for replacement and the less the need for skilled mechanics. There should be a standardization of one type of vehicle in each classification. Perhaps the Army will decide to adopt some commercial type of vehicle as its standard; perhaps they may decide to develop their own. They should not be subject, however, to political agitation which would seek to provide them with vehicles unsuited for their direct needs. When this standard has been adopted, it should be reasonably adhered to for a period of approximately five years before any serious change is contemplated (unless, of course, there is some tremendously radical alteration in motor-vehicle design).

41 Standardization of equipment, limitation of sizes of tires, of bodies, of lamps and tools and of all the hundred-and-one things that go to make up the various automotive equipment will be found not only desirable but highly essential. A long paper could be written on this subject alone but it is felt sufficient to here only mention this part of the Army's problem and also to state that by carrying this out the Army may be reasonably sure of having its vehicles give them a useful life of, say, ten years, if not more, without any scrappage or undue delays or economic waste due to failure to function. When war comes, as it will come again in spite of all the philosophers, the army will have motor-vehicle equipment whose use is perfectly familiar to its men, whose mechanical details have been developed over a period of years, whose defects are known and can be repaired intelligently and quickly by a trained corps of mechanics, and whose long, continued use has shown the whereabouts for the provision of proper stores for the keeping of spare parts and the necessary supplies. An army equipped like this can go into battle with full confidence that its transportation system will not desert it and confident that that great element, contributing more to the morale of an army than any other factor, will always be present.

TABLE 1 DATA ON TWO PASSENGER CARS STANDARDIZED FOR MILITARY USE

	Cadillac	Dodge
Capacity, no. of passengers	7	5
Power plant	8-cyl. V.L.-Head	4-cyl. L.-Head
Engine size, in	$3\frac{1}{8} \times 5\frac{1}{8}$	$3\frac{7}{8} \times 4\frac{1}{2}$
Piston displacement, cu. in	314.4	212.3
Gear ratio — 1st	15.49: 1	19.37: 1
2nd	7.69: 1	7.98: 1
3rd	4.437: 1	4.16: 1
Reverse	18.35: 1	19.37: 1
Tire size — Front, in	$36 \times 4\frac{1}{2}$	33×4
Tire size — Rear, in	$36 \times 4\frac{1}{2}$	33×4
Piston displacement per ft., high	148	102.3
Piston displacement per ft., low	516	476
Weight of car loaded, lb	5430	3385
Weight of car empty, lb	4310	2585
Piston displacement per 1000 lb. per ft., high	27.4	30.2
Piston displacement per 1000 lb. per ft., low	95	140.5
Speed on high in m.p.h. at 1000 r.p.m. of engine	23.8	23.6
Frame section, in	$\frac{7}{16} \times 1\frac{3}{4} \times 8$	$\frac{5}{32} \times 2\frac{1}{2} \times 4$
Width and length of front spring, in	2×42	2×36
Width and length of rear spring, in	2×54	$2 \times 43\frac{1}{2}$
Overall length, in	190	153
Overall width, in	68	65
Ground clearance — new tires, in	$8\frac{7}{8}$	$9\frac{3}{8}$
Size service brake, in	$2\frac{1}{4} \times 17$	$2\frac{1}{4} \times 14$
Size emergency brake, in	$2\frac{1}{4} \times 16\frac{1}{8}$	$2\frac{1}{4} \times 13\frac{7}{8}$
Nature of oiling system	Pressure	Splash
Oil pump	Gear	Eccentric
Oil-pump drive	Sp. Gear Camshaft	Sp. Gear Camshaft
Type of carburetor	Own	Stromberg
Type of rear axle	Sp. Bev. Full Float	Sp. Bev. Full Float
Type of radiator	Tube and Plate	Flat Tube
Wheelbase, in	125	114
Tread, in	56	56
Turning circle, ft	45	40
Gasoline capacity, gal	20	15
Type of governor	None	None
Ignition	Battery	Battery

TABLE 2 DATA ON THE 4-WHEEL-DRIVE TRUCKS, WITH A FEW COMPARISONS WITH THE "B" TYPE

	Four-Wheel-Drive Truck	Nash Truck[1]	Ordnance Truck-Tractor	Type "B" Truck
Motor	Wisconsin	Buda	Wisconsin	...
Cylinder dimensions, in.	4¾ × 5½	4¼ × 5½	4¾ × 5½	4¾ × 6
Displacement, sq. in.	389.9	312.5	389.9	425
Clutch	Hele-Shaw 3 Speed & Rev.	Borg & Beck 4 Speed & Rev.	Hele-Shaw 4 Speed & Rev.	...
Transmission ratio			Truck. Tractor	...
High speed	8.8 to 1	8.5 to 1	8.99 to 1	12.05 to 1
First speed	35.5 to 1	39.6 to 1	53.5 to 1	71.5 to 1
	Non-Adj.	Non-Adj.	Adj. Chain	
Transfer case	Chain	Chain	—	...
Axles	Full Floating	Int. Gear	Int. Gear	
Differential	3 Bevel	2 M & S	3 Large M & S	...
Springs	5 Semi-Elip.	4 Semi-Elip.	4 Semi-Elip.	...
Frame sizes, in.	5¼ × 2 × ¼	5 × 2 × ¼	7 × 3 × ¼	...
Steering gear	Ross	Lavine	Q. M. "B"	...
Brakes				
Service	Trans. 10 × 3½	Wheel 16¾ × 2½	Live Axle 9 × 4½	...
Emergency	Wheels 15¼ × 2¾	Trans. 8 × 2½	Trans. 9 × 4½	...
Wheel base, in.	122	122	126	...
Tread, in.	56	61	63	...
Tires, in.	36 × 6	36 × 6	36 × 7 Single	...
Weight (lb). with ammunition body	8,421	8,044	9,200	...
Capacity, lb.	6,000	4,000	6,000	...
Gross weight, lb.	14,421	12,044	15,200	...
Ground clearance, in.	9½ & 15	11 & 18	14 & 21	...
Drawbar pull, lb.	4,000	3,200	7,000	...
Actual weights, lb.				
Empty (all equip. & drivers, tools, etc.)	8,660	8,480	10,670	11,240
Loaded, Front Axle	5,780	5,700	7,760	5,510
Rear Axle	7,040	5,560	7,910	11,480
Total	12,890	11,330	15,690	17,015
Pay Load	4,230	2,850	5,020	5,775
Gross weight with full pay load	14,660	12,480	16,670	21,240
Piston displacement per ft. moved				
High gear, cu. in.	363	284	500 or 375	390
per 1000 lb gross	24.6	22.6	30 or 22.4	18.2
First gear, cu. in.	1,470	1,320	3000 or 2250	2,340
per 1000 lb. gross	100	106	180 or 134	109
Piston displacement per ft. per 1000 lb. pay load	60.5	71	83 or 62.5	39

[1] The Nash is the Marine Corps Standard.

TABLE 3 DATA ON ¾-TON-TYPE VEHICLES

	Class "AA"[1]	G.M.C.[2]	Commerce	White
Capacity, lb.	1500	1500	2000	2000
Power plant	4-cyl. Block	4-cyl. Block	4-cyl. Block	4-cyl. Block
Engine size, in.	4 × 5	3¾ × 5	3¾ × 5	4⅛ × 6¾
Piston displacement, cu. in.	251.3	220.9	220.9	361.7
Max. torque, lb.	1666	1560	1560	2700
Gear ratio, 1st	22.3: 1	21.9: 1	16.67: 1	14.25: 1
2nd	9.78: 1	9.48: 1	9.77: 1	7.74: 1
3rd	5.54: 1	6: 1	5.21: 1	4.66. 1
4th	...	...	...	3.56: 1
Reverse	27.3: 1	26.6: 1	20.2: 1	19.02: 1
Tire size — front, in.	35 × 5	35 × 5	36 × 6	36 × 6
Tire size — rear, in.	35 × 5	35 × 5	36 × 6	36 × 6
Weight of truck loaded, lb.	7850	7443	7720	8350
Weight of chassis only, lb.	3290	2883	3160	3790
Torque, in-lb. per 1000 lb. per ft., high	124.4	132.8	106	155
Torque, in-lb.per 1000 lb. per ft., low	464	451	304	440
Speed on high in m.p.h. at 1000 r.p.m. of engine	19.4	17.4	20.6	23.1
Maximum frame section, in.	3/16 × 3 × 6	5/32 × 2¼ × 4½	3/16 × 3 × 6 1/16	7/16 × 3½ × 4 7/16
Width and length of front spring, in.	2¼ × 42	2¼ × 38	2¼ × 36	2 × 40½
Width and length of rear spring, in.	2½ × 58	2½ × 54	3 × 50	2½ × 50½
Overall length, in.	199	188		
Overall width, in.	66	68	66	68
Size of emergency brake, in.	2½ × 15¾	1¾ × 13½	2¼ × 14½	2½ × 11¾
Size of service brake, in.	2½ × 16	1¾ × 14	2¼ × 15	2½ × 17
Ratio conn. rod bearing to area of piston	0.36: 1	0.369: 1	0.369: 1	0.4: 1
Ratio camshaft bearing to area of piston	1.51: 1	1.59: 1	1.59: 1	Ball Brgs.
Nature of oiling system	Pressure	Splash Pr.	Splash Pr.	Splash Pr.
Oil pump	Gear	Sing. Plung.	Sing. Plung.	Double Plung.
Oil-pump drive	Spiral Gear Camshaft	Eccentric Camshaft	Eccentric Camshaft	Eccentric Camshaft
Type of rear axle	Full Floating Sp. Bevel	¾ Floating Bevel	Int. Gear	Semi-Floating Sp. Bevel
Type of radiator	Finned Tube	Tube	Finned Tube	Cellular
Wheelbase, in.	130	132	133½	140
Tread fronts, rear, in.	56	56	56	56
Turning circle, ft.	44	46	52	56
Gasoline capacity, gal.	20	24	20	19
Type of governor	Flyball	Suction	None	None
Type of carburetor	U.S.A.	Marvel	Zenith	Own
Ignition	Mag. & Batt. Double	Magneto Single	Magneto Single	Magneto Single
Ground clearance, in.	11	8¾	10½	8½

[1] The "AA" was the truck designed by the Motor Transport Corps, but was completed too late to be approved for production.

[2] The G.M.C. was adopted as the official standard and was used extensively for ambulances.

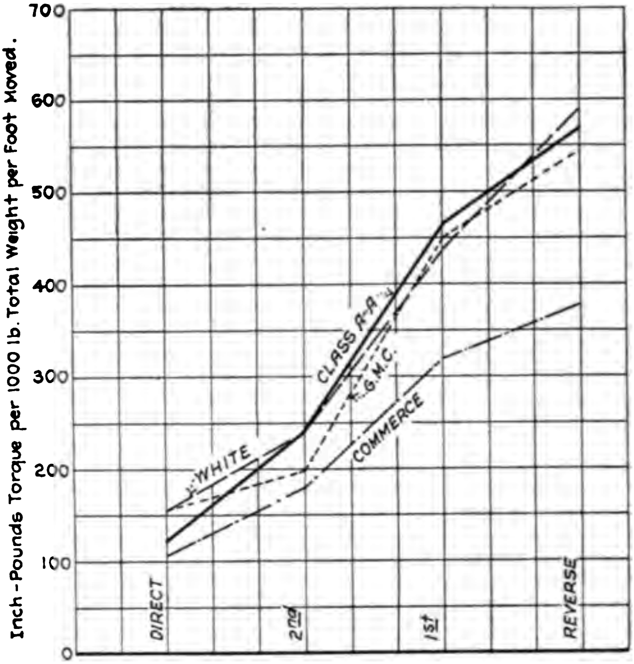


FIG. 12 CHARACTERISTIC TORQUE CURVES OF ¾-TON-TYPE VEHICLES WHOSE SPECIFICATIONS ARE GIVEN IN TABLE 3

(The torque is measured at the rear wheel in in-lb. per 1000 lb. per ft., different gear ratios being used.)

During the war the ¾-ton trucks found many uses, because they were capable of traveling over exceedingly rough ground. For this reason they made excellent 4-patient ambulances. They were also valuable for reconnaissance as they could easily carry a crew of men with the necessary range finders, instruments, tools and guns. This type of vehicle also made an excellent machine-gun attack.

TABLE 4 DATA ON 1½–2–TON–TYPE VEHICLES

	Class "A"[1]	Garford	Pierce	Packard	White[2]
Capacity, lb.	4,000	4,000	4,000	4,000	3,000
Power plant (4 cyl.)	Block L	Block L	Pairs T	Block L	Block L
Engine size, in.	4¼ × 5½	4¼ × 5½	4 × 5½	4 × 5½	3¾ × 5⅛
Piston displacement, cu. in.	312	312	276.5	276.5	226.4
Max. torque, lb.	2268	2100	1800	2000	1575
Gear Ratio — 1st solid tire	40.4: 1	2.601: 1	42.82: 1	33.3: 1	42.7: 1
2nd solid tire	26.45: 1	11.63: 1	20.79: 1	18.65: 1	19.2: 1
3rd solid tire	13.6: 1	7.75: 1	12.87: 1	10.76: 1	12.1 Direct
4th solid tire	8: 1	None	8.25: 1	7.25: 1	9.1: 1
Reverse solid tire	40.9: 1	33.48: 1	57.08: 1	43.7: 1	56.4: 1
Gear Ratio — 1st pneu. tire	40.4: 1	26.04: 1	36.3: 1	33.3: 1	32.9: 1
2nd pneu. tire	26.45: 1	11.63: 1	17.6: 1	18.65: 1	14.8: 1
3rd pneu. tire	13.6: 1	7.75: 1	10.91: 1	10.76: 1	9.25: 1 Direct
4th pneu. tire	8: 1	None	7: 1	7.25: 1	6.99: 1
Reverse pneu. tire	40.9: 1	33.48: 1	48.4: 1	43.7: 1	43.47: 1
Weight of truck loaded, lb., solid tires	10,765	10,530	11,160	10,400	9700
Weight of truck loaded, lb., pneumatic tires	10,580	9,900	10,170	10,020	9,540
Weight of chassis, only, lb., solid tires	5,565	5,330	5,960	5,200	4,500
Weight of chassis, only, lb., pneumatic tires	5,380	4,700	5,970	4,820	4,340
Speed in high in m.p.h. at 1000 r.p.m. Eng. solid tire	13.45	13.9	12.2	13.9	9
Pneu. tire	14.9	15.4	17	15.7	12.9
Max. frame section, in.	¼ × 3 × 6	¼ × 2½ × 5	5/16 × 2 1/16 × 6 1/16	1¾×1 5/32×6 RO	¼ × 3 × 5¼
Width and length—front spring, in.	2½ × 42	2½ × 42	2½ × 41	2½ × 41	3 × 41
Width and length — rear spring, in.	3 × 56	2½ × 50	3 × 54	2½ × 46	3 × 48
Overall length, in.	210¾	223½	218½	220	202
Overall width, in.	72½	68	71½	74¾	73
Size of emergency brake, in.	2½ × 18	1¾ × 18	3 × 17¼	2½ × 17	3½ × 17
Size of service brake, in.	4 × 8½ Trans.	1¾ × 18	3¾ × 10 Trans.	4¾ × 10 Trans.	3½ × 18
Ratio conn. rod bear. to area piston	0.396: 1	0.374: 1	0.442: 1	0.381: 1	0.4:1
Ratio crank shaft to area of piston	1.506: 1	1.57: 1	1.31: 1	1.87: 1	Ball brgs.
Nature of oiling system	Pressure	Sp. pressure	Pressure	Pressure	Sp. pressure
Oil pump	Gear	Gear	Gear	Gear	Dbl. plunger
Oil-pump drive on cam shaft	Sp. gear	Sp. gear	Sp. Gear	Sp. Gear	Eccentric
Type of rear axle	Internal Gear	Worm, full-floating	Worm, full-floating	Worm, full-floating	Double rod, semi-float'g
Type of radiator	Vertical Finned Tube	Cellular	Vertical Finned Tube	Cellular	Cellular
Wheel base, in.	144	142	150	144	145½
Tread, front or rear in.	F .60 R 61¾	F .56 R .60¾	F .59¾ R .56½	F .58½ R .55½	56 56
Turning circle, ft., solid tire	......	......	48½	56	54
Gasoline capacity, gal.	23	30	30	30	20
Type of governor	Fly Ball	None	Fly Ball	Cent'f'l	Suction
Type of carburetor	U.S.A.	Rayfield	Own	Own	Own
Type of ignition	Mag. & Batt. Double	Magneto Single	Mag. & Batt. Dual	Mag. & Batt. Dual	Magneto
Ground clearance, in., new tire	12¾	9½	9¾	9¾	8½

[1] Developed by the Motor Transport Corps, but not adopted. [2] Adopted as the official standard.

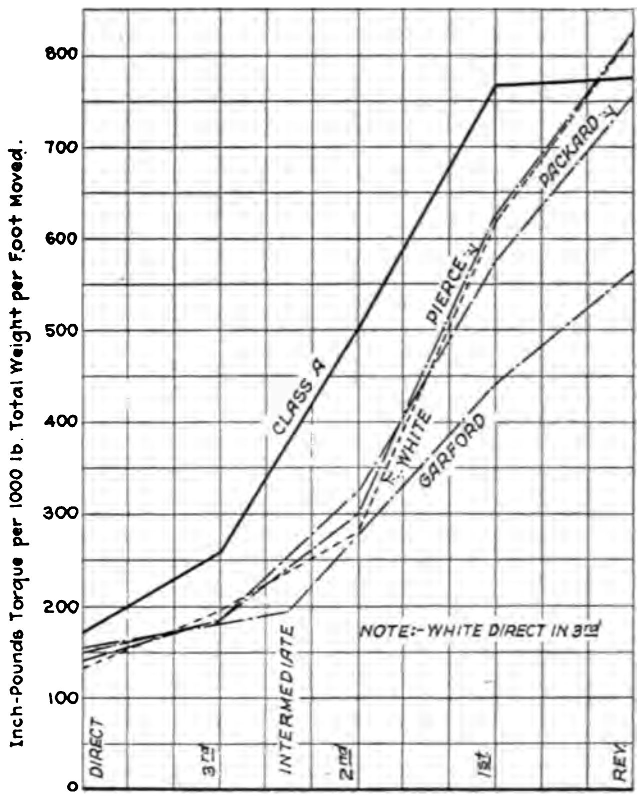


FIG. 13 CHARACTERISTIC TORQUE CURVES OF 1½-TON-TYPE VEHICLES WHOSE SPECIFICATIONS ARE GIVEN IN TABLE 4

(The torque is measured at the rear wheels in in-lb. per 1000 lb. per ft., different gear ratios being used.)

The 1½-ton truck was looked upon by the Army as the maid of all work. Cargo of all kinds, men, supplies, food and munitions were carried by this steady-going vehicle. This type of truck was able to run at 16 m.p.h. and because of its light weight was able to go over the type of bridge usually found. In other words, this vehicle replaced in large measure the standard mule escort of the old Quartermaster's Department.

TABLE 5 DATA ON THE 5-TON-TYPE VEHICLES

	Class "B"[1]	Mack[3]	Pierce-Arrow
Capacity, lb	10,000	11,000	10,000
Power plant	4 Cyl. Prs. L.	4 Cyl. Prs. T.	4 Cyl. Prs. T.
Engine size, in	$4\frac{1}{4} \times 6$	5×6	$4\frac{3}{8} \times 6$
Piston Disp., cu. in	425.3	471.2	448
Max. torque, in-lb	3120	3078	3096
Gear Ratio — 1st	56.7 to 1	35.6 to 1	36.86 to 1
2nd	31.1 to 1	22 to 1	20.7 to 1
3rd	16.8 to 1	12.8 to 1	9.75 to 1
4th	9.56 to 1	....	....
Reverse	56.7 to 1	48 to 1	47.5 to 1
Tire size — front, in	36×5	36×6	36×5
Tire size — rear, in	40×6D	40×6D	40×6D
Weight of truck loaded,[2] lb	19,980	20,240	19,762
Weight of chassis only, lb	8180	8440	7962
Torque, in-lb. per 1000 lb. per ft., high	135.5	176.5	138.8
Torque, in-lb. per 1000 lb. per ft., low	677	439	442
Speed on high in m.p.h. at 1000 r.p.m. of engine	12.44	10.37	12.12
Frame section, in	$8 \times 3 \times \frac{1}{4}$	$\frac{7}{32} \times 3 \times 8$	$8\frac{1}{2} \times 2\frac{9}{16} \times \frac{9}{16}$
Width and length front spring, in	44×3	$3\frac{1}{2} \times 46$	$2\frac{1}{2} \times 33$
Width and length rear spring, in	58×4	4×52	3×48
Overall length, in	$244\frac{1}{2}$	222	$238\frac{1}{2}$
Overall width, in	155	$90\frac{3}{4}$	....
Size emergency brake, in	$24 \times 2\frac{1}{2}$	$20 \times 3\frac{1}{2}$	$19\frac{1}{2} \times 4\frac{1}{2}$
Size service brake, in	$24 \times 2\frac{1}{2}$	15×3	10×5
Ratio conn.-rod area to area piston	0.402 to 1	0.393 to 1	0.35 to 1
Ratio crankshaft area to area piston	1.53 to 1	1.62 to 1	1.15 to 1
Nature of oiling system	Force	Force	Gravity
Oil pump	Gear	Gear	Gear
Oil-pump drive	Helical (Camshaft)	Helical (Camshaft)	Helical (Camshaft)
Type of rear axle	Worm-Bevel Diff. Full	Chain	Worm Spur Diff. Full
Type of radiator	Vert. Tube	Tubular-Dash	Vert. Tube
Wheelbase, in	$160\frac{1}{2}$	156	168
Tread, front rear, in	F. 64 R. $64\frac{1}{2}$	F. 68 R. $74\frac{1}{2}$	F. 68 R. 64
Turning circle, ft	$56\frac{1}{2}$	50	$51\frac{3}{4}$
Gasoline capacity, gal	31	25	25
Type of governor	Fly Ball	Centrifugal	Fly Ball
Type of carburetor	U.S.A.	Stromberg	Own
Ton-miles pay load per gal. gas	22.3	25.4	19.8
Ignition	Mag. & Batt. Double	Magneto Single	Mag. & Batt. Dual
Ground clearance, new tires	$11\frac{3}{4}$	$10\frac{3}{4}$	10
Ground clearance, old tires	$9\frac{3}{4}$	$8\frac{3}{4}$	8

[1] The "B" Type is the official standard.
[2] Includes water, full lighting equipment, dash and floor boards only.
[3] The gear ratios of the first Mack trucks were as in this table. Later these were changed to be more in accord with those of the Class "B" trucks.

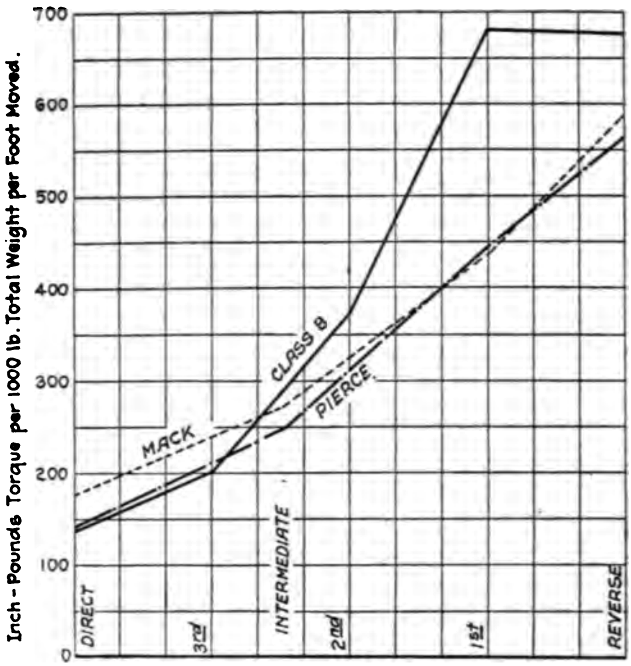


FIG. 14 CHARACTERISTIC TORQUE CURVES OF THE 3–5-TON TYPE VEHICLES WHOSE SPECIFICATIONS ARE GIVEN IN TABLE 5

(The torque is measured at the rear wheels in in-lb per 1000 lb. per ft., different gear ratios being used. The Mack was later changed, nearer the Class "B".)

The Great War is responsible for the creation of the standard 3–5-ton truck of the Army. Because of its weight the commercial type is practically limited to use in and about cities, and for this reason the Government designed and developed in 1917 the standard Liberty or "B" type truck. Approximately 18,000 of these trucks were manufactured and the special points required by military use were built into the truck and not added as an afterthought. A comparison of Tables 3, 4 and 5 will show that this truck possesses approximately 50 per cent more torque than any similar commercially designed vehicle.

DISCUSSION

B. F. MILLER[1] divided motor vehicles for the United States Army into two classes, those owned and maintained by the War Department in time of peace and those required as a result of an emergency and which must be supplied after the commencement of the emergency.

He believed that the motor vehicles owned and maintained during peace times would be but a small fraction of the requirements for war. When required in time of war, the additional vehicles necessary could be secured by commandeering previously surveyed and registered motor vehicles in accordance with agreements made in advance with subsidized owners. To accommodate the increased personnel mobilized at the beginning of the emergency and subsequently and for replacements, motors would be designed and produced.

He next referred to the use of the Ford for work near the lines where the roads were bad and of the four-wheel driven truck for cargo as well as off-road hauling. In his opinion the use of trailers should be limited to hauls obviously suited to them and for mounting certain special equipment and bodies that are required to be moved infrequently.

He agreed with the author that a cargo-carrying, track-laying type of motor vehicle should be developed without delay to replace animal transportation in such places as the Mexican border and in general over bad roads in rural communities. He was of the opinion that it should be of 1½ to 2 tons capacity and designed for the purpose for which it was intended rather than combining standard parts of trucks and tractors into a makeshift.

F. H. POPE[2] (written). I have read Mr. Younger's paper with great interest and consider it a very able presentation of the subject of types of vehicles needed for military use. Naturally, in a subject of this kind, about which so much controversy has raged, there still exist certain honest differences of opinion. Therefore, in this discussion of the paper, it is desired to set down certain arguments, pro and con, in reference to several details touched on by Mr. Younger. The information of the writer has been acquired through continuous service with Army Motor Transport for a period of nearly

[1] Captain, U. S. A. Formerly Lieutenant-Colonel, Motor Transport Corps.
[2] Colonel, U. S. A., Motor Transport Corps.

four years, ranging from the command of a Motor Truck Company in Mexico to the organization and operation of the Motor Transport overseas.

In the first place, I can see but very little difference between the principles of military truck operation and commercial truck operation. They both have the same result to accomplish; the former controlled by the necessity of "results now" at the expense of cost, and the latter controlled by the cost element. Motor transport, in quantity, cannot operate without roads. Doubtless, there exists the necessity of more rugged truck construction in military life than in the construction of certain specialized commercial types, such as the Fifth Avenue Busses. Our Army experience is apt to be misleading on this point, as the great cause for vehicle breakdown was poor operation. The correction of this fault is the job of the training man, not the job of the constructor.

Passing to the controversy relative to the 1½-ton and the 3-ton type as standard: In the Fall of 1916, urgent recommendation was made to the War Department from the Southern Department by those in actual touch with the motor transport work along the border, to adopt this 3-ton type as standard. As stated by Mr. Younger, the advisability of the 3-ton type was appreciated by both the English and French. But I would like to call attention to the fact that this type was urgently recommended for our Army in 1916 by those officers of actual motor transport experience. I would like to call attention here to a minor objection, which consists in speaking of a 3–5-ton truck. It is either a 3-ton or 5-ton. It would seem to be as logical to call it a 0–5-ton truck as starting at the figure 3. The truck user, especially in military life, is concerned with the truck capacity, and has no interest in a nomenclature that the commercial producer may have instituted for various reasons.

In military life, the 1½-ton truck has its special value over a heavier truck in being more rapid, entailing less loss of capacity in bulky loads, and being the truck par excellence for personnel carrying. It is open to the objection to increased number of operating personnel and increased road space, mentioned by Mr. Younger. I do not think it can negotiate, in quantity, worse roads than a 3-ton truck. Now, as its special value lies in its increased speed, I think that the proposition of equipping with pneumatic tires all 1½-ton trucks in military service is one that should be seriously considered.

Mention is made of certain special type vehicles, or rather, special body types, both in the passenger and cargo types. The essential principles governing special types, I believe to be these: Special vehicles should be looked upon as luxuries and should not be allowed if a standard type would answer the purpose. Motor cars and trucks should be districted by make or model, either in divisions or territorial districts. This brings us to the subject of the four-wheel drive type. These have a very positive value, chiefly for artillery work where their tractor properties can be utilized, but they are also valuable in cargo work over certain kinds of mountainous or difficult country. For cargo work, their use should be determined by the terrain. The caterpillar type is the only one that can negotiate really difficult territory. It is not believed that the so-called caterpillar adapters have been brought to a successful solution. In the light ambulance illustrated, the adapter appears to be successful, but what about the wounded patriot lying within the ambulance.

The use of trailers, in my opinion, is by no means as extensive as it should be. That more use is not made of trailers is due to the unwillingness of the vehicle operator to be bothered with them.

In the last analysis, the great problem of military motor transport is to secure the very great volume of vehicles that are suddenly demanded by an emergency, such as confronted us in 1917. As far as concerns the increased periods of serviceability of vehicles we may obtain, this is a matter that will be taken care of by a thorough training scheme based on our past experience. But, how will the greatly increased production demand be met by the automotive industry? Policy and prudence dictate the closer approximation of commercial and military standards.

A. F. Masury wrote that he thought the author's paper largely a brief for the adoption of a special vehicle and he wished to have a few words on the other side of the argument.

He pointed out that the splendid work of the Engineer Corps during the war was due largely to their willingness to use standard materials and articles rather than to demand special types as yet unproduced. Undoubtedly a specially designed vehicle to meet the requirements of the army would be superior to a commercial vehicle if for no other reason than on account of standard parts. But as this is a peaceful nation the War Department cannot be expected for carry on hand a large stock of up-to-date vehicles. The only alternative is to have on hand at various factories blueprints and specifications of these special vehicles and, in event of war, to change these factories over into the manufacture of war vehicles. To such a plan, every manufacturer would naturally object. For these reasons the best expedient would seem to be to use commercial vehicles as largely as possible and restrict the specially designed vehicles to those of special usage.

In Fig. 9 the author shows a searchlight truck of special type built for the Engineer Corps and says that the power plant is independent of the truck power plant. Some of this type were furnished, but the particular photograph shown is of a truck in which the generator for the searchlight is directly connected to the truck engine. The generator is contained in the frame extension in front. One hundred and eighty of this type of truck were built in this country and 40 were built up in France using French generators mounted upon another type of American truck.

It should be noted in connection with the figures of Table 4 that the engine torque of the Pierce-Arrow truck is apparently based upon a higher mean effective pressure than that of the other trucks as the piston displacement is slightly less. In comparing the transmission efficiency of the trucks in this table it should be noted that by actual test the overall efficiency of the chain drive is much better than the worm drive when operating in low gear, the difference being due to the great friction in the worm at high tooth pressures. This accounts for the fact that a chain drive truck will often perform better in pulling out of a hole than a worm drive of greater ratio.

A. J. Slade advocated the standardization of motor trucks through the use of parts and equipment which have become recognized as standard and which can readily be assembled in any of the leading automobile plants of the country. By this means, trucks of standard design could be produced in large quantities, the repairs and replacements would be facilitated, and the confusion would be avoided which comes from the use of the multiplicity of design or of standardized trucks in which parts of special design are incorporated.

He said that while the European War Departments had for many years prepared comprehensively for the use of motor vehicles, our own War Department had not done so previous to the war except to the extent of purchasing some 80 or 90 motor trucks, and of organizing a number of truck trains during the Mexican trouble which were manned by civilian employees furnished by the manufacturers.

Upon our entrance into the war, plans for motorized trains for use by the A. E. F. were taken up by the various staff corps, and as the Quartermaster Corps was at that time responsible for all transportation, the most conspicuously prominent work in this direction was done by that corps.

In July 1917, contracts were placed for several sizes of Pierce Arrow, Packard, Locomobile, Garford, F.W.D. and Nash trucks. If standardization had been in mind at that time, it could hardly have been possible to have selected vehicles which varied so greatly from one another so far as interchangeability of parts and units was concerned.

Following this, the design of new trucks in three sizes was taken up, which, while conventional so far as their general type was concerned, incorporated specially-designed engines, transmissions, axles and other units not already in commercial production.

This program got as far as the production of models of the different sizes, but quantity production was never started, except in the case of the 3-5-ton Quartermaster Class B truck. At the same time that the Quartermaster Corps was making its purchases the Medical Department arranged for the production of Ford ambulances and for larger ambulances mounted on ¾-ton general motors chassis, and the Engineer Corps contracted for a large number of Mack trucks.

About this time, learning that the Quartermaster Corps could not meet the needs of the Air Service, the latter undertook the construction of trucks comprised of commercial units such as engines, transmissions, axles, steering gears, etc., which were being produced in the largest quantities by established manufacturers and could be and were assembled by a large number of motor truck builders. Incidentally, this was the method of standardization which the Society of Automotive Engineers, working with the Quartermaster Corps, had considered, early in 1917, but for some reason the plan had been discarded in favor of the design and production of entirely new vehicles. While these several efforts were in progress, the A.E.F., finding itself in great need of motor trucks, had purchased considerable quantities of British makes, the Air Service had some French and Italian trucks, and a little later, in the summer of 1918, still other equipment began to arrive from the United States, including White and Commerce trucks and the Quartermaster Class B trucks, so that there were about thirty makes or types used in the A. E. F. in considerable numbers.

Continuing, Mr. Slade said, "The principal point which strikes me in reading Mr. Younger's paper is that the so-called standardized types of vehicles at present adopted or approved by the War Department are still types which vary from one another quite substantially, and I am frankly at a loss to know on what basis of experience data the selections were made. All but one are of commercial types, and of the commercial types one is of the so-called manufactured type and another of the so-called assembled type.

"Another point mentioned in Mr. Younger's paper in which the most intelligent opinion of the A. E. F. will disagree, has to do with the operation of trucks off the roads. Even a caterpillar truck can sink in the mud if there is no foundation. A four-wheel-driven vehicle will go farther into difficult situations than a rear-wheel-driven vehicle, but any vehicle rolling on wheels requires a supporting foundation and before motor trucks, and to a certain extent passenger cars, could keep pace in the rear of the advancing troops, it was always necessary to repair the worst spots in the roads, especially those caused by trenches, shell holes, etc.

"The most important point in connection with military motor transport, touched on only very casually in Mr. Younger's paper, has to do with the training of experienced personnel, and intelligent organizations for operation and maintenance. With the most highly refined and developed types of equipment, military motor transport in the mass cannot function effectively without systematic organization, including road traffic control and road construction, as well as convoy operation; nor without effective highway traffic regulations.

"I feel strongly that now is the time, when army reorganization is under consideration, to plan how units of personnel can be organized, familiarized with the equipment which they will have to use and the methods under which they will operate.

"I would like to leave the thought that the solutions of these intricate questions concerning equipment, organization and personnel can best be found by a commission of really qualified motor transportation experts experienced in investigation work who have been associated with the varied M.T.C. activities in the A.E.F. and who, in coöperation with a board of Army officers who have also had similar experience in military motor transportation, especially with combat divisions, could work out a sound constructive program which would justify the support alike of the regular army and the general public."

G. R. YOUNG [1] wrote that the question of standardization was fundamental in the determination of motor transport policy. The author had set forth very clearly the necessity of types of motor vehicles for military use which were not always constructed in accordance with existing commercial practice. This fact was one reason for the adoption of a standardized program. Another reason which had been touched upon only slightly was the importance of reducing as much as possible the variety of makes and types of vehicles in the army.

The writer continued by stating the reasons for the adoption by the government of the standardized program and an exposition of it.

WILLIAM P. KENNEDY said that it must be recognized that commercial manufacturers will insist upon the supply to the government of commercial trucks when a demand arises for military trucks. This being so, the question arises is it desirable to standardize exclusively in any other direction. If preparation for the future is of vital importance, standardization must be effected by means of compromise between army and commercial interests. If trucks are developed by army interests alone for particular military requirements there will be difficulty in producing them in the event of a crisis. A truck which is satisfactory may be developed, but if the type is perpetuated it will become obsolete in view of progress certain to take place in the commercial field. The compromise to be effected is to induce manufacturers to accept government guidance through an organization to consist of government engineers on the one hand and commercial engineers representing prominent manufacturers on the other.

THE AUTHOR. The question of personnel raised by Colonel Slade is of tremendous importance. The automobile industry had to furnish personnel for the expansion of the aircraft, tank, tractor and submarine chaser. For this reason there was left for the operation of trucks only an insufficient supply of intelligent operators.

As Mr. Kennedy had said, it was highly probable that a compromise between the government and manufacturers would have to be effected.

[1] Captain, U. S. A., Chief, Engineering Branch, Motor Transport Corps.

No. 1728

AN INVESTIGATION OF STRAINS IN THE ROLLING OF METAL

By Alfred Musso, New York, N. Y.
Associate-Member of the Society

In every manufacturing process, while raw stock is being transformed into a finished article, a certain amount of material is incidentally wasted, and the minimizing of such a waste constitutes one of the most important problems to be solved by the operating engineer. Rolling-mill operations are no exception to the general rule and the problem may be specifically set forth as follows:

What is the most convenient length and width of the piece of metal to be rolled, in order to produce a finished article of certain definite dimensions, so that the waste of material may be reduced to a minimum?

A complete answer to this complex question depends on data of various natures resulting from actual investigation. We may, however, group these data in two classes: To the first class belong all data of a general character obtained through mechanical investigation; and to the second, data obtained through investigation of the special condition under which the manufacturing takes place. The scope of this article is the exposition of some fundamental principles leading to data of the first class.

WHEN a piece of metal is put through the rolls in a rolling mill, its linear dimensions are strained and the whole piece itself is deformed; in other words, the thickness of the piece becomes smaller, while its length and width are increased. These strains are the effect of the pressure which the rolls exert on the piece, and as introductory to our specific subject we will first consider the behavior of a piece of metal under compression.

2 Let $ABCD$ (Fig. 1) be the cross-section of a piece of metal as subjected to the pressures P and P_1 normal to the faces shown in cross-section as AB and CD. When the compression is carried beyond the elastic limit of the material, the two faces AB and CD will come closer together as shown by the dotted lines A_1B_1 and C_1D_1, while the two side faces AC and BD will bulge outward as shown by the dotted curved lines A_1C_1 and B_1D_1, and if the forces P and P_1 are still increased the piece will ultimately fail by shearing along its diagonals.

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

3 The ultimate behavior of the failing piece is influenced by the structure of the metal itself. In fact, experience shows that ductile metals possessing a homogeneous structure, such as copper, aluminum, etc., will become plastic and flatten down to a disk, while fibrous metals, such as steel, wrought iron, etc., where strength across the grain is much lower than with the grain, are not susceptible of any plastic state at all and will ultimately fail by splitting sideways. As it is beyond our present scope, however, to discuss this subject of the failure of metals under compression, it is sufficient for our purpose to regard the piece as shown by Fig. 1 as divided into six pyramids having as a common vertex its pressure center O, and as bases its six faces.

4 It is obvious that while the compressive forces P and P_1

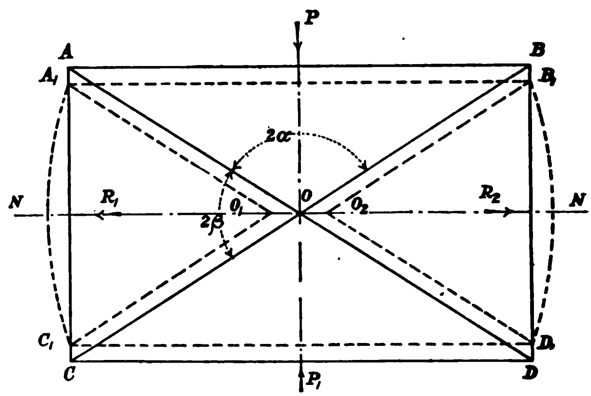


FIG. 1 CROSS-SECTION OF A PIECE OF METAL SUBJECTED TO PRESSURES P AND P_1

are tending to telescope into each other along the pressure line PP_1 of the upper and the lower pyramids, the other four pyramids will be pushed outside by the expulsive forces R_1 and R_2 consequently set up by the aforesaid pressures P and P_1. The value of the expulsive forces evidently depends on the two angles AOB and AOC, ordinarily known as *pressure angle* and *expulsion angle*, respectively. We will now apply the foregoing considerations to the rolling-mill process, and begin our investigation of the effects of the roll pressure at any point of the surface of the piece.

5 Let Fig. 2 represent a cross-section of a rolling mill through the rolls. NN_1 is the piece of metal going through the rolls C and C_1. Inasmuch as the pressure is exerted radially we may consider it at any point, say at A, which is the initial point of the contact arc

AMB. The pressure P may be resolved into two components, one normal to the piece and the other parallel to its surface, thus:

$$n = P \cos \phi, \text{ normal component}$$
$$a = P \sin \phi, \text{ parallel component}$$

where ϕ = angle ACB, known as the *approach angle*. The directions of the components clearly show that while n is the force which actually compresses the piece, a is the component of the pressure

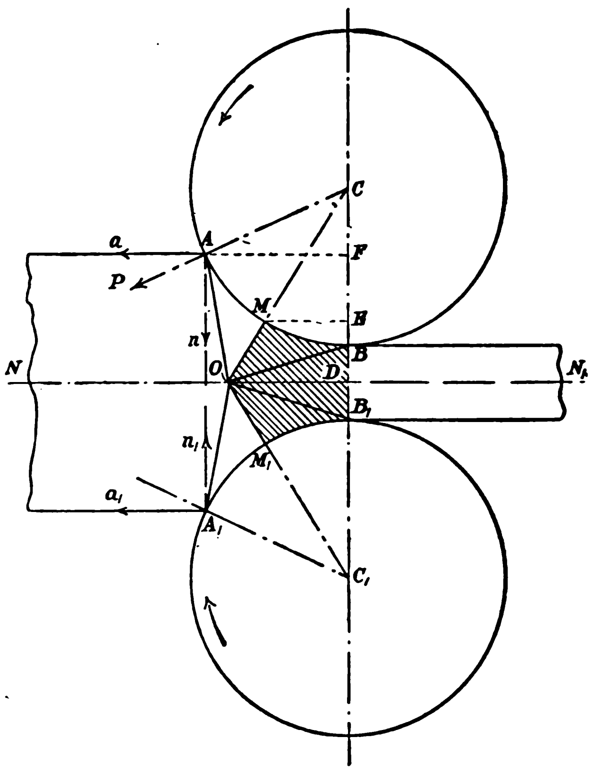


FIG. 2 CROSS-SECTION OF A ROLLING MILL THROUGH THE ROLLS

which hinders the piece from entering the rolls, and unless n is larger than a, the rolls will not grip the piece.

6 The useful limit of ϕ is easily found from the relation

$$n > a$$

or
$$P \cos \phi > P \sin \phi$$

Eliminating P,
$$\cos \phi > \sin \phi$$

from which
$$\phi < 45 \text{ deg.}$$

The actual value of ϕ for metals is given by $\tan \phi = m$, the coefficient of friction between roll and piece, but for metals

$$m = 0.577 \text{ (approx.)}$$

therefore

$$\phi = \tan^{-1} 0.577 = 30 \text{ deg.}$$

which according to actual practice is the maximum value of the approach angle for which the rolls will grip the piece.

7 Let us now deal with the effect of the pressure on the whole arc of contact AB. We may consider the pressure as applied to the middle point M and the whole piece represented in cross-section by ABB_1A_1 will be divided into pressure pyramids with a common vertex at O. In accordance with previous remarks concerning the behavior of a piece under compression, we deduce that the part of the piece bounded by the contact arcs and shown shaded in the figure is pushed outward by the rolls in the direction of the pass, consequently it is the one we must consider in looking for the elongation of the piece. Obviously, when the piece has gone through the rolls, the shaded area $OMBB_1M_1$ will have been transformed into a rectangle, its height being equal to the pass and its base longer than the average base of the original expelled part.

8 To determine, therefore, the elongation of the piece we will proceed as follows:

Area $OMBB_1M_1$ = area of triangle COC_1 – twice the area of sector CMB; but

$$\text{Area of triangle } COC_1 = \frac{CC_1 \times OD}{2} \quad . \; . \; . \; . \; . \; . \; . [1]$$

and

$$\text{Area of sector } CMB = \frac{\text{area of circle } C}{360} \times \text{angle } MCB \quad . \; . \; . [2]$$

In order to symbolize Equations [1] and [2] we will adopt the following notation:

r = radius of the roll

p = pass (BB_1)

t = original thickness of the piece (AA_1)

ϕ = approach angle (ACB)

Then Equation [1] becomes

$$\text{Area of triangle } COC_1 = \frac{(2r + p)^2 \tan \frac{\phi}{2}}{4}$$

but

$$\tan \frac{\phi}{2} = \frac{1 - \cos \phi}{\sin \phi}$$

and
$$\cos\phi = 1 - \frac{(t-p)}{2r}$$

whence
$$\tan\frac{\phi}{2} = \frac{t-p}{2r\sin\phi}$$

and the above equation becomes

$$\text{Area of triangle } COC_1 = \frac{(2r+p)^2\,(t-p)}{8r\sin\phi}$$

9 Symbolizing Equation [2] we obtain

$$\text{Area of sector } CMB = \frac{\pi r^2}{360}\,\phi$$

and the shaded area will be given by

$$OMBB_1M_1 = \frac{(2r+p)^2\,(t-p)}{8r\sin\phi} - \frac{\pi r^2}{360}\,\phi \quad . \; . \; . \; . \; . [3]$$

As we have noted above, this must be equal to the area of a rectangle of height p and base of length l, the value of which latter quantity is given by the following equation:

$$l = \frac{1}{p}\left[\frac{(2r+p)^2(t-p)}{8r\sin\phi} - \frac{\pi r^2}{360}\,\phi\right] \quad . \; . \; . \; . \; . \; . [4]$$

The value of the average base l_0 of the shaded area is

$$l_0 = \frac{OD+ME}{2} = \frac{1}{2}\left[\frac{(2r+p)}{2}\tan\frac{\phi}{2} + r\sin\frac{\phi}{2}\right]$$

but
$$\tan\frac{\phi}{2} = \frac{t-p}{2r\sin\phi}$$

therefore

$$l_0 = \frac{1}{2}\left[\frac{(2r+p)\,(t-p)}{4r\sin\phi} + r\sin\frac{\phi}{2}\right] \quad . \; . \; . \; . \; . \; . [5]$$

10 The elongation Δ of the piece will evidently be $l - l_0$, and referring Δ thus found to the total length of the piece gripped by the rolls, we will have the elongation per unit length Δ_1. The total length of the piece gripped by the rolls is (Fig. 2)

$$AF = r\sin\phi$$

whence
$$\Delta_1 = \frac{\Delta}{r\sin\phi} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . [6]$$

The percentage elongation will be

$$\Delta_{\%} = \frac{100\,\Delta}{r\sin\phi} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . [7]$$

Equation [7], besides showing us the elongation obtained by rolling a piece of metal, furnishes also other valuable information, namely, a check on the safety of the rolling operation, by comparing the

$\Delta_{\%}$ with the tensile-strength test of the material in question. In the event that the value of $\Delta_{\%}$ is too large for safely straining the piece, or should it happen to be too conservative, it will always be possible to readjust the pass of the rolling mill to suit, because Δ is a function of ϕ.

11 In order to find the side spread (σ) of the piece caused by the roll pressure, we will refer to Fig. 3, which is a longitudinal cross-section through the rolls. Let CC and C_1C_1 be the axes of the rolls, $AA_1 = t$ the original thickness of the piece, and $BB_1 = p$ the pass.

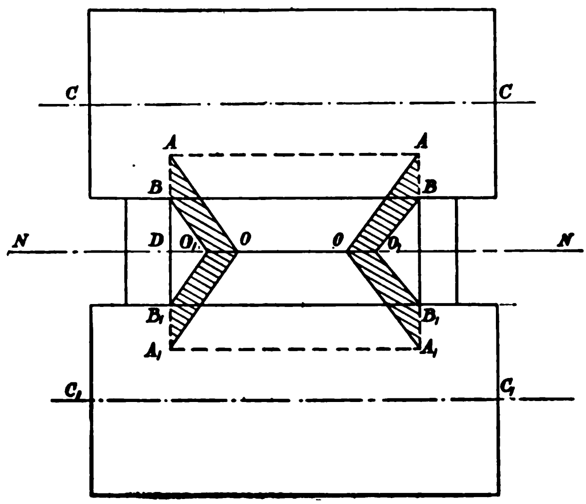


FIG. 3 LONGITUDINAL CROSS-SECTION THROUGH THE ROLLS

In this case, since the expulsion angles AOA_1 are equal for both sides of the piece, the total spread will be twice the spread of one side.

12 By reasoning the same as we did before in the case of Fig. 1, we can easily see that the amount of side spread is given by the shaded areas, namely, AOA_1 and B_1O_1B. In this case the shaded area is given by the difference of the areas of the triangles AOA_1 and BO_1B_1, which depend upon the value of the pressure angle a because angle $DAO = a/2$. The value of a may be found by returning to Fig. 2 and noting that

$$\text{Angle } COB = \text{angle } DBO - \text{angle } OCB$$

but $$\tan DBO = \frac{OD}{DB}$$

and $$OD = (CB + BD) \tan OCD$$

whence $$\tan DBO = \frac{(CB + BD) \tan OCD}{DB}$$

from which $$\text{angle } DBO = \tan^{-1}\left[\frac{(CB + BD) \tan OCD}{DB}\right]$$

and substituting this value in the equation above,

$$\text{Angle } COB = \tan^{-1}\left[\frac{(CB + BD) \tan OCD}{DB}\right] - \text{angle } OCB$$

13 To symbolize, now, this equation, we may use the notation already adopted, when the value of the pressure angle will be given by

$$\frac{a}{2} = \tan^{-1}\left[\frac{\left(r + \frac{p}{2}\right) \tan \frac{\phi}{2}}{\frac{p}{2}}\right] - \frac{\phi}{2} \quad \ldots\ldots\ldots [8]$$

14 It should be noted that the pressure angle is also a function of the approach angle ϕ, and once the value of the pressure angle has been calculated it is easy to find (Fig. 3) the area of the shaded portion, which divided by the pass will give the amount of the side spread σ of the piece; in fact,

$$\text{Area of triangle } AOA_1 = \frac{1}{2}(AA_1 \times DO) = \frac{1}{2}\left[AA_1 \times \frac{AA_1}{2} \tan \frac{a}{2}\right]$$

Similarly,

$$\text{Area of triangle } BOB_1 = \frac{1}{2}\left[BB_1 \times \frac{BB_1}{2} \tan \frac{a}{2}\right]$$

15 Symbolizing and subtracting the second expression from the first, we obtain:

$$\text{Area of shaded portion} = (t^2 - p^2) \tan \frac{a}{2}$$

and the side spread at each side of the piece will be

$$\sigma = \frac{(t^2 - p^2)}{p} \tan \frac{a}{2} \quad \ldots\ldots\ldots [9]$$

If w is the width of the piece, the total spread per unit of width will be

$$2\sigma_1 = \frac{2\sigma}{w}$$

and the percentage spread

$$2\sigma\% = \frac{200\,\sigma}{w} \quad \ldots\ldots\ldots [10]$$

Equation [9] shows that the side spread is independent of the width of the piece, but it depends entirely on the values of the original thickness t and the pass p.

16 In conclusion the author wishes to present a practical application of the foregoing formulæ by solving the following problem: In the cold-rolling operation a strip of 0.20 per cent carbon steel is put through a 10-in. mill. The strip before rolling is 2 in. wide and 0.065 in. thick. The pass is 0.050 in. Find the percentage elongation and side spread. The first step is to find the approach angle. The fundamental equation of a rolling mill is

$$t = p + 2r(1 - \cos \phi)$$

Solving for ϕ,

$$\cos \phi = 1 - \frac{t - p}{2r}$$

and from the data above

$$\cos \phi = 0.9985$$

or

$$\phi = 3 \text{ deg. } 8 \text{ min.}$$

Next we must find the angle of pressure and from the above data and Equation [8] we obtain

$$\frac{a}{2} = \tan^{-1}\left[\frac{\left(r + \frac{p}{2}\right)\tan\frac{\phi}{2}}{\frac{p}{2}}\right] - \frac{\phi}{2}$$

$$= \tan^{-1}\left[\frac{5.025 \times \tan (1 \text{ deg. } 34 \text{ min.})}{0.025}\right] - 1 \text{ deg. } 34 \text{ min.}$$

$$= 78 \text{ deg. } 7 \text{ min.}$$

17 We can now arrange our data in full as follows:

Roll radius, $r = 5$ in.

Original thickness, $t = 0.065$ in.

Pass, $p = 0.050$ in.

Width of strip, $w = 2$ in.

Approach, $\phi = 3$ deg. 8 min. $= 3.133$ deg.

$\sin \phi = 0.05466$

$\cos \phi = 0.9985$

$\frac{\phi}{2} = 1$ deg. 34 min.

$\sin \frac{\phi}{2} = 0.02734$

$\frac{\phi}{360} = 0.0087$

Area of roll cross-section, $\pi r^2 = 78.5$ sq. in.

$$\text{Pressure angle, } \alpha = 156 \text{ deg. } 14 \text{ min.}$$

$$\frac{\alpha}{2} = 78 \text{ deg. } 7 \text{ min.}$$

$$\tan\frac{\alpha}{2} = 4.75219$$

18 To find the elongation, using Equations [4], [5] and [6], we obtain

$$l = \frac{1}{0.050}\left[\frac{(10.05)^2 \times (0.015)}{8 \times 5 \times 0.05466} - 0.0087 \times 78.5\right] = 0.2 \text{ in.}$$

$$l_0 = \frac{1}{2}\left[\left(\frac{10.05 \times 0.015}{4 \times 5 \times 0.05466}\right) + (5 \times 0.02734)\right] = 0.13725 \text{ in.}$$

and

$$\Delta = l - l_0 = 0.06275 \text{ in.}$$

or in percentage,

$$\Delta_{\%} = \frac{6.275}{5 \times 0.05466} = 24.6 \text{ per cent of the original length of the}$$

piece gripped by the rolls.

19 To find the side spread on either side, using Equations [9] and [10] we obtain

$$\sigma = \frac{(0.065)^2 - (0.050)^2}{0.05} \times 4.75219 = 0.164 \text{ in.}$$

The percentage spread is

$$2\,\sigma_{\%} = 16.4 \text{ per cent}$$

20 The result thus obtained evidently shows that $\sigma_{\%}$ is inversely proportional to w, the width of the piece. Now, because of the fact that in rolling-mill practice we are interested only in obtaining the highest possible value of the elongation Δ, the best economic operative conditions will be furthered by using the widest possible strip consistent with the dimensions of the rolls, the power of the mill and above all the degree of uniformity desired in the thickness of the finished strip.

DISCUSSION

E. O. Goss (written). In order to determine the ratio of side spread to elongation in actual cold rolling of brass the following experiment was undertaken with the results indicated.

A piece of annealed brass, $8\frac{1}{2}$ in. wide, was cut into pieces 2, 4 and 8 in. wide by 12 in. long, and was reduced in one pass on a 20-in.

mill. The calculations shown below are based on the actual dimensions made on the individual specimens.

Before Rolling Size, in. Gage, in.	After Rolling Size, in. Gage, in.	Per cent Reduction by Rolling	Per cent Elong.	Per cent Side Spread
1.999 × 12.005 × 0.069	2.043 × 16.5 × 0.048	29.7	37.4	2.20
4.000 × 12.005 × 0.068	4.030 × 16.0 × 0.050	26.4	33.2	0.75
6.000 × 12.001 × 0.068	6.018 × 16.6 × 0.0485	28.6	38.3	0.30
8.000 × 12.000 × 0.067	8.020 × 16.7 × 0.0475	29.1	39.1	0.25

In the case of the 2-in. specimen the metal was rolled at an angle owing to the fact that the guides were too wide. For this reason the side spread was rather high; nevertheless it is much lower than the figures given by the author. The other specimens rolled much straighter.

Although the author considers rolled steel and the above experiments were conducted on brass, the writer does not believe that the wide difference in per cent of side spread can be due entirely to the difference in the material.

No. 1729

MODERN ELECTRIC FURNACE PRACTICE AS RELATED TO FOUNDRIES IN PARTICULAR

By W. E. Moore, Pittsburgh, Pa.
Member of the Society

Since 1913 *the number of electric furnaces employed in the steel industry in the United States and Canada has risen from* 19 *to* 335, *practically all of which are of the arc type.*

After first giving particulars regarding the operation of acid and basic furnaces, the author enumerates the superior properties which have created the present demand for electric steel. He then points out the advantages of the electric furnace over the open-hearth furnaces in steel melting and states the features of the various classes of arc furnaces which have made that type practically supreme in its field.

Following this the crucible and converter processes are briefly described and it is shown how the acid type of furnace is the best for foundry work.

Comparative operating costs of producing liquid steel by the converter and electric-furnace processes are then given, which bring out the marked economy of the latter. The paper concludes with notes on the employment of the electric furnace in malleable-iron foundries and on the selection of the most suitable type of furnace for a given installation.

UP to the present time the electric furnace has seen its largest commercial development, first, in the manufacture of aluminum, second, in the manufacture of steel, third, in the manufacture of ferroalloys, and fourth, in the manufacture of calcium carbide. At the end of 1913 there were only nineteen electric furnaces installed in the steel-making industry in America. This number had increased to 136 at the end of 1916 and to 269 at the end of 1917. At the present time the number of steel-making furnaces in use in various industrial countries of the world is 815, of which 290 are in the United States and 45 in Canada.

2 The average capacity of these furnaces in America is 7.3 tons per heat when used for ingots, and in the foundry business 1.7 tons per heat, though the ordinary size now most generally used in foundries is the 3-ton. More than 99 per cent of all the steel-

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

making furnaces are of the arc type, less than 1 per cent being of the induction type, which was popular in the early days of the art.

3 Electric furnaces are used for the following principal purposes in the metal industries:

a Forging steels, tool steels, alloy steels, etc.
b Making steel castings in foundries
c Making high grades of strong cast iron for difficult or fine castings
d Melting brass, bronze and other non-ferrous metals.

ACID AND BASIC FURNACES

4 Acid furnaces are those lined with silicious refractories and are used for melting and alloying steel made from a high grade of scrap, where the sulphur and phosphorus impurities are so low that much refining is not deemed necessary as in making steel castings or merchant-bar steel.

5 The strength of cast iron is greatly improved by melting or treating in an acid electric furnace, though where it is necessary to reduce the sulphur or phosphorus content materially the basic furnace must be used. It is possible, however, in a properly operated acid furnace to reduce such impurities by only a small percentage, say, 10 to 15 per cent.

6 With basic furnaces the refractory linings are resistant to the lime slags carried as a molten blanket on top of the steel bath for the purpose of taking up and fluxing out the sulphur and phosphorus. Phosphorus is removed in the first operation. High-calcium slags, when in contact with metal under oxidizing conditions as when iron ore or mill scale is added to the bath and at not too high temperature, have a strong affinity for phosphorus. It is quickly absorbed by the slag during that period of the heat known to the melters as the "boil." The phosphorus from the metal is thus oxidized into phosphoric acid and converted into phosphate of lime, in which form it is a stable constituent of the slag, provided the temperature is not carried too high. This first or oxidizing slag is then skimmed off and another slag is made by adding lime and carbon, which is fused by the heat of the arc. This slag then forms calcium carbide, or the "carbide slag." Carbon in the form of granular coke, retort carbon or anthracite coal, is used with high-calcium, low-magnesia lime. This carbide slag under high heat and in a reducing atmosphere has a strong affinity for sulphur and carries it off from the steel absorbing it in the form of calcium

sulphate. It is thus possible, though not always commercially practicable, to make a very high grade of tool steel from very impure scrap in an electric furnace. Heretofore tool steel could be made only by melting the purest forms of Norway or Swedish iron, and then only by melting in crucibles, which is a very slow and expensive process and one in which no refining is possible.

SUPERIOR PROPERTIES OF ELECTRIC STEEL

7 Primarily, electric steel became popular because of its superior physical properties. While such steel can be made with a more satisfactory chemical analysis, using a given grade of raw materials, than by other processes, experience has abundantly demonstrated that when made according to the same chemical analysis it will have about 15 per cent greater tensile strength or ductility,

TABLE 1 COMPARISON OF OPEN-HEARTH AND ELECTRIC-FURNACE STEEL

	Open hearth	Electric furnace
Elastic limit, lb. per sq. in.	41,060	64,850
Tensile strength, lb. per sq. in.	89,100	105,140
Per cent elongation in 2 in.	21.5	22.0
Per cent reduction of area	31.74	52.37
Elastic torsion	16,750	33,700
Character of fracture	Granular	Silky cup

depending upon its heat treatment, and will be more resistant to shock and better able to receive heat treatment. The reason for this is that the steel, being made in a closed furnace and in a reducing atmosphere away from the contaminating influences of combustion gases, is more solid, freer from gases and less prone to non-metallic inclusions of slag, oxides, etc. As an example, the tests in Table 1, made by R. W. Hunt and Company, Chicago, January 30, 1919, for the Chicago Surface Lines, illustrate the physical properties of A.E.R.A. specification heat-treated electric-furnace axle steel. It was heated to from 1450 to 1460 deg. fahr., held one hour, quenched in 65-deg. oil, drawn at from 1185 to 1200 deg. fahr. for one hour and then slowly cooled in the furnace. The open-hearth steel was also heat-treated in the same manner.

8 The Bureau of Standards reports regarding the superior qualities of electric steel as follows:

The characteristics of electric steel are great homogenity and freedom from segregation. It is somewhat higher in tensile strength and elastic limit than other steels and, owing to its greater density, shows a marked resistance to fatigue.

9 Being absolutely "dead" when properly made and averaging lower in sulphur, electric steel in the foundry is less liable to show shrinkage cracks between ribs of castings. Being more fluid, it is not so liable to piping, blowholes, cold shuts or misruns. In tool steel it takes heat treatment more effectively and will stand greater abuse in heating.

10 The electric furnace is especially useful for making alloy steels. Since the metal is treated in a reducing atmosphere, there need be no large losses of the added elements, such as silicon, manganese, vanadium and chromium, which in the ordinary open-hearth practice are oxidized in large quantities and carried to waste in the slag, thus producing uncertain mixtures. Indeed, the added elements in open-hearth practice frequently show losses of from 30 to 50 per cent, while in the electric furnace they will be practically nil.

11 With the electric furnace it is much easier to carry the finishing operation to a more exact limit in carbon and silicon content than is practicable in the case of the open-hearth furnace, and to finish with an absence of the gases of solution and inclusion, such as oxygen and nitrogen.

ELECTRIC VERSUS OPEN-HEARTH FURNACES

12 In open-hearth furnaces it is impracticable to melt down fine scrap such as turnings in quantity without excessive additions of pig iron, for the reason that the oxidizing flames, which furnish the heat to the furnace, will reduce the metal when in an attenuated form to a mass of oxide before it becomes molten, whereas with the electric furnace it is entirely practicable to melt turnings exclusively, which under ordinary market conditions are purchasable at a price from five to ten dollars per ton lower than that for the heavy melting grade of steel required in open-hearth practice.

13 In the electric furnace it is possible to obtain a heat-transfer efficiency of from 60 to 70 per cent of the heat energy of the electric power supply put into the molten charge, whereas with open-hearth practice the efficiency ranges from 8 to 15 per cent and in crucible practice from 2 to 6 per cent. The fuel-developed heat unit in the open-hearth furnace is, however, bought in a much cheaper form than the heat unit supplied by electric power, and if the electric furnace did not have the other advantages mentioned it

could not at present compete against the open-hearth furnace on the basis of cost.

14 On account of the very intense heat of the electric arc, it is entirely feasible to melt down quite rapidly; thus in the electric furnace it is possible to melt down and refine a charge of foundry steel in one hour or less which in the open-hearth furnace might require from 6 to 14 hours. In other words, a 12-ton electric furnace may be practically equivalent to an 80-ton open-hearth furnace, so far as steel output is concerned, and involve far less installation cost.

15 A 12-ton electric furnace of course requires a correspondingly smaller ladle, crane and building structure, for it is the practice to tap the entire heat into one ladle with either type of furnace.

THE ARC TYPE OF ELECTRIC FURNACE

16 The arc type of electric furnace is practically the only one being installed for steel making today; however, there have been a few of the induction-type furnaces constructed. At first glance the induction-type furnace appears to have many advantages over the arc-type furnace, but practice has shown that it is nowise a competitor of a properly constructed arc furnace. In large sizes the power factor is extremely low, the efficiency of the furnace poor and the cost of replacing the refractories very high. It is not as good a refining furnace as the arc type, where the slag is hotter than the steel, which case is reversed in the induction furnace.

17 Arc furnaces may again be classified into long-arc and short-arc types. There are many theoretical inducements for using the long-arc furnaces, for with a given energy input the electrode is correspondingly smaller and the electrode cost therefore reduced proportionately. With the water-cooled bottom-contact type of furnace either the furnace size must be kept small or the voltage must be greatly increased in order to keep the current low and prevent the bottom from overheating, so it is the custom to operate the bottom-electrode furnace with quite a long arc and proportionally low current, and in small sizes only.

18 Arc furnaces may again be classified into single-phase, two-phase, and three-phase types. The single-phase furnace is ideally simple but is poorly adapted to modern power-plant conditions, as central-station power today is universally generated and transmitted in three phases. The long-arc single-phase furnaces, too, have the very great disadvantage of operating on extremely low power factor, thus causing a great waste in transformer, line

and generator capacity, which usually makes them prohibitive from the central-station man's viewpoint in any but quite small sizes, say, ½ ton to 1 ton capacity per heat.

19 The two-phase furnace may be operated either from two- or three-phase power, but when operated from the usual three-phase power the phases have to be transformed by Scott-connected transformers, which are more costly, less efficient and frequently unbalance the power system. The two-phase furnace is usually built with four arcing electrodes and, while it gives a theoretically balanced load on the power system, it has the objection of requiring an additional electrode, which increases the electrode consumption 33 per cent over the three-phase furnace.

20 The three-phase furnace for installations of moderate and large size is the most universally satisfactory and popular furnace, fulfilling all the conditions as to balanced load and high power factor required by the central stations, and at the same time giving the minimum electrode loss and the simplest form of automatic electrode adjusting gear.

21 It is possible to obtain satisfactory operation of the direct-arc-type furnace for melting non-ferrous metals only where the content of metals which volatilize at low temperatures is small, as, for instance, in making bronze and low-zinc metals. Where the zinc or aluminum content is high, as in yellow brass, Muntz metal, etc., this type of furnace is highly unsatisfactory and results in great waste of the more volatile metals and the making of porous castings. Consequently special furnaces of the resistor, rocking, rolling, tumbling induction or distributed-arc types are required, the latter referring to furnaces where heat of arc is transmitted by radiation alone.

DISADVANTAGES OF CRUCIBLE FURNACES

22 The crucible furnace is the oldest method for making steel castings and first-class tool steels. It is now, however, being practically displaced by the electric furnace, which has many advantages such as rapidity and reduced cost, as well as the ability to make sounder castings and better tool and alloy steels. The cost of crucibles and fuel alone in crucible-steel foundries frequently runs up to $60 per ton. With crucible steel the heats are of such small quantity, 100 to 200 lb. per pot, that the contents of several pots must be combined into one ladle to make a casting of ordinary size.

Sometimes the different pots of steel are of such different compositions as to cause the ladle to boil when they are combined, tending to make unsound castings.

23 Due to the absorption of carbon from the crucible, it is difficult to make castings low enough in carbon to obtain the ductility desired for many purposes. Furthermore, the steel reduces the silica from the clay of the crucible, tending to run the silicon content of the product high. The overpowering objection to the crucible process, however, is the high cost of the products, due to:

- *a* High cost of pure melting stock, as no refining is practicable
- *b* Very high labor cost on account of the small heats handled
- *c* Crucible furnaces are extravagant in fuel consumption, sometimes using 3 tons of coal per ton of steel melted
- *d* High cost of crucible renewals, often averaging two to four crucibles per ton melted at a cost of \$9 to \$11 each, or \$18 to \$44 per ton for crucibles alone.

For these reasons the crucible-melting shop is rapidly going out of use for castings, as well as for tool steels.

THE CONVERTER PROCESS IN STEEL FOUNDING

24 During later years the side-blow converter process has become very popular in steel foundries making castings of medium and small size. This process requires high-grade, high-silicon, low-phosphorus and low-sulphur pig iron to be premelted in a cupola furnace with the finest grade of coke obtainable. The quite hot liquid iron is then tapped into a ladle, transported to and dumped into the converter. The converter is then tilted until the blast tuyeres, which enter at the side of the vessel, are turned down to blow directly on to the surface of the metal. The blast, which is generally from a Root-type blower, is then turned on, impinging sharply against the surface of the metal, which is thus violently agitated and oxidized.

25 The air blast burns out the silicon and then the carbon, the products of combustion being CO_2 and SiO_2. This combustion greatly increases the heat of the metal as brought from the cupola. When the process has gone far enough, which the operators guess at from the "drop of the flame," the ferroalloys are thrown into the bath to reduce the oxides, neutralize the sulphur and "kill" the steel. It would otherwise be "wild," i.e., full of effervescing gases,

when poured. It would also be "hot-short," that is, prone to crack when freezing in the molds, unless "doctored" liberally by the addition of manganese, as the converter effectively burns out the manganese of the pig-iron charge. Notwithstanding the doctoring, the steel is permeated by oxides, nitrides and non-metallic inclusions.

26 The steel is dumped from the converter into a bull ladle into which aluminum amounting to from 1 to 3 lb. per ton has been charged. It is transported by traveling crane directly to the larger molds, or to be shanked off into the smaller molds by small hand-shank ladles.

27 The advantages of the converter process are: The steel may be made quite hot and fluid enough for reasonably thin castings; the heats, usually running one to two tons, are of convenient size to be poured off quickly before cooling; the fuel consumption is moderate, though rather higher than for the cast-iron foundry cupola, averaging from 400 to 600 lb. of coke per ton; the first cost of the apparatus is low and the process is available for intermittent service.

28 The disadvantages are: The metal must be handled twice in the ladle; the metal picks up sulphur and phosphorus and nitrides from contact with the fuel and the air blast; the losses in the cupola and converter are quite high, running from 16 to 24 per cent, which further concentrate and increase the percentages of impurities in the original metal and waste costly melting stock; the steel is full of oxides and gases and requires large quantities of expensive ferroalloys to kill; the quality of the steel, physically as well as chemically, is below par; a heat once blown too cold cannot again be brought up in heat enough to cast and must be "pigged;" only the highest grades of melting stock may be used, costing generally from $15 to $25 per ton more than for the acid open-hearth furnace and from $20 to $35 per ton more than for the electric furnace, and the refractory maintenance is high, as the cupola and converter linings must be repaired after each 10-hour run. Liquid-metal costs of converter steel frequently run up to from $60 or $80 per ton.

29 The electric furnace is the most modern steel-producing agency and is gaining in popularity more rapidly than all others. It is the most compact furnace and the rapidity with which it will melt down cold charges adapts it splendidly to the making of steel castings as well as forging and tool steels. It is the cleanest and

most certain method of making steel, and its small bulk makes it feasible to locate the furnace near the center of the floor where the metal need be transported short distances only. The size is small and convenient and the heats come rapidly, so that no large floor area need be tied up in molds per heat.

ACID-TYPE ELECTRIC FURNACE BEST FOR FOUNDRY WORK

30 The acid-type furnace is best suited for foundry work and the most popular size has a capacity of 3 tons per heat, though sometimes 1.5-ton or smaller furnaces are required. The more highly powered and rapid furnaces for such work turn out from 8 to 16 heats in 24 hours, and with a power consumption of from 500 to 650 kw-hr. per ton of liquid steel. Considering the ultimate efficiency of the large modern turbo-generator power house at, say, 1.5 lb. of coal per kw-hr., its fuel consumption might be said to be the equivalent to from 750 to 900 lb. of coal per ton melted, and the coal need not be of high grade nor low in sulphur and phosphorus as is necessary with fuel-fired furnaces. Basic furnaces require more time and power, heats ranging from four to eight per day, and since the charge is melted in a reducing atmosphere there is practically no oxidation of the metal; consequently thin scrap, light turnings or scrap of other forms such as can be conveniently charged into the furnace, may be melted. Such scrap on the present market sells for approximately from $5 to $10 per ton less than low-phosphorus, heavy melting scrap necessary with the ordinary acid open-hearth melting furnace installation.

31 The reducing atmosphere makes it easier to refine and kill the steel, resulting in a saving amounting frequently to half of the ferroalloys necessary with converter steel, and effecting a saving of, say, $2 per ton. The melting losses in the electric furnace are much the lowest of any modern process, averaging from 2 to 5 per cent as against 6 to 9 per cent in the open-hearth and 16 to 24 per cent in the converter process.

32 The electric furnace does not contaminate the metal as do fuel-heated furnaces and an acid electric will therefore readily make No. 3-U.S.A. specification steel, whereas it is practically impossible to find melting stock sufficiently pure to do so with the converter process. The saving alone in the cost of melting stock will more than pay for the entire conversion cost of electric steel. The deader, more dense electric steel yields a larger percentage of good castings and the lower sulphur renders them

free from shrinkage, flaws and cracks, while the hotter and more fluid steel renders possible thinner and lighter-weight sections than can be produced commercially by other processes. The greatest points in favor of the electric furnace are the higher grades of steel produced and the higher percentage yield in castings and bars.

COMPARATIVE COSTS OF ELECTRIC AND CONVERTER STEEL

33 With the electric furnace, men can more readily make and check their steel to an exact percentage of carbon, manganese

TABLE 2 AVERAGE COST PER TON FOR TWO TONS OF CONVERTER STEEL DIVIDED INTO FOUR CUPOLA CHARGES

Two-ton Converter Charge	Cost per Ton of Liquid Steel
Low-phosphorus pig iron	$14.00
Bessemer pig iron	7.80
Steel scrap	10.14
Silicon and spiegel	5.55
Coke, 863 lb.	1.01
Cost of material per ton of liquid steel	$38.68
Additions per ton of steel:	
10 lb. 80 per cent ferromanganese at 6 cents	0.60
6 lb. 50 per cent ferrosilicon at 5 cents	0.30
2 lb. aluminum at 30 cents	0.60
Power for blower motors	1.25
Cost of materials and power per ton of liquid steel	$41.43
Average cost of cupola and converter linings	1.20
Labor costs	3.00
Cost of converter steel per net ton in ladle	$45.63

and silicon, and can more easily keep the undesirable sulphur and phosphorus to low limits than by any other process. The steel may be readily alloyed with nickel, chromium and vanadium to make the higher grades of steel castings to replace forgings and for special purposes, such as may be required for parts of unusual strength, ductility or for cutting tools. It is entirely feasible to make castings which will run up to an ultimate strength of 130,000 lb. per sq. in., or to cast high-speed-steel milling cutters and reamers to form for grinding. The figures in Tables 2 and 3 show present-day comparative operating costs for liquid steel in the ladle under favorable conditions—as in a steel foundry under 24-hr.-per-day operation.

34 The acid open-hearth furnace is still frequently used, generally with oil fuel and mostly in foundries making the heavier classes of steel castings. With the acid open-hearth furnace the standard price must be paid for 50 per cent of the charge in low-phosphorus heavy melting scrap and 50 per cent of the charge in bessemer pig iron. The fuel-oil consumption for such open-hearth furnaces in foundries usually runs from 45 to 90 gal., costing at the present time from $3.37 to $7.50 per ton of liquid steel. In the largest of the steel foundries, it is true, producer gas is frequently used at less fuel cost. However, the contaminating effect of the sulphur content of the coal and the complications and expense of

TABLE 3 AVERAGE COST PER TON FOR THREE-TON ACID-LINED, RAPID-TYPE, POLYPHASE, ELECTRIC FOUNDRY FURNACE STEEL

Three-ton Electric-Furnace Charge	Cost per Ton of Liquid Steel
Axle turnings	$12.40
(included above 3 per cent losses, 200 lb.)	
Mill scale	0.09
(included above 60 per cent losses, 60 lb.)	
Electrodes at 7 cents	1.40
1650 kw-hr. (550 per ton) at 1 cent per kw-hr	5.50
Losses in melting 260 lb.:	
80 per cent ferromanganese	0.40
50 per cent ferrosilicon	0.25
Aluminum at 30 cents	0.15
Cost of material per ton of liquid steel	$20.19
Average cost of linings and roofs	0.40
Labor cost on furnace attendance	1.00
Cost of electric steel per net ton in ladle	$21.59

the producer plant as a rule deter the ordinary steel foundry from using coal producer gas as open-hearth fuel.

35 The great drawback to the open-hearth furnace is the high cost of melting stock and its well-known inability to furnish steel sufficiently hot to make medium and small castings satisfactorily without undue costs for refractories and largely increased fuel consumption. The inconvenience of large heats from the open-hearth furnace, from 15 to 40 tons, counts heavily against it for small casting work.

36 The electric furnace is coming into use for the finer grades of cast iron and for melting cheap cast-iron scrap, such as borings, which cannot be successfully melted in the cupola. The electric

furnace greatly improves the strength of cast iron, even in acid melting.

37 G. K. Elliott reports a cupola iron showing a transverse load of 2950 lb. with a 0.10-in. deflection in a standard arbitration test bar. After 25 min. treatment in the electric furnace, using 104 kw-hr. per ton, a similar bar was cast and broke at 4400 lb. with a 0.115-in. deflection.

THE ELECTRIC FURNACE IN MALLEABLE-IRON FOUNDING

38 Very fine results have been obtained with malleable iron made by treating cupola metal or by melting cold scrap in an electric furnace.

39 The ability to refine for sulphur and phosphorus and to add ferroalloys to adjust the mixture to the proper malleablizing formula, together with its very rapid operation, give the electric furnace a decided advantage in malleable-iron foundries, particularly in working on high-phosphorus southern irons. Where an ample supply of cheap, light scrap, such as cast-iron borings, is obtainable, along with suitable electric power, the cost of electric-furnace cast iron in some cases is less than that of the cupola-melted iron. The electric-furnace iron may always be brought to the proper formula for difficult casting conditions, for instance, in casting gas-engine piston rings to size ready for grinding in a cast-iron chill mold.

40 In the non-ferrous-metal industry the electric furnace has shown remarkable economies, due to the saving of cost of crucibles and the greatly reduced metal losses caused by oxidation and volatilization. For such work special types of furnaces designed to avoid localized heating are necessary.

SELECTION OF AN ELECTRIC FURNACE FOR A GIVEN INSTALLATION

41 As to the most suitable type of electric furnace for a given installation, if the scrap be inferior and high in sulphur and phosphorus, then the extra cost, slower operation and shorter refractory life of the basic furnace must be endured to obtain the lower limits of sulphur and phosphorus not practicable to reach with the acid furnace using poorer grades of scrap. At present the call is for acid-lined foundry furnaces, and cheap, good scrap is available in large quantities. The acid furnace is simpler, cheaper and faster to

operate and the steel casts more easily. The basic furnace, however, is essential for tool steels.

42 It is nevertheless strongly recommended in any case that a furnace purchased be so designed and constructed that it is adaptable to basic operation. This means that the furnace shell must be of large diameter and the bath of large area and shallow. The furnace should not, it is thought, be of the long-arc type, nor of the small-diameter deep-bath type if the best and most rapid work is contemplated. Indeed, even for acid melting there is a noticeable difference in the quality of the steel obtained from the large-diameter, shallow-bath furnaces compared with that made in the deep-bath-type furnace, for with the latter it is not feasible to obtain the same mechanical reactions from the additions put in to refine and kill the steel as when the bath is of the shallower type. Nor is it possible so thoroughly to deoxidize the metal in the reducing atmosphere in the furnace.

43 For rapid work it is especially important to have the furnace constructed with all possible operating conveniences and facilities, so that one heat may follow another with the utmost rapidity and with a minimum loss of time for the necessary furnace adjustments. It is therefore important to look carefully to the facilities for making bottom and fettling the banks.

44 It is quite important that the furnace should operate at the highest practicable power factor that can be obtained without undue disturbance of the power company's load, for by so doing the electrode, transformer, line and generator losses are maintained at a minimum. Engineering skill of a high order is required to forecast and select the best type of equipment, under the many varied power-supply conditions which obtain in different localities.

45 By reason of the now generally acknowledged superior quality of the product, greater flexibility of operation, quicker, more-convenient-sized heats, and saving in alloys and in cost of melting stock, the electric furnace is rapidly coming to the front in the steel foundry as well as in alloy- and tool-steel works wherever suitable power is available and progressive policies are in vogue. It is making possible the profitable operation of widely distributed small steel foundries to an extent not generally realized and greatly reducing the investment cost required in tool-steel works.

DISCUSSION

JOHN C. GRAF (written). In Par. 14, the author compares a 12-ton electric furnace with an 80-ton open-hearth furnace. This would be a true comparison where small castings are made, as a 12-ton furnace would make about a 9-ton casting after deducting risers and gates. As there are many calls for castings weighing above 9 tons, there would be necessary, in the case of a 45-ton casting, for instance, at least five 12-ton electric furnaces.

On a basis of cost, the writer has found that foundries equipped with electric furnaces cannot compete with foundries equipped with the converter furnace except, perhaps, for some classes of work. The same applies to open-hearth furnaces which are even more economical than converters. The writer knows of one foundry equipped with a converter in which repeated tests of steel have shown tensile strength ranging from 90,000 to 100,000 lb. and averaging 93,000 lb.

H. L. HESS (written). Early in 1913 the Hess Steel Corporation became interested in electric furnaces as a side issue in connection with a crucible melting plant. One small German furnace was installed, one or two of a design of the company's and finally a 1-ton Stassano furnace. Very little commercial steel was turned out but sufficient data were gathered to prove definitely and conclusively the superiority of electric steel. As a result of this experience, the company now has started the installation of a standardized type of furnace and has a combined electric furnace melting capacity of 40 to 46 tons in 6- and 7-ton units.

It appears that there are limitations to the economical size of furnaces used for the melting of cold stock. The author touches on this phase of the subject in his reference to the diameter of the furnace shell, which, of course, determines the area of the hearth, or bath, and this in turn influences the depth of the bath. His preference is for large diameter, shallow furnaces. In a furnace with bottom-heating facilities, this may be economically the best. For a furnace with no bottom-heating facilities a shallow hearth has certain advantages as there is a tendency for the molten metal to lie sluggishly on the bottom of the deep hearth and the heavier alloys find the lower level.

The most satisfactory refining furnace seems to be the direct-arc type where the metal is heated not only by radiation but also from direct transmission of the arc itself through the metal and its surface layer of slag.

To obtain the maximum heat efficiency, the heat input into the bath should be as uniform throughout the area of this bath as possible.

Furnaces having bottom-heating facilities have definite advantages electrically but to date rather serious disadvantages mechanically, although it is probable that these points may be worked out and that a satisfactory method of applying heat through the bottom and satisfactory lining materials will be developed.

The spreading effect or radiation of heat from electrodes above the surface is limited. Very large electrodes are difficult to obtain, expensive and unwieldy, and a multiplication of electrodes, even if arranged in electrically balanced groups, means additional mechanical equipment, increased upkeep, and increased delays in operation. It is therefore necessary to compromise and use as deep a bath as is economically possible and yet obtain thorough melting to the deepest point just as rapidly as the charge melts around the edges of the hearth.

Every furnace operator has noticed the marked effect upon melting time of a bottom built up too high as a result of constant patching, or of one allowed to burn out to too great an extent.

A furnace that is too deep can be improved by building up, but one with too great a diameter can be changed only at great expense.

The table given by the author in which open-hearth and electric steels are compared is most interesting. Numerous tests of similar character check this information.

The most marked advantage in favor of electric steel, however, is noticeable in dynamic test results, this advantage probably being due, as the author has pointed out, to the close-grained, beautiful structure of correctly melted, true electric steel.

It would not pay to use the same practice or raw materials in making even the best alloy steels for automobile gears and certain machine parts that are necessary in the manufacture of tool steel. In making the former, the practice followed in general is the best basic open-hearth practice. In the manufacture of tool steels, however, very different methods become necessary. Here the expense of material and time are of secondary importance. Correctly made electric tool steel successfully challenges the best grades of crucible steel.

The costs of manufacture of electric steel do not in any sense make it competitive with open-hearth or Bessemer steel as far as price goes. The field of application of the electric steel is for parts requiring maximum strength combined with uniformity and light weight, and the ability to take on the maximum physical properties as a result of heat treatment.

The costs of making electric steel vary greatly, depending upon the type of steel being manufactured. A simple melting down of miscellaneous scrap into molten metal that is to be poured into a casting gives the cheapest result. At the other end of the range may be considered the manufacture of the highest grades of tool steel, where the raw materials must be the best obtainable and the greatest care must be taken throughout the entire operation of melting and pouring.

An ordinary low chrome nickel steel for ingots to be rolled into bars or billets may be considered as a possible mean as far as costs of production and quality of steel are concerned. A typical two-slag low chrome nickel ingot heat, based on 24-hour operation and melted in a 6- or 7-ton furnace might run approximately as follows:

Metal base, metallic addition, flux, re-carburizing material and raw material handling	$33.50
Charging, melting, molds, ladle, crane, ingot-handling and general labor	7.00
Power, fuel for melting, heating ladles, etc.	16.50
General operating, laboratory, water, crane, delay, etc., expense	6.25
Repairs and renewals	4.00
Depreciation	2.50
Total	$69.75

No overhead is figured in the above.

The approximate time of melting and refining such a heat may be from 6.5 to 8 hours, depending upon the type of scrap used and the delays incurred in operation. The power consumed might vary from 750 to 950 kw-hr. per ton, depending upon the same factors.

Before the war, when electrodes were of uniformly high quality, the cost of electrodes ran slightly below $1.00 per ton. At the present time the average electrode consumption is easily double this, and has run, exclusive of breakage, as high as $4.50 per ton of metal produced.

These figures, which might seem abnormally high to the manufacturers of certain grades of steel, are for the production of high-grade material where every precaution is used to obtain maximum quality and where care is used in the selection of raw materials, in thorough refining and in the elimination of foreign inclusions such as slag and dirt.

THE AUTHOR Mr. Graf, when establishing the weight of casting which can be produced from a 12-ton electric furnace as 9 tons is in error, in that the standard modern type electric furnaces of 12 tons rating are capable of making at least 20 tons of steel, so that the weight of the casting would be correspondingly enlarged.

The writer does not know what cost data Mr. Graf has in mind when he states that the electric furnace cannot compete with the converter furnace. The writer finds that under usual and ordinary conditions of power supply at the present time, the electric furnace will turn out foundry steel figured on cost "at the spout" at from one-half to two-thirds the cost of converter steel. These figures are not the result of one instance, but of data accumulated in a number of cases. The cost of electric steel compared to the cost of ordinary oil-fired, acid, open-hearth steel usually runs from two-thirds to three-fourths as much. These figures represent average results, not extraordinary or unusual results in any way.

The excellent discussion by Mr. Hess is very much to the point and very greatly appreciated. In general, the author is in quite hearty agreement with Mr. Hess.

At present no technical limitation to the size of a scientifically designed rapid arc-type electric furnace has been reached. For melting cold stock, though, it must be admitted that some of the best manufacturers have through erroneous design run into size limitations, more properly power limitations, on the larger furnaces when melting cold scrap, especially when used on 60-cycle power. It is, however, quite certain that at least 50-ton electric furnaces can be made quite successfully for melting cold scrap, if the commercial conditions demand that quantity of steel.

Bottom heating, for some alloy steels, becomes quite necessary, but in all cases only a limited amount of power must be put into a furnace through the bottom. Water-cooled electrodes must be used, if the furnace be more than miniature in size, where any current approximating one-fifth or one-third of the total power is put through the bottom, otherwise such bottom heating almost invariably results in serious trouble in maintaining the bottom and in numerous "cut-throughs." With moderate amounts of power up to 20 per cent of the furnace capacity entering through the bottom, the durability of the bottom is practically equivalent to that of a plain bottom furnace.

The furnace shell of circular section seems the most simple and most practical form and it is essential to have the width of bath sufficiently great to prevent the intense radiation from the electrodes cutting the banks where the banks too nearly approach the electrodes.

Mr. Hess's cost analysis for making automobile steel is valuable. It is typical of the old-fashioned slow-type electric furnace. Such furnaces are necessarily very much more expensive in operation, consuming larger amounts of power and electrodes and producing no better results than when operated at a more rapid rate. Records are now at hand for 3-ton modern rapid-type furnaces capable of turning out 40 tons of very hot foundry steel in 12 hours with a power consumption down to or below 500 kw-hr. per ton and this with an electrode consumption of less than 18 lb. of carbon electrodes per ton.

The criticism made with regard to trouble from electrode clamps applies only to furnaces with improperly designed clamps. With improved multi-part water-cooled clamps there is practically no trouble at all.

Answering Mr. Blanchard, the writer would say that the electric furnace for non-ferrous metals, such as brass and bronze is of an entirely different construction than that used for steel, fine cast iron and malleable iron. For such metals having a volatile element such as zinc, the distributed arc type of furnace is essential to keep the volatilization down to the lowest limits. Such furnaces will readily melt a charge of brass or bronze in 40 minutes to one hour, depending upon the size of the furnace and the character of metal charged. The power consumptions range from 250 to 350 kw-hr. per ton, being smaller for high zinc brasses and large furnaces, and higher for smaller furnaces and metals of greater tin content. Such furnaces are made in standard sizes, $\frac{1}{8}$-, $\frac{1}{4}$-, $\frac{1}{2}$-, and 1-ton capacity per heat and have generally heretofore been made only for single-phase operation, though recently a polyphase distributed arc furnace has been commercialized.

As a rule power companies favor users of electric furnaces with special rates and very frequently eliminate the "readiness-to-serve charge" or partially eliminate it, where the furnace user agrees to keep off or to shut down during "peak load" hours. Replying to a question by the Chairman, the author said that the reducing atmosphere referred to in the paper is a CO atmosphere.

No. 1730

THREAD FORMS FOR WORMS AND HOBS

By B. F. Waterman,[1] Providence, R. I.
Non-Member

The use of worm gearing is steadily increasing, and accompanying this increase, perhaps the cause of it, is a corresponding increase in efficiency and durability which is the result of a better understanding of both the theoretical and mechanical problems. This paper points out some of the problems involved and some of the mechanical difficulties in the use of worm gearing, for failure to take these problems and difficulties into consideration in the past has often led to the discrediting of this form of drive. It also suggests methods for bringing about a uniform and satisfactory practice in worm gearing.

IT is only recently that engineers have considered a worm gear in any other light than that of a necessary evil, and this is probably due to the fact that heretofore the finer points of manufacture were not appreciated and were not obtained except in those places where this type of gear had received more than the usual amount of study. It has been known for a long time that the efficiency of worm gearing depends largely on the angle of thread or helix angle of the worm, and literature on the subject emphasizes this point. The actual manufacture of the worm and gear, however, presents certain mechanical difficulties and inaccuracies which are not apparent in any theoretical discussion of the subject. These difficulties appear in making the worms and hobs, especially with multiple threads; in fact, it might be said they appear in making worms or hobs whose helix angle is greater than 18 deg., and, although no attempt has been made to show these difficulties mathematically, enough models have been made to clearly indicate them. It might also be said that these uncertainties are due to the differences in the thread forms produced by the different methods of cutting the worms: first, with an axial tool, the use of which is limited to a rather low helix angle; second, with a normal tool which has no

[1] Brown and Sharpe Manufacturing Company.

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

limit for angle; and third, with a rotary cutter. Also the difference in helix angle at the top and bottom of the thread affects the method of cutting and the form of thread.

2 The most common worm has a single thread. This is usually made with the sides of the threads on the axis forming an included angle of 29 deg. and it can be cut with a lathe tool of 29

TABLE 1 DATA FOR WORMS

No. of Worm	No. of Threads	Outside Diam.	Angle of Thread with Axis, deg., min.	$90^\circ - \alpha = \partial$, deg., min.	Cutting Tool Used	Included Angle of Tool Used, deg.	Tool Cut on Axis or Normal	Diam. Cutter	Addendum S	$D+f$	Normal Thickness t	Lead	Bottom Diam.
1	Single	3.50	83–39	6–21	Lathe tool	29 on axis	Axis	Same tool as No. 2	.3183	.6866	.497	1.00	2.127
2	Double	3.50	77–28	12–32	"	"	"	Same tool as No. 1	.3183	.6866	.488	2.00	2.127
3	Triple	3.467	71–33	18–27	"	"	"	..	.302	.651	.475	3.00	2.165
4	Five	3.420	60–56	29–4	"	"	"	..	.278	.600	.437	5.00	2.220
5	Five	3.50	60–56	29–4	"	"	"	..	.3183	.6866	.437	5.00	2.127
6	..	..	..	..	..	..	..	..	..	..	..	..	..
7	Single	3.50	83–39	6–21	Cutter	29	Normal	3½	.3183	.6866	.497	1.00	2.127
8	Double	3.50	77–28	12–32	"	"	"	"	.3183	.6866	.488	2.00	2.127
9	Triple	3.467	71–33	18–27	"	"	"	"	.302	.651	.475	3.00	2.165
10	Five	3.420	60–56	29–4	"	"	"	"	.278	.600	.437	5.00	2.220
11	Five	3.420	60–56	29–4	"	50	"	"	.278	.600	.437	5.00	2.220
12	Single	3.500	83–39	6–21	"	29	"	6	.3183	.687	.497	1.00	2.127
13	Five	3.420	60–56	29–4	"	50	"	6	.278	.600	.437	5.00	2.220
14	Five	3.420	60–56	29–4	Hob	29	"	..	.278	.600	.437	5.00	2.220

All dimensions in inches. Worms are in every case 2.8634 in. pitch diameter and 1 inch in axial pitch

α = Angle of thread with axis — **S = Addendum = top of thread to pitch line**
∂ = Helix angle — **t = Thickness of thread at pitch line**
$D+f$ = Whole depth of thread

deg., the cutting edge of which is set parallel either to the axis or to the normal section. It can also be cut with a rotary cutter of the same included angle in a thread-milling machine, and the results in shape of thread will not differ enough to affect the working of it in any way. In other words, the hob, which must be backed off with a tool set either to the normal or on the axis, can be made either way and the wheel produced will mesh properly with the worm made in either of the three ways just mentioned. Any general statement, however, has its exceptions. For instance, in making worms for index wheels the very slight error or difference in shape produced by the three methods just mentioned may be too great to be allowable; also, the size of the thread or pitch will affect the

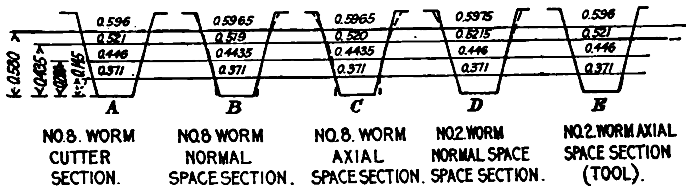


FIG. 1 DOUBLE-THREADED WORM

degree of error, the error varying with the pitch; and the effect produced by any number of threads will depend on the thread angle. The preceding statements as to a single-threaded worm apply to a double-threaded worm, except that in the case of the latter the

TABLE 2 TOOTH PARTS FOR DIFFERENT WORMS

(All dimensions in inches.)

No. of Threads	Applies to Worms Nos.	Thread Angle, 90−α = β, deg., min.	Top Angle, deg., min.	Bottom Angle, deg., min.	Addendum, Axial	Addendum, Normal	t Axial	t Normal	$D+f$ Axial	$D+f$ Normal	Lead
Single	1-7-12	6-21	5-12	8-31	.3183	..	.500	.497	.6866	..	1.00
Double	2-8	12-32	10-18	16-40	.3183	..	.500	.488	.6866	..	2.00
Triple	3-9	18-27	15-24	23-48	..	.302	.500	.475	..	.651	3.00
Five	4-10-11-13-14	29-4	24-57	35-38	..	.278	..	.437	..	.600	5.00
Five	5	29-4	24-27	36-48	.3183	..	.500	.437	.6866	..	5.00

difference between the normal and axial sections begins to be noticeable.

3 To demonstrate the difference mentioned above, 13 worms were made to the dimensions given in Table 1, which with Table 2 gives all the necessary working information for producing them. It should be noted that the pitch diameter and axial pitch are the same for all.

4 In the case of worm No. 8, Table 1, which is a double-threaded worm and was cut with a straight-sided milling cutter, it was noticed

that when the thread is held to the light, with the cutter or fly tool set in the space, the thread is perceptibly convex. This difference is shown in Fig. 1, where the dimensions of the cutter which cut worm No. 8 are shown at *A*, with the dimensions of tools fitted to the normal and axial sections at *B* and *C*. In all the references to the normal or axial sections it is to be understood that the space section is meant. This method of analysis permits a visual inspection of any difference in shape of thread, as the model

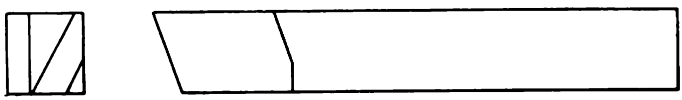


FIG. 2 AXIAL LATHE TOOL FOR WORM NO. 3, TABLE 1, SHOWING UNSUITABLE CUTTING EDGES

or dummy worm can be held to the light and the difference readily seen. The figures show in thousandths of an inch what the difference is, the dotted lines showing divergence of the sides of the teeth from the straight in an exaggerated way.

5 When a triple-threaded worm is considered, the method of cutting becomes of greater importance, since the difference in form between the axial and normal sections (which results from the different methods of cutting) is great enough to require that the hob-teeth shape be made exactly like that of the worm. This ap-

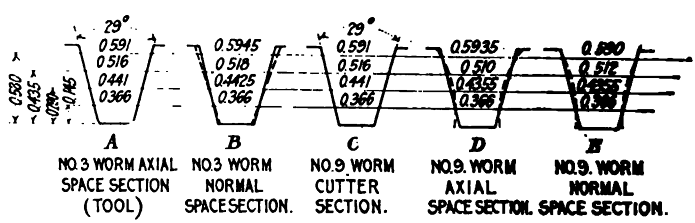


FIG. 3 TRIPLE-THREADED WORM

pears to be a superfluous statement, but it is the exception for a purchaser of a hob to state how he intends to cut the worm, no matter how many threads it has. Worms No. 3 and No. 9 are triple-threaded, No. 3 being cut with an axial tool in a lathe and No. 9 with a rotary cutter. The unfavorable cutting edges on this lathe tool will be noted in Fig. 2. There is a very weak cutting edge with an acute side clearance on one side, while the other side has a very unsatisfactory cutting edge and clearance. Here a tool made on the normal would be better. The differences in the normal and axial sections of worms No. 3 and No. 9 are shown in Fig. 3. The difficulty just mentioned as to cutting clearance on the axial lathe tool may also be experienced with a hob that has axial grooves as shown in Fig. 4, so that when the angle becomes 18 deg. or greater it is well to use spiral grooves, as the hob, unless it has ample radial clearance, may drag on one side. The hob shown is correct in size for worm No. 3.

6 A more extreme case, but now a very common one, is a five-threaded worm where the helix angle for the diameter of the model here shown is approximately 30 deg. Here it will be noticed that an axial tool (Fig. 5) is almost impossible, although worms No. 4 and No. 5 were cut with it. The angle of the thread at the

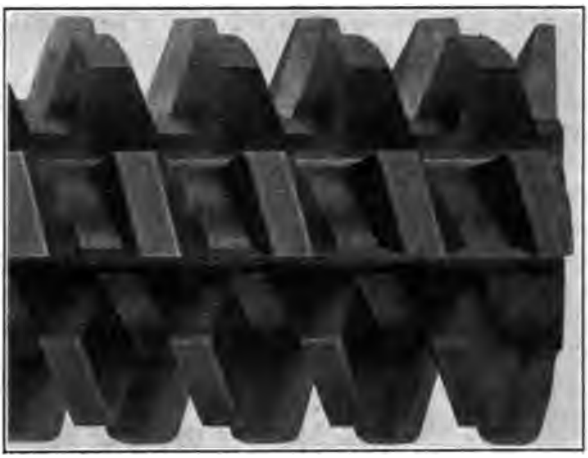


Fig. 4 Cutter for Cutting Worm Wheel to be used with Worm No. 3, Table 1

root governs the clearance angle on the tool and in the case of worm No. 4 the bottom helix angle is 35 deg. 39 min., and in worm No. 5 the helix angle at the bottom is 36 deg. 48 min. The depth of worm No. 4 is based on the normal pitch, while the depth of No. 5 is based on the axial pitch. The shallower tooth, of course, eliminates some of the difficulties experienced with these extreme helix angles and it is good practice to use the normal depth as soon as an 18-deg. angle is reached, or when spiral grooves are used in the hob. In fact, with a helix angle of 45 deg., which is quite common with axial depth, and 29-deg. thread, either axial or normal, it is very difficult to make the hob, as the tool must have such excessive clearance at the point that it may not be possible to make the hob with a tool that can be sharpened without changing its thickness. Also, the hob tends to drag on the side and any distortion in the hob or in the worm, due to hardening, interferes very much with the meshing of the worm and wheel. The differences in the shape of the various sections of a five-threaded worm as made with the straight-sided tool shown in Fig. 5 are shown in Fig. 6.

7 Worm No. 10 has five threads and was cut with a 29-deg. included-angle cutter 3½ in. in diameter, and the differences in

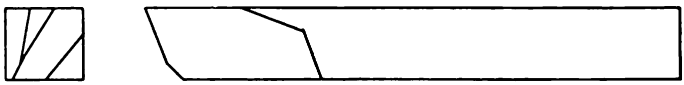


Fig. 5 Axial Lathe Tool for Worms Nos. 4 and 5, Table 1, Showing Unsuitable Cutting Edges

shape between tools fitted to the normal and axial sections of No. 10 worm may be compared with the normal and axial sections of worm No. 4.

8 When the helix angle is greater than 18 deg., it is well to consider using a greater pressure angle. To illustrate, worms Nos. 11 and 13 were made and cut with a cutter whose sides formed an included angle of 50 deg. No. 11 was cut with a cutter of 3½ in. diameter and No. 13 with one 6 in. in diameter. This change

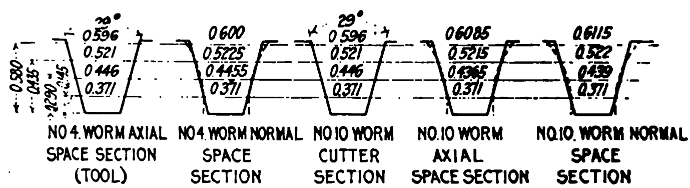


Fig. 6 Five-Threaded Worms

in size of cutter was made to show the difference in shape produced by two cutters of different diameters.

9 The most durable pair of worm gears, other things being equal, is that with a hardened and ground worm. If the worm is to be ground, the shape of the worm thread must be such that the thread surface can be readily reached, and when the wide-angle cutter is used this is possible. The best cutter to use is the one that will give ample working space. This angle, however, should be no greater than is necessary to obtain such results, as the smaller the angle the better, because the pressure on the bearings varies about as the tangent of the angle of pressure.

10 Worm No. 13 might be considered as having been ground with a wheel 6 in. in diameter and the difference in the shape on the axial and normal sections between Nos. 11 and 13 is shown in Fig. 7. A worm might be cut as No. 11 with a cutter 3½ in. in diameter and ground readily with a wheel 6 in. in diameter. The error, or difference, is principally at the outside of the worm thread, since there is a decided rounding off at this point. The hob made to conform to the finished worm would produce a shape to suit the ground worm. Fig. 7 shows, at *F* and *G*, how nearly straight the sides of the teeth are on both No. 11 and No. 13, a tool fitted to the axial section being compared with a straight-sided tool. This is also interesting as it forestalls the fear that there may be a loss in efficiency due to the lack of straight-sided teeth on the axial section.

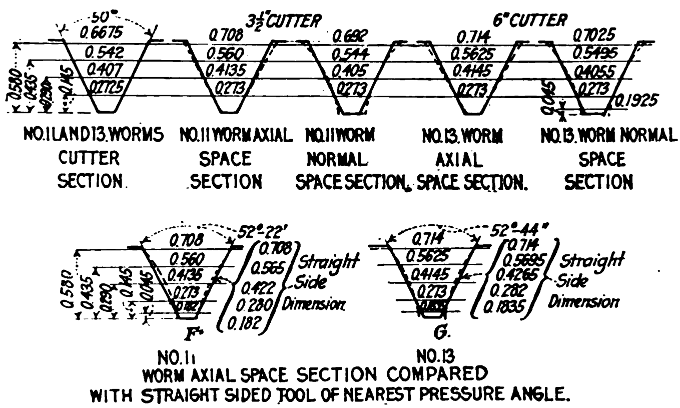


FIG. 7 FIVE-THREADED WORMS

Another advantage of this cutter is that it has straight sides, which as a starting point to make a worm or hob is most simple. This cutter can also be made as an ordinary milling cutter, which cuts more freely than a formed cutter which must be used if the section is other than that produced with a straight cutter.

11 If it is known that a given included angle is the basis for the cutter it can be produced by any one, even if its diameter is not known, as quite a difference in cutter diameters can be used without any serious difference in shape if it is borne in mind that the greater the angle of the cutter the less the variation in shape due to the diameter of cutter.

12 Another method of producing worms with five threads would be to hob them. Worm No. 14 was cut with a single-threaded

14½-deg. pressure-angle hob (29 deg. included angle) and the shape of the space on the normal is shown by dotted lines in Fig. 8, at *H*; this is compared with the normal section of worm No. 10, Fig. 6, and is reproduced at *A*, Fig. 8. It is apparent that this method produces something quite different and a shape that cannot be readily ground. If a hob of 25 deg. pressure angle or 50 deg. included angle on the axis was used, the shape produced would be much nearer that of worms Nos. 11 and 13.

13 That there is a general lack of knowledge of the foregoing

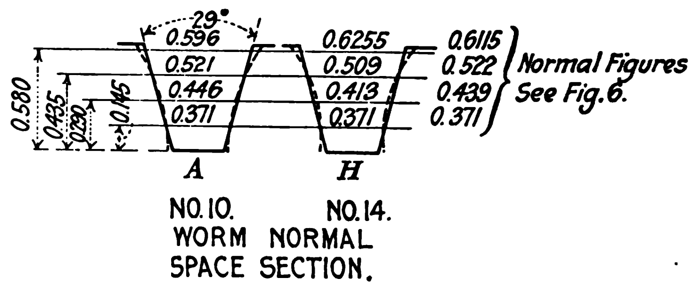


Fig. 8 Five-Threaded Worms

facts is apparent to any one who is familiar with the manufacture of worm gears and the tools for producing them. It would be desirable to establish and follow a standard line of procedure as the use of worm gears is increasing rapidly. Any method adopted, however, should be based on simple principles, such as straight-sided cutters with a change in pressure angle at some stated angle of helix and a change from axial to normal depth at the same point, and the desirability of this latter method is due to its mechanical advantages and not to any theoretical ones.

DISCUSSION

Elmer H. Neff said that the paper was a valuable contribution to the Society and that it emphasized the relations between the method of producing the worm, the method of making the hob and the operations of hobbing gears.

Henry J. Eberhardt read the following excerpts from a book by Edward Sang published by Adam & Charles Black, North Bridge, Edinburgh, in 1852, and entitled, "A New General Theory of the Teeth of Wheels."

"There is no such thing as a pair of wheels suited to each other alone; every pair forms part only of a system. Thus, if from any arbitrarily assumed wheel 'A,' we deduce a conjugate wheel 'B,' these typify two series of wheels, so related to each other that each one of the series 'A' may be geared into any one of the series 'B' and their permutability requires that the two series be identical: now the extreme limits of each series is the straight rack, and therefore the rack belonging to the series 'A' must be an exact counterpart of the rack belonging to the series 'B.'

"Referring to Fig. 9, *RTYVZ*, etc., is the outline of a rack which will generate a system of permutable wheels.

"Having made a cutter with *XTYVZ* for its outline, and mounted it on a carriage provided with a traverse motion and lead-

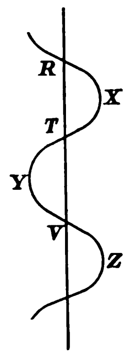


FIG. 9

ing screw, let us connect this leading screw with the stop-notch ring in such a way that one turn of the ring may give a transverse motion equal to *RV* the distance between two teeth. Then the cutter being arranged so that its pitch line may pass at the proper distance from the axis of the index wheel, bring up the cutter until in descending it slightly graze the edge of the prepared blank, and make in the ordinary way, a series of cuts all round. By turning the notch ring a little forward, bring the cutter deeper on the blank, make another series of cuts; again advance the notch ring, and continue this process until the cutter passes entirely clear of the metal, then shall we have a wheel whose teeth are truly formed to work with our assumed rack; and all wheels formed in this way will gear truly into each other.

"All wheels of one pitch are thus cut by one tool and, as the machine can easily be rendered self-feeding, the operation, though apparently tedious, need not be very expensive.

"Instead of having assumed the rack as the prototype of our system, we might have assumed any one wheel. Thus, to give the subject a practical aspect, having obtained a pinion wire, we may require to cut out the teeth of a wheel which may be led by it. Let the assumed form of the wheel be extended into a cylindroid as in the pinion wire; let it be notched so as to form a series of cutting edges as on the bar cutter; and let it be freed in such a manner as to slide parallelly to the axis of the index wheel, while it may be turned upon its own axis by means of an index plate. Having brought the pinion wire down upon the prepared blank so as to make a cut, repeat this in the usual way all around. Turn now the pinion wire on its own axis by a fraction of a tooth, and at the same time, turn the stop ring by a similar fraction of a revolution; repeat the cut all around in the usual way, and continue this until the stop ring has made a complete revolution, then is the blank toothed in such a way as to gear truly with the pinion."

HENRY HESS spoke of his early interest in the use of worm gear reductions and of the investigations of Professor Stribeck. These investigations, which were of a highly mathematical character, and impractical for the use of designers, were later reduced to a comprehensive basis by a student of Stribeck, Ernst, who published a booklet of great value to designers of worm gearing.

No. 1731

SCIENTIFIC DEVELOPMENT OF THE STEAM LOCOMOTIVE

By John E. Muhlfeld, New York, N. Y.
Member of the Society

The future of the steam locomotive has been a topic for frequent discussion, particularly since the electrification of the main line of the Baltimore and Ohio Railroad through the city of Baltimore in 1895.

Little attention was given to improving its efficiency until the tunnel, bridge and track-clearance and weight limitations and the rising costs for fuel and labor made it necessary to find means to increase capacity and economy, since which time the compound, Mallet articulated, and superheated-steam types of locomotives have been generally adopted, although the working steam pressures, due to the continuation of the existing locomotive type of boiler, have remained relatively low, with a few extreme cases of from 225 to 250 lb.

In this paper the author discusses at length the scientific factors that have been considered in the design and development of a new, high-powered freight locomotive for the purpose of substantially increasing the average thermal efficiency, as well as the maximum and sustained drawbar pull and horsepower per unit of weight, all of which are now limited by the capacity of the generally adopted boiler superheater.

STEAM railroads, to be successful, must, through executive foresight and engineering progressiveness, respect the same law of additions and betterments that applies to other profitable industries and which demands continuous modernization of plant and equipment in order to effectively and economically meet the necessity for greater production and speed by means and methods that will result in the least possible artificial age being capitalized in the improvements when installed.

2 Marked progress has been made in the development of the steam locomotive as the result of superior engineering ability. This progress, however, has been confined largely to an increase in size, weight, evaporating capacity and hauling power, and while the general use of superheaters and firebox baffle walls during the past

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

ten years has substantially assisted in improving sustained boiler capacity and increasing thermal efficiency as well as in keeping the steam locomotive in advance of the electric locomotive, the opportunity for further improvement in thermal and machine efficiency and to reduce smoke, cinders, sparks and noise is untold.

3 The desiderata in a steam locomotive may be summed up as: a reasonable first cost; maximum capacity for the service within roadway weight, curvature and clearance requirements; ability to handle the heaviest gross tonnage practicable at the highest permissible speed; positive control of mechanical operation; economy as regards fuel and water consumption and repairs; minimum manual labor for road and terminal handling; construction of the least number of parts, and capacity to perform continuous mileage without failure.

4 Modern types of steam locomotives fulfill quite satisfactorily all of these requirements with the exception of wastefulness in fuel, water and steam consumption, as may be gathered from the fact that the thermal efficiencies now obtained are only from 50 to 65 per cent at the boiler, from 60 to 75 per cent for the combined boiler and superheater, and from 4 to 6 per cent at the drawbar. These as compared with thermal efficiencies of from 3 to 5 per cent at the drawbar of an electric locomotive, 18 to 19 per cent at the switchboard of a modern steam-electric central power station, 25 to 30 per cent for internal-combustion engines, and 40 to 45 per cent as claimed for the full range of from one-quarter to full load for combination internal-combustion and steam motors.

5 The increase in the first cost and in the cost for labor, fuel, material and supplies for operation and maintenance of the steam locomotive has been most marked during the past ten years, particularly since the war. It is now being operated and maintained by highly paid enginemen and mechanics, with high-priced materials and supplies, and the machine and its performance must be brought up to a more respectable basis of engineering efficiency if it is to be perpetuated.

6 The supporting data of this paper, which apply to the United States, present the reasons why the general improvement of the steam locomotive should embrace the following changes which, it may be opportune to state here, are now being embodied in the design, specification and construction of a new type of locomotive, the first of which it is planned to have in regular service in 1920 on a prominent and progressive Eastern railroad:

a Steam at a pressure of about 350 lb. to be employed, superheated about 300 deg. fahr.
b Improved boiler, furnace and front-end design and appliances
c Greater percentage of adhesive to total weight, and a lower factor of adhesion
d More efficient methods of combustion
e Use of exhaust-steam heater and flue-gas economizer for boiler feedwater
f Better steam distribution and utilization
g Reduced cylinder clearances and back pressure
h Lighter and properly balanced reciprocating and revolving parts
i Lower heat, frictional and wind-resistance losses
j Improved safety and time-, fuel- and labor-saving devices.

EXISTING STEAM LOCOMOTIVES

TABLE 1 STEAM LOCOMOTIVES IN THE UNITED STATES

Item	Total no.	Tractive power, average, lb.	Weight on drivers, average, lb.
Single-expansion cylinders	61,336	30,500	135,000
Two-cylinder compound	500	32,000	145,000
Four-cylinder compound	1,300	33,000	148,000
Mallet articulated compound	1,750	79,000	350,000

7 The steam-locomotive stock in the United States on the railroads under the control of the United States Railroad Administration may be stated as follows:

Number of locomotives, total	64,886
Number equipped with coal stokers	4,010
Number equipped with fuel oil	3,358
Number equipped with coal fired by hand	57,518
Number equipped with superheaters	24,242
Number equipped with firebrick baffle walls	34,824
Tractive power, average, lb.	35,100

This total locomotive stock may be divided approximately as shown in Table 1.

8 During the month of June 1919 the cost per mile of service from these locomotives averaged as follows, the total of \$1.071 comparing with \$0.974 for June 1918:

Fuel	$0.391
Repairs	0.361
Enginemen	0.196
Enginehouse expenses	0.088
Other supplies	0.035
	$1.071

Assuming a total of 2,000,000,000 locomotive-miles per annum, at this average cost the locomotive service would represent a total expenditure of $2,142,000,000, or over $33,000 annually per locomotive.

REASONS FOR PERPETUATION OF THE STEAM LOCOMOTIVE

9 In August 1918 the newspapers reported that Director General William G. McAdoo, after a two months' tour of the West, had expressed the opinion that the great unused water power of the country ought to be developed for national electrification of the railroads and that he would recommend this program to Congress, if Government control of the railroads continued for any considerable period, for the purpose of reducing coal haul, releasing labor engaged in mining and handling fuel, and to conserve fuel.

10 The foregoing and much more has been said and written during the last few years about the electrification of the steam roads of the United States for the purpose of fuel and labor saving and conservation, but practically nothing has been set forth as to the possibilities to accomplish much greater results per dollar of investment and operating cost by a scientific development of the steam locomotive.

11 Referring for the moment to Mr. McAdoo's proposed railroad-electrification plan, the particular advantage of hydroelectric power is fuel saving and conservation, but as this does not offset the difference in the overhead charges as between a hydroelectric and a steam installation, the former can be considered commercially only where enormous quantities of power are available and used continuously for the purpose of reducing the cost for mass production and where the distance for transmission does not exceed economic limitations. For example, the logical place for large hydroelectric developments is in the Pacific States — in the western portion of Montana and in Idaho, Washington, Oregon and California — where over 50 per cent of the developed and potential flow and glacial water powers in the United States are located and commercially controllable. In the Atlantic and Central States, where the greatest power consumption takes place, no large continuous

water-power resources are to be found; therefore in those districts any national effort should be concentrated in the production of the largest possible saving through improvement in and changing of the existing wasteful methods in the use of the locally available fuels.

12 As the average use of power at any considerable load factor is for only 8 hours per day, and as there is more or less irregularity in the demand, due to the small use on Sundays and holidays, the available water power would be used only about 2400 out of a possible 8760 hours per annum, or about 27 per cent of the time, so that the remaining 73 per cent would be wasted. Therefore, where continuous water power is available it should be diverted to the special requirements of large and regular amounts, such as in electrochemical and metallurgical processes, in order to reduce this waste to a minimum.

13 Furthermore, as most water power is of variable or seasonal flow, with impounding impracticable and with large fluctuations between the minimum and maximum supply, the use of auxiliary steam plants becomes a necessity, and this again greatly reduces the economic value of the hydroelectric development. For example, during the year 1916 the Oakland steam plant, of a capacity of 10,500 kw. and an auxiliary to the hydroelectric installation of the Great Western Power Company, utilized only from 8 to 9 per cent of its capacity, and the Worth Beach steam plant, of 27,000 kw. capacity and an auxiliary to the hydroelectric installation of the Sierra and San Francisco Power Company, utilized only from 4 to 5 per cent of its capacity.

14 The methods at present employed for generating electric power from fuel in large modern central power stations represent from 18 to 19 per cent thermal efficiency, and as the investment cost for a steam plant is from one-third to one-quarter of that for a water-power development, the same total investment would produce from three to four times as much power from a steam as from a hydroelectric plant.

15 The fact that a St. Louis power company, in its purchases of current generated by water power at a rate of about $20 per horsepower-year, increased its average cost of steam and water power generated and was benefited only through the delayed investments in steam-plant equipment, and that the Buffalo General Electric Company has built a modern steam plant within view of the Niagara Falls, gives some idea as to the limitations of hydroelectric development.

16 Complete electrification of some portions of the large transcontinental trunk lines has been effected, all of which are representative of progressive engineering skill, but reliable reports and statistics available have not proven the actual operating economies predicted, and with the present unsettled state of the electrical art numerous objections present themselves, among which may be noted:

- *a* Prohibitive capital and non-productive cost per mile for road, equipment and facilities
- *b* Non-interchangeable equipment adaptable to certain electric zones only
- *c* Entire operation dependent upon single power plants and transmission systems
- *d* Widely varying load factors — dependent upon business conditions — requiring enormous outlay to meet uncertain peak movement and emergency conditions
- *e* Complication and congestion of road and terminal trackage with transmission and contact lines
- *f* First cost from five to ten times, and operating cost from two to three times, that of steam
- *g* Liability for complete tie-ups due to storms, snow, sleet, rain and short-circuits.

17 There is no comparing the 265,000 miles of railroad spread over an area of over 3,000,000 square miles in the United States, with an average population of 35 people per square mile, with a situation such as obtains in Belgium, where there are only 5500 miles of railroad within a concentrated area of 11,400 square miles, or less than that of the states of Massachusetts and Connecticut, and where the population averages over 650 people per square mile. Neither can comparison be made with France, which has 32,000 miles of railroad within an area of 207,000 square miles, with the population averaging about 200 people per square mile, and where the electrification of about 5000 miles of railroad is under consideration with hydroelectric power from the Alps, Pyrenees and the Upper Dordogne River.

18 For special cases, such as New York terminals, where the steam roads enter the city with long tunnel approaches, electrification is a necessity and in the interest of public service. This applies as well to the electrification of the Baltimore and Ohio through Mt. Royal tunnel in the city of Baltimore, and also to the electrification

of the Norfolk and Western at a point where it distributes its coal as expediting this traffic in a congested district was the logical plan for increasing main-line capacity. However, in the case of the St. Paul, where 440 continuous miles of line between Harlowton, Montana, and Avery, Idaho, are already electrified, and 211 additional miles into Seattle and Tacoma across the Rocky and Cascade Mountains are in process of electrification, the probability is that the investment, fixed charges and operating results during the next five years, particularly in view of insufficient uniform traffic density and the necessity for a combination of hydroelectric and steam-electric power even at a minimum cost for current of from 5 to 6 mills per drawbar horsepower-hour, will not justify the undertaking as compared with modernized steam-locomotive operation.

19 Therefore any general plan to electrify the steam roads to meet other than terminal and trunk-line congestion conditions, at an absurd cost, would mean lack of efficiency and prohibitive financing, which would result in bankruptcy for most of the railroads affected and in a further burden upon the public. In fact, it would be a source of real danger to the rehabilitation of these transportation systems, as to discard steam locomotives where coal or oil is available and can be burned with efficiency, comfort and economy, represents absolute waste.

PROPOSED ORDER OF DEVELOPMENT OF THE STEAM LOCOMOTIVE

20 With the operating ratio for the United States Railroad Administration railroads at 88.5 per cent and the cost of locomotive service averaging over 90 cents per 1000 gross ton-miles for the first half of 1919, the opportunity for steam locomotives to produce economy by increasing train loads, reducing transportation and mechanical delays and saving fuel and labor, is enormous.

21 In the design and specification for a new steam locomotive for any railroad the first and principal items for consideration are:

a Bridge, tunnel, track and terminal clearances, weight, and curvature limitations
b Service and train load
c Curves and grades
d Speed and length of runs
e Adaptation to fuel and water conditions.

22 With data on the foregoing items the general dimensions and weights can be determined upon, after which the adhesive

weight and tractive power can be calculated, the cylinder and boiler horsepower ratios established, and a general design outlined to meet the power and speed requirements.

23 The principal parts of a steam-locomotive assembly are the boiler, engine, running gear, tender, and special appliances, and the functioning of these parts in operation, jointly or independently, will involve particular factors that are capable of scientific development, viz.:

1 Design, Material and Workmanship
2 Adhesive Weight, Tractive Power and Factor of Adhesion
3 Tracking, Curving and Riding
4 Boiler Feedwater
5 Boiler-Feedwater Purifying
6 Fuel
7 Combustion
8 Boiler-Water Circulation
9 Heat Radiation, Convection and Conduction
10 Steam Generation
11 Steam-Pressure Increase
12 Steam Superheating
13 Steam Distribution and Utilization
14 Waste-Heat Distribution and Utilization
15 Friction and Resistance
16 Acceleration
17 Deceleration
18 Lubrication
19 Insulation
20 Safety Appliances
21 Special Appliances
22 Power for Accessories
23 Time Saving
24 Fuel Saving
25 Labor Saving

The supporting data relating to the improvement of these factors are presented in the following pages.

1 DESIGN, MATERIAL AND WORKMANSHIP

24 In this country the steam locomotive has not been classified as a refined piece of engineering mechanism, such as a marine

or stationary steam engine, due primarily to the indifferent conditions under which it has had to be operated and maintained and the relatively low cost for its fuel and supplies. However, with existing public demands for greater safety, speed, comfort, efficiency and economy in the movement of traffic, only by greater refinement in construction can requisite operating results be produced to offset the increased cost of equipment, supplies and labor.

25 Therefore the designing should now be done along more scientific lines through the substitution of boiler, cylinder and drawbar-horsepower and drawbar-pull calculations for tractive power; thermal efficiency for evaporation results; distributed for centralized thrusts, strains and stresses; light, high-grade alloy and high-carbon steels and other metals for heavy, low-grade plates, forgings and castings; and in the more general use of high-grade engineering practice in lieu of rule-of-thumb methods.

26 In the modern high-capacity locomotive it is necessary that certain parts be made as light as possible. On the other hand, the items of fatigue and shock of metals due to continued vibrations and impact as well as of inherent combinations of weakening chemical and physical characteristics, are responsible for many sudden failures of staybolts, plates, springs, axles, crankpins, tires, piston and main rods, frames and like parts that are subject to reversal of stress or to hundreds of thousands of repeated and localized loads. As it has been found that the elastic limit is not necessarily representative of the fatigue strength, these factors require that further careful research and study be made for the purpose of determining upon a reliable quick test that will insure against unsuitable material entering into the construction.

27 The same degree of refinement applies equally to workmanship for construction and upkeep, which should be brought up to the same standard as obtains in other machinery that is producing more efficient and economical power for other modes of travel.

2 ADHESIVE WEIGHT, TRACTIVE POWER AND FACTOR OF ADHESION

28 *Adhesive Weight.* In the ideal locomotive all of the weight is carried on the driving wheels for utilization as tractive power. The extended use of non-productive trailing wheels and the four-wheel leading truck has become an expensive fashion in that it has greatly reduced the percentage of total engine weight on drivers for adhesive purposes. For example, where a modern Mikado-

type locomotive will average 75 per cent adhesive to total engine weight, a modern Consolidation will run as high as 92 per cent, thereby utilizing much more of its weight to produce drawbar pull, hauling power and earning capacity.

29 Boiler design and weight distribution should be so correlated to the running gear as to make the use of trailer wheels unnecessary except where required by wheel-load limitations, and with the more recent improvement in constant-resistance leading-truck designs any four-wheel arrangement, except for high-speed passenger service, should be entirely satisfactory.

30 *Tractive Power.* The tractive "power," or "force," or "effort," of a steam locomotive is the pressure exerted by the action of the steam against the pistons in the cylinders to turn the driving wheels and cause the locomotive to advance, and is measured at the tread of the driving wheels, the internal friction of the engine being neglected.

31 In calculating tractive power the usual practice is to use 85 per cent of the indicated boiler pressure in lb. per sq. in. for two- and three-cylinder single-expansion, and 52 per cent for two- and four-cylinder compound engines. However, for a superheated-steam locomotive the use of a higher percentage of the indicated boiler pressure should receive due consideration when making tonnage-rating schedules before the train load is finally determined upon, as dynamometer tests have indicated that as high as 92 per cent for two-cylinder single-expansion locomotives is permissible for train-loading purposes.

32 *Factor of Adhesion.* In the same way that the leading- and trailing-truck type of locomotive has reduced the percentage of adhesive weight, so also has it increased the factor or ratio of adhesion, due to the "bridging effect" thus obtained over the driving wheels, tending to release the weight on the latter.

33 Whereas many successful Consolidation types of locomotives are now operating with factors of adhesion of from 3.55 to 4.0, the Mikado types will usually range from 4.0 to 4.5.

34 The coefficient of static friction or adhesion between driving-wheel tires and very dry, clean rails reaches a maximum of about 0.35, and for moist, muddy, greasy and frosty rails a minimum of from 0.15 to 0.20, giving factors ranging from 2.85 to 6.65.

35 In general, the factor of adhesion should be as low as practicable in order that the maximum power will always be available to start trains that can be easily handled when in motion and

should about equal the ratio between the limiting friction in pounds and the weight on driving wheels in pounds, which, for average dry rails is from 3.5 to 4.

3 TRACKING, CURVING AND RIDING

36 With the increased length, higher center of gravity, extended front and back overhang, and smaller proportion of spring-borne weight there have been many difficulties to overcome in order to maintain proper tracking, curving and riding qualities in locomotives of great power, and in the majority of cases these have been met with unusually satisfactory results.

37 Certain changes can be made, however, that will bring about a general betterment in the way of reduced rolling, oscillation, nosing and pounding: namely, reduced spread of cylinders; more uniform distribution and equalization of weight over driving and truck wheels; maximum permissible diameter of driving wheels; reduction in weight of revolving and reciprocating parts and counterbalance and proper distribution between wheels; improvement in constant-resistance lateral-motion devices; more uniform cylinder pressures when using steam and drifting; and greater refinement in control of end play, wheel and rail clearances, and tire-tread coning.

38 As the reciprocating forces must be neutralized by means of a revolving body, which cannot produce a perfect balance, it is of the greatest importance that every allowable effort be made to reduce the weights of pistons, crossheads and main rods and divide all weights as uniformly as practicable over all driving wheels, in order that excess balance may be reduced to the minimum and the greatest possible static weight permitted on the driving wheels.

39 As the centrifugal power force of surplus counterbalance, the swinging movement of spring-borne weight, and the rotative force on the crankpins are constantly changing in combination with speed and cut-off, the importance of giving particular attention to all of the foregoing factors cannot be overestimated.

4 BOILER FEEDWATER

40 No item in the operation and maintenance of steam locomotives contributes to greater unnecessary cost than boiler feedwater containing excessive incrusting, corrosive and foaming elements. The use of such water not only results in continual delays

and expense for washing, rinsing and blowing out, but also in premature repairs and renewals of boiler and firebox sheets, flues, tubes, seams, staybolts, valve and cylinder parts, and superheater elements, due to pitting, grooving, corrosion, incrustation, overheating or priming, all of which increase the fuel, water and lubricant consumption and cause continual and otherwise avoidable delays and expense in the movement of traffic.

41 Observations and experiments indicate that any scale porous to water has little effect on boiler economy. However, such scale when dried out or hardened next to the metal by the expulsion of the carbonic acid, as usually occurs when boilers are forced, will not only become an excellent heat insulator and cause a heat loss of about 10 per cent when $\frac{1}{8}$ in. thick, but it exposes the sheets and staybolts to overheating and "mud burning," with resulting leakage and shopping for repairs and cleaning.

42 In the preparation of recent data on the expense resulting from bad water supplied to steam locomotives on eight divisions of a large Eastern trunk line, the cost averaged $1000 per locomotive per annum for fuel, repairs, time out of service and boiler washing, with specific cases of mud burning costing as high as $400 for repairs and time out of service only.

43 In view of the increasing size of locomotive boilers and the high ratings to which they are subjected, the importance of purifying unsuitable water to prevent incrustation, corrosion, leakage and burning, as well as to eliminate delays and cost for cleaning, repairing and extra fuel consumed, cannot be overestimated, and until the many existing conditions of this kind are corrected, neither the existing nor improved steam locomotives can be expected to render satisfactory and economical service.

5 BOILER-FEEDWATER PURIFYING

44 When an adequate and suitable supply of boiler feedwater cannot be obtained from the usual sources, then the proper treatment of the available unsuitable water becomes necessary by settling; filtration; chemical treatment in treating plants, supply tanks or tenders; or, in the case of suspended matter and carbonates, by partial purification in a combination open and closed type of exhaust-steam feedwater heater on the locomotive.

45 While the supplying of suitable natural or treated boiler water to the locomotive tender is the most satisfactory and economical method, in the absence of such the tender treatment or feed-

water-purification method will be an improvement over feeding the raw water into the boiler without treatment, or attempting to treat it in the boiler.

6 FUEL

46 The principal fuels now used in steam locomotives are the commercial grades of bituminous and anthracite coal and fuel oil. While millions of tons of the by-products of anthracite and bituminous coal mining are available, as yet practically no progress has been made in their utilization although satisfactory means and methods are now developed. This applies as well to the enormous deposits of sub-bituminous coal and lignite that are only awaiting mining operations to come into effective and economical use as locomotive fuel.

47 Regardless of the kind of fuel now used by steam locomotives, more general attention is being given to its proper preparation for the class of service to be performed and the method of firing to be followed, before it is supplied to tenders. However, the factors of kind and size of coal and method of firing must each be carefully considered and coördinated in order to insure the best results, as may be shown by Table 2 which gives comparative performances of a stoker-fired modern Mikado type of locomotive with superheater and firebrick baffle wall supported on water-circulating tubes.

48 It will be noted from Table 2 that while the heat value (Item 9) of each of the four fuels tested was practically the same, there is a difference of over 30 per cent in the quantity fired and of over 43 per cent in the fuel cost for the same work done, when comparing the poorest with the best performance, and that the use of either grade of soft coal is absolutely prohibitive so long as either grade of gas coal can be obtained.

49 With the cost for locomotive fuel on tenders practically doubled during the past two years, and next to labor the largest single item of railway operating expense, the best methods for its use will now begin to receive the consideration that this large item of operating cost justifies.

7 COMBUSTION

50 As a matter of historical interest the following data are presented as taken from a copy of the report made November 21, 1854, by Captain Geo. B. McClellan, Corps of Engineers, to Jefferson Davis, Secretary of War, on the Baltimore and Ohio, Pennsylvania

Central, Virginia Central, Massachusetts Western, Boston and Worcester, Boston and Providence, Boston and Maine, Boston and Lowell, and Burlington and Rutland Railroads.

On the Boston and Maine Railway a load of 170.5 tons (gross, exclusive of engine and tender — net tons of 2000 pounds) was drawn 74 miles at a velocity of 14.5 miles per hour, and the average of six trips gave as a result that 10.59 pounds of anthracite evaporates 7.48 gallons (1 cubic foot) of water (5.88 pounds of water per pound of coal). The trip with Cumberland coal indicated that 9.19 pounds of it will evaporate 7.48 gallons of water (6.78 pounds of water per pound of coal).

The average waste of steam while engines are at rest, stopping on the road, steaming up, etc., is one-third of the whole amount generated.

51 Comparing the foregoing figures of 65 years ago with the average saturated-steam locomotive performance of today, it will be noted that little improvement has been made in the average road service.

TABLE 2 COMPARATIVE TESTS OF A STOKER-FIRED MIKADO-TYPE LOCOMOTIVE USING DIFFERENT SIZES AND KINDS OF COAL

Item	Test No.	1	2	3	4
1	Coal, kind	Bituminous	Bituminous	Bituminous	Bituminous
2	Coal, class	Gas	Gas	Soft	Soft
3	Coal, grade	Nut, pea & slack	Slack	Run-of-mine	Screenings
4	Coal, moisture, per cent	1.23	1.57	0.75	0.81
5	Coal, volatile, per cent	36.47	35.74	18.17	17.52
6	Coal, fixed carbon, per cent	53.94	52.78	69.07	70.06
7	Coal, ash, per cent	8.36	9.91	12.01	11.61
8	Coal, sulphur, per cent	2.59	3.30	3.33	2.31
9	Coal, B.t.u. (calorimeter)	13,910	13,790	13,880	13,970
10	Total miles run	505	505	505	505
11	Kind of firing	Stoker	Stoker	Stoker	Stoker
12	Coal per average drawbar hp-hr., lb.	3.74	4.19	4.14	4.89
13	Boiler efficiency, per cent	46.6	41.9	46.1	38.0
14	Relative pounds of coal fired, per cent	100	111.87	110.54	130.56
15	Relative cost of coal, per cent	100	102.14	123.58	143.64

52 The locomotive fuel bill for the year 1918 was approximately $750,000,000, and while full recognition is given to the fact that from 25 to 50 per cent of the available energy in the fuel is still needlessly wasted and that present methods of mechanically firing, as compared with the average hand firing, and burning coal on grates or in retorts increase this waste, but little has been accomplished in regulating combustion so that this loss may be reduced.

53 The capacity of the average steam-locomotive boiler is dependent upon the activity, temperature and radiation of combustion, which in turn are usually controlled by the limitations of combustion when fuel is burned on grates, the furnace volume and evaporating surfaces, the length of the boiler flues, tubes and baffle-wall arrangement, and the draft, and not so much upon the inability of the evaporating and superheating surfaces to absorb the heat.

TABLE 3 CONSUMPTION OF DRY BITUMINOUS COAL BY LOCOMOTIVE WITH THE BEST HAND FIRING

Total indicated horsepower of locomotive	Dry coal per i. hp-hr.
500	2.8
750	2.7
1000	2.6
1250	2.5
1500	2.6
1750	2.8
2000	3.0
2250	3.2
2500	3.4

54 The combustion rate generally follows the increase in draft until about 100 lb. of bituminous and about 50 lb. of anthracite coal are burned per square foot of grate area. After this the additional coal supplied is not effectively consumed due to the difficulty in supplying sufficient air, uniformly distributed, through the grates and fuel bed to oxidize the fixed carbon and volatile matter in process of combustion without a large excess of air such as obtains when forcing takes place, and it becomes necessary to open the fire door so that combustion can be completed by the admission of air above the fuel bed.

55 The greatest loss in heat is that due to the heat carried off in the stack gases, sparks and cinders, which usually results in a smokebox temperature of from 500 to 750 deg. fahr. for the best practice. Adding to this the heat losses due to combustible in ash, vapors of combustion, carbon monoxide and otherwise, leaves an average of from 25 to 40 per cent of the heat in the fuel as fired unabsorbed by the boiler and superheater.

56 Where locomotives are worked at from 25 to 35 per cent cut-off and hand-fired, with a thermal efficiency of about 65 per

cent for the combined boiler and superheater, the heat balance will be approximately as follows:

	Per cent
Heat absorbed by boiler	55
Heat absorbed by superheater	10
Heat loss in smokebox gases	14
Heat loss in cinders	8
Heat loss in vapors of combustion	4
Heat loss in combustible in ash	3
Heat loss in carbon monoxide	2
Heat loss in radiation and unaccounted for	4
Total	100

57 However, at high rates of boiler capacity and draft, when stoker-fired coal is burned on grates the front-end and stack cinder and spark losses will run as high as from 12 to 25 per cent, the carbon monoxide from 2 to 7 per cent, and the unburned fuel from 10 to 35 per cent.

58 With the best hand firing, when using dry bituminous coal averaging 14,400 B.t.u. and 60 per cent fixed carbon, 32 per cent volatile and 8 per cent ash, the fuel rates in Table 3 will usually obtain.

59 As compared with hand firing, stoker firing will result in an increase of from 10 to 25 per cent in the fuel fired, while if the same coal be pulverized and burned in suspension there will be a decrease of from 15 to 25 per cent in the amount of fuel fired.

60 As the locomotive firebox, which in the best practice represents only from 7 to 10 per cent of the total boiler evaporating surface, must generate all and absorb from 30 to 40 per cent of the heat energy that is converted into drawbar horsepower, the fuel effectively consumed, not fired, is the measure of work done. Therefore the largest permissible combination of firebox and combustion-chamber volume, heating surface and grate area should be provided and equipped with an arrangement of firebrick baffle walls placed on water-circulating supports in a manner to produce long flame travel, high firebox temperature and the maximum radiant heat for absorption by the surrounding water.

61 With the usual limitations in firebox volume, too much importance cannot be placed on the arrangement of heat-absorbing and -radiating walls for the purpose of flame and radiant-heat propagation. Carefully conducted tests have shown that the best results are obtained from solid firebrick baffle walls and that the unburned-gas, coal-dust, spark, cinder and smoke losses are reduced with an

increase in their length and gas-passage arrangement, and a saving of from 10 to 15 per cent in bituminous coal as fired is effected.

62 The greatest difficulty in controlling combustion occurs at high horsepowers and long cut-offs, where grates are used, and for the best results the air openings should be equal to about 50 per cent and those in the ashpans to about 15 per cent of the total grate area so that firebox temperatures of from 2000 to 2500 deg. fahr. can be obtained and the unburned solid fuel, carbon monoxide and excess air over the fuel bed reduced to the minimum.

63 Other important factors influencing combustion, as well as evaporation and superheating, that should receive consideration are: ratios of length to diameter of boiler flues and tubes and the spacing between them; distribution of gas area between boiler flues and tubes; the effect of closed superheater dampers on firebox draft when locomotive is not using steam; free passage of gases through front end by elimination of unnecessary baffles, steam pipes and superheater parts; arrangement of exhaust stand and nozzle to change form of exhaust jet and produce greater entrainment of gases and improved coördination of exhaust jet and stack.

64 As with dry pulverized coal of 12,430 B.t.u. value, an average boiler efficiency of 69.2 per cent at 1080 boiler hp., and an average combined boiler and superheater efficiency of 78.1 per cent at 1220 boiler hp., with an equivalent evaporation averaging 42,100 lb. of water per hour from and at 212 deg. fahr., has already been obtained on a Mikado simple-cylinder type of locomotive hauling fast freight trains over a 113-mile division, the possibilities for reducing the steam-locomotive fuel consumption are practically unlimited and much remains to be done in this direction by good hand firing, through a combination of the fireman's eyes, brain and brawn, provided the thermal efficiency of the modern locomotive at the drawbar is brought up to where it can and should be.

8 BOILER-WATER CIRCULATION

65 Water is practically a non-conductor of heat but expands when heated above 39 deg. fahr. and rises due to its relatively lower specific gravity. Unimpeded circulation will therefore increase its ability to take up heat, maintain greater uniformity of temperature throughout the boiler, and decrease the liability of incrustation of heat-absorbing surfaces and of priming.

66 In designing a boiler it is extremely desirable to secure the

most rapid circulation practicable, as with high combustion rates and temperatures and the abnormal state and behavior of the water film in contact with the heating surfaces, the load on the firebox sheets is very intense, the conduction rate averaging from 75,000 to 100,000 B.t.u. per square foot of evaporating surface per hour.

67 Therefore, in order to avoid resistance to heat transfer, with resultant overheating of metal and reduced efficiency, a relatively high velocity of circulation and at least a rate of 125 ft. per min. in the most sluggish locality is very essential.

68 The average locomotive boiler, with its combination of cylindrical and box shell, water legs, staybolts and rods, flues, tubes and generally irregular design of water spaces does not present ideal water-circulation possibilities, but the enlarging of contracted spaces, increasing of water-leg, flue and tube clearances, and provision of suitable outlets from choked water pockets will not only reduce the resistance to the "slip" of the steam bubbles through the water, but will enable the accelerated action of the former to increase the velocity of the latter and thereby improve general circulation and heat-transfer results.

9 HEAT RADIATION, CONVECTION AND CONDUCTION

69 The transmission of the heat of combustion produced in a locomotive boiler is by means of radiation and convection to the firebox, flue, tube and superheater heating surfaces, by conduction to the water in the boiler and the steam in the superheater, and by convection through the boiler water and the superheater-steam mass. In addition there are the direct radiation losses, which in many instances are considerable.

70 *Heat Radiation.* The transformation of the molecular energy of a hot body into the wave motion of the surrounding ether and the propagation of these ether waves through space is termed heat radiation; and in a locomotive boiler the efficiency of combustion heat transfer through the firebox plates and boiler flues and tubes is from 20 to 25 per cent greater as applying to those heating surfaces directly affected when subjected to the radiant effect of the incandescent combustible and non-combustible particles which have passed through the minimum distance, than the heat-transfer efficiency when convection only is available. For example, when coal is hand- or stoker-fired and burned on grates or in retorts the radiant heat is at a minimum and applies only to the heat-absorbing

surfaces adjacent to the fire bed while the heat of convection is at a maximum; whereas when the coal is burned in pulverized form in suspension this condition is reversed, as is evidenced by the intense incandescent flame which obtains not only in the furnace and combustion chambers of the firebox proper, but well into the boiler flues and tubes. The locomotive boiler of the future will undoubtedly depend more largely on radiant heat.

71 With respect to the loss of power through radiation to the atmosphere from all parts of locomotive boilers and machinery that are generators and containers of heat and pressure — to prevent which rather indifferent efforts have as yet been put forth, as the rate at which this loss of heat extends will depend upon the difference between the temperature of the body emitting the heat and the temperature and velocity of the surrounding atmosphere, there is sufficient justification for completely and properly lagging the boiler, firebox, cylinders and heads, steam chests and all other radiating surfaces, as well as for polishing certain machinery parts, in order to reduce the dissipation of heat that now takes place through these parts from the existing steam pressures and superheat.

72 *Heat Convection.* The process whereby the diffusion of heat is rendered more rapid by the movement of the hot substance from one place to another is termed convection, and in the locomotive boiler applies particularly to the transfer and diffusion of the heat in the products of combustion throughout the firebox, flues, tubes and superheater by means of the smokebox draft and in the carrying of the heat through the boiler-water mass by the currents produced by circulation. In the present locomotive boiler by far the greatest proportion of the heat is imparted to the evaporating and superheater surfaces by convection.

73 To secure the fullest benefit from heat convection the combustion volumes and gas areas must be so coördinated as to establish a "velocity pressure" or "frictional" action between the gases and the heat-absorbing plates and tubes in order to increase the rate of heat transmission. Likewise must the boiler circulation be expedited in order to quickly disengage and release the steam bubbles from the water side of the same plates and tubes in the final heat transfer.

74 The possibilities for improving heat transmission by convection in the locomotive boiler with its high water rate, i.e., a boiler horsepower for an average of less than two square feet of total evaporating surface, fully justifies additional study.

75 *Heat Conduction.* The transmission of heat from one body of high temperature to another body of low temperature by contact, and from one part of a body to another part, is termed external and internal conduction, respectively, and in the locomotive boiler is principally associated with the thermal conductivity of the firebox, flue, tube and superheater materials and with the accumulation of soot and scale on the fire and water or steam sides, respectively.

76 Any increase in the rate of external conductivity, considering the present kinds and thicknesses of firebox, flue and tube materials as practically fixed, must be through an increase in the rate of flow of the heated gases, and this in turn means the expenditure of a greater amount of energy to pull these gases through the boiler.

77 However, questions as to the proper gas areas, rate of flow of gases, best sizes of flues and tubes for the maximum rate of heat transfer, and relating to like factors should be carefully analyzed in order that the highest absorptive efficiency may be obtained, not only with the high but also with the low gas temperatures. While there is no difficulty in now obtaining a boiler horsepower from each 1½ to 2 sq. ft. of total evaporating surface, whatever further improvement can be made in this direction will provide just that much more margin of boiler over cylinder horsepower requirements and produce a corresponding gain in efficiency.

10 STEAM GENERATION

78 Efficient absorption of heat for the generation of steam in the modern locomotive boiler can be more readily provided for than can suitable feedwater, effective boiler-water circulation, efficient combustion or the maximum pounds of dry saturated steam per hour, which latter is a fundamental requirement.

79 In present locomotive operation the quality of the steam, i.e., the percentage of vapor in a mixture of vapor and water, is one of the most important and least-referred-to factors in road and laboratory test reports, particularly as the average modern locomotive boiler is notorious for delivering saturated steam to the superheater or to the steam pipes with a high percentage of entrained moisture. This is due largely to the relatively small steam space in the boiler, the close proximity of the water level to the throttle valve and the backlash due to the firebox tube sheet, and also to the fact that the most rapid movement of the steam is next to the throttle valve

so that any water coming near it is immediately entrained due to the high velocity.

80 Road tests recently conducted on modern Mikado types of locomotives showed an average quality of from 94.7 to 96.3 per cent for the saturated steam as delivered to the superheater, indicating from 5.3 to 4.7 per cent of moisture, which is valueless so far as its power for doing work is concerned but which greatly increases the work to be performed by the superheater by throwing upon it work which should properly be done in the boiler.

81 The delivery of dry saturated steam from the boiler is an item that has been given but little consideration in steam-locomotive practice, the principal idea having been to produce evaporating capacity and depend upon the superheater to perform auxiliary boiler functions. Many changes can and should be made to improve this condition.

11 STEAM-PRESSURE INCREASE

82 One of the greatest and simplest improvements to be made in the steam locomotive can be effected by an increase in the boiler pressure in combination with greater quantity and better quality of saturated-steam production, higher and more uniform superheat, and compounding.

83 The writer advocated a steam pressure of 250 lb. in 1902 when with the Canadian Government Railways, and inaugurated the use of 235 lb. boiler pressure in combination with 21-ft. boiler tubes in 1903 in the Baltimore and Ohio Railroad Mallet articulated compound locomotive No. 2400, with excellent results from both an operating and maintenance standpoint. This at a time when the general tendency was to reduce rather than to increase locomotive steam pressures from an established practice of about 200 lb.

84 While the loss in steam pressure between the boiler and the valve chests of saturated-steam locomotives is considerable, this loss is substantially increased in a superheated-steam locomotive, as will be noted from the approximate data in Table 4, taken from a laboratory test of a Pacific-type locomotive operating at a uniform rate of speed.

85 Other tests have also indicated that the loss in boiler pressure at the valve chests when working at low rates of speed and cut-off will be about 5 per cent, at medium rates about 10 per cent, and at high rates about 15 per cent.

86 During recent years stationary-boiler engineers have not only determined upon their efficiency but have inaugurated the use of relatively high steam pressures, and with the urgent necessity for keeping the cylinders as small in diameter and the reciprocating and revolving parts as light as practicable, there would appear to be no good reason for not now utilizing saturated steam of 350 lb. pressure, which, in combination with 300 deg. fahr. of superheat, should provide, in addition to the many other advantages, a much greater opportunity for economy in power generation. In fact, many small single- and double-expansion compressed-air locomotives now in use with from 40,000 to 50,000 lb. total weight on the driving wheels and 12-in. × 16-in. high-pressure cylinders, carry from 800 to 1000 lb. air pressure in their storage tanks, which in turn is reduced to 250 lb. pressure before entering the high-pressure cylinder, and are operated without trouble and with considerable efficiency and economy.

TABLE 4 LOSS IN STEAM PRESSURE BETWEEN BOILER AND VALVE CHESTS OF A SATURATED-STEAM LOCOMOTIVE

Test No.	1	2	3	4
Speed, miles per hour	56	56	56	56
Boiler pressure, lb.	206	206	206	206
Superheat, deg. fahr.	150	210	255	265
Cylinder cut-off, per cent of stroke	20	30	40	50
Loss in boiler pressure at valve chest, lb.	6	12	19	31

12 STEAM SUPERHEATING

87 The use of superheated steam has done more to increase sustained hauling power, reduce fuel and water consumption and increase thermal efficiency than any of the other means and methods that have been generally adopted on the steam locomotive since its introduction, either singly or in combination. Sustained hauling capacity is increased due to the longer cut-off possible at comparative speeds and fuel and water economy result from the elimination of cylinder condensation, the increase in efficiency being progressive and in proportion to the amount of superheat up to the point at which the exhaust steam begins to show superheat.

88 With the average superheat now used, from 175 to 250 deg. fahr., the drawbar pull at a speed of 20 miles per hour is in-

creased about 15 per cent, and at 50 miles per hour about 40 per cent; and due to the combination of superheat, larger diameter of cylinders and reduced cylinder back pressure — resulting from the use of superheated steam — it is possible to increase train tonnage about 30 per cent at speeds of about 30 miles per hour.

89 In the best existing steam-locomotive practice the superheat generally increases with the cut-off up to 50 per cent cut-off, beyond which there is usually a falling off in the superheat. Furthermore, with short cut-off a fair water rate, i.e., about 19 lb. per i.hp., can be maintained; but if the cut-off at the same speed is increased to over 50 per cent the superheat must be increased to about 300 deg. fahr. in order to maintain the same water rate, or otherwise, for example, at 67 per cent cut-off, the steam consumption will increase to 21 lb. or more per i.hp. This for the reason that as the amount of superheat is increased the range of temperature in the cylinder during the stroke of the piston is decreased, until with sufficient superheat the changes in temperature cease entirely.

90 While the increased superheat results in a greater number of B.t.u. being exhausted from the cylinder, any such loss of a marked degree is more than offset by the smaller amount of heat exhausted per stroke, due to the fewer B.t.u. admitted to the cylinder per stroke at a given cut-off.

91 The use of highly superheated steam results in a saving of about 35 per cent of the total water evaporation per unit of power and in from 10 to 45 per cent saving in fuel, when using steam, depending upon the power output.

92 Existing fire-tube superheaters produce the maximum superheat only when the locomotive is forced to its boiler capacity, whereas the maximum economy is more desirable when the locomotive is working under average conditions at economical cut-offs and when the superheater should give as nearly as possible a uniform degree of high superheat under all conditions of working, regardless of the boiler evaporation. For example, if the degree of superheat obtainable at speeds of 50 miles per hour with 50 per cent cut-off could be obtained at 25 per cent cut-off, a water rate of considerably less than 15 lb. could be obtained as compared with existing rates of about 19 lb. Therefore, as the present limitation in the hauling power of the modern superheated-steam locomotive is the capacity of the boiler to produce continuously sufficient dry saturated steam of high pressure and of the superheater to maintain a uniform high degree of superheat, the possibility of improving it by

means of average higher boiler pressures and superheat temperatures and better utilization of fuel, steam and waste heat, in combination with radical changes in the design and arrangement of the boiler and superheater equipment and in the saturated- and superheated-steam connections, offers one of the greatest opportunities to increase efficiency and economy. This applies particularly to the larger locomotives, many of which consume more fuel and water and do less work than the smaller locomotives of the same general design and equipment.

93 The proposed changes, while applying especially to the production of greater efficiency at economical cut-offs for maximum power and speed, would also improve the maintenance and operation of superheaters, boilers, flues, front ends, valves, cylinders and exhaust nozzles and provide for the better equalization of a lower draft through the flues and tubes, lower front-end temperatures, less throwing of smoke, sparks and cinders, and lower cylinder back pressure, all of which would reduce loss of power, fuel consumption and wear and tear on machinery.

94 Some of the particular troubles reflected in both maintenance and operation, due to the existing, generally used boiler and superheater equipment, may be stated as follows:

a Air leaks around outside steam pipes where they pass through the front ends, resulting in steam failures, burning out of front ends, reduction in the size of exhaust nozzles for the purpose of making engines steam, and increased water and fuel consumption

b Joints between superheater units and the saturated and superheated chambers of the headers leaking, and the cutting out of the units at the neck, between the ball joint and the tube

c Too little water and steam space over top of firebox and combustion-chamber sheets and flues, particularly on grades and curves, contributing to lower superheat temperature and cylinder efficiency, and to superheater-unit tubes distorting due to entrained water being carried over with the saturated steam from the boiler to the superheater, causing obstructions in and damage to superheater tubes and obstructions at the header

d Extreme losses in steam pressure between boiler and steam chests *e* Boiler flues clogging, due to ash and cinders packing in around return bends and centering clamps and tubes.

95 While the superheater has generally been considered as a part of the boiler, particularly as regards its evaporation of entrained moisture in the saturated steam, it has no relation whatsoever thereto insofar as its individual functioning is concerned, and the more that the saturated-steam-conducting and superheated-steam-delivering conduits, as well as the superheater equipment in itself, can be divorced from the boiler and front-end connections and their proper functions without introducing separately fired apparatus, the better will be the general results from the standpoint of efficiency, maintenance, operation and economy of the locomotive as a whole.

96 Some of the points to be considered in correcting existing deficiencies may be stated as follows:

97 *Steam Temperature.* The steam temperature should be uniform for the variable speeds and capacities of operation. At the present time high temperatures obtain only at high speeds and capacities. A minimum temperature of 650 deg. fahr. quickly after starting, and of 700 deg. at maximum power and speed, would be much more effective and economical. For example, a locomotive equipped for generating 350 lb. steam pressure and 300 deg. superheat — representing a total temperature of about 736.4 deg. fahr. — will, as compared with one using 200 lb. steam pressure and 300 deg. superheat — representing a total temperature of about 687.9 deg. fahr. — require an increase of only 18 B.t.u., or 1.3 per cent in total heat in the steam, and an increase of only 48.5 deg., or 7.05 per cent in the temperature of the steam to produce an increase of 150 lb. or 75 per cent in the steam pressure.

98 *Dome or Steam Outlet.* This should be fitted with baffles for the purpose of reducing liability of priming and entrainment of water with saturated steam.

99 *Saturated-Steam Delivery Pipe.* This should be located outside of the boiler and be of adequate cross-sectional area to reduce steam-pressure losses.

100 *Steam Trap or Separator.* A steam trap or separator should be installed between the saturated-steam delivery pipe and the superheater saturated-steam chamber for the purpose of further eliminating moisture and condensation from the superheater units and also as a reëvaporation chamber.

101 *Superheater Header or Saturated- and Superheated-Steam Chambers* should be removed from the interior of the front end.

102 *Superheater Units* should consist of not more than two tubes per boiler flue and should be of such design and arrangement as will admit of location close to the top of the flue, in order to permit free passage for cinder and ash and cleaning of flues.

103 *Unit Joints to Saturated- and Superheated-Steam Chambers.* Unit joints should be removed from the direct path of gases and cinders so as to avoid cutting out, and should be supported in a positive, equalized and flexibly yielding manner to prevent leakage due to the loosening of one joint causing the loosening of another and so that the joint bolts can be tightened at the top of the header castings.

104 *Superheater Dampers.* These should be kept in good operating condition so that when the steam ceases to flow through the superheater units the products of combustion will stop flowing through the superheater flues, particularly when drifting at high speeds.

105 *Steam Delivery Pipes from the Superheated-Steam Chamber* should be made of adequate cross-sectional area to reduce steam-pressure losses and removed from the interior of the front end so that no joint between these pipes and where they pass through the front end will be necessary.

106 *Automatic Saturated-Steam Supply When Drifting.* This is essential to eliminate the human element and insure a proper supply of saturated steam with the superheated steam just before the throttle closes and continuously thereafter. A jet of saturated steam should also be supplied to the exhaust nozzle to neutralize the gases ordinarily drawn through the same into the valve chests and cylinders.

13 STEAM DISTRIBUTION AND UTILIZATION

107 The work developed by a steam locomotive is measured in drawbar pull and drawbar horsepower and its hauling power at speed is dependent entirely upon its ability to maintain the mean effective pressure. In other words, a locomotive developing a pull of 12,000 lb. at the drawbar at a speed of 37½ miles per hour travels at the rate of 3300 ft. per min. and develops 39,600,000 ft-lb. or 1200 drawbar hp. When developing 6000 lb. drawbar pull at 75 miles per hour the rate will be 6600 ft. per min. and the drawbar hp. developed will be 1200, or the same as in the previous case. There-

fore drawbar pull should be considered as the force required to do the work and drawbar horsepower should be considered as the measure of the rate for doing the work, and the two should not be confused in calculating the hauling capacity and speed which may be produced by any type of locomotive.

108 Modern types of locomotives have developed at low speeds 3000 i.hp. and at high speeds 3200 i.hp., and comparative average water rates through the complete range of the effective capacity of the locomotive, with piston speeds of from 600 to 1000 ft. per min., have been obtained as shown in Table 5. At piston speeds of less than 600 ft. per min. the water rate of the double-expansion saturated-steam locomotive will approximate that of the single-expansion superheated-steam locomotive.

TABLE 5 COMPARATIVE WATER RATES OF LOCOMOTIVES WHEN USING SATURATED AND SUPERHEATED STEAM

Cylinders	Steam	Water rate per i.hp-hr., lb.
Single-expansion	Superheated	16 to 20
Single-expansion	Saturated	24 to 29
Double-expansion	Superheated	15 to 18
Double-expansion	Saturated	19 to 22

109 *Compounding.* With the exception of the Mallet articulated type of compounding, the multiple-expansion system of steam utilization, which has been so successful in marine and stationary practice, has not made the progress in this country that it has in Europe.

110 The failure of various types of cross, four-cylinder, four-cylinder balanced and tandem double-expansion locomotives, introduced from 25 to 15 years ago, to produce the predicted economy was due largely to factors of indifferent design, low boiler pressure, excessive condensation, lack of proper maintenance and operation, poor fuel and road failures. Clearance limitations also restricted the size and arrangement of the low-pressure cylinders, while at the same time the single-expansion-cylinder superheated-steam locomotives gave opportunity for greater hauling capacity and economy.

111 The three-cylinder compound has frequently been advocated owing to the allowable reduced cylinder diameters and piston thrusts and a more uniform turning movement, but its use has been deferred owing to central main-rod and axle complications.

112 There is no doubt but that a properly designed superheated cross-compound locomotive embodies many advantageous features such as greater starting and hauling capacity per unit of weight, less evaporating surface per indicated horsepower, reduced fuel and water consumption and less boiler repairs, and that it will return to favor for freight service in combination with higher boiler pressures and superheat, due to the necessity for greater drawbar pull and horsepower and for utilizing all superheat before its final exhaust.

113 *Valve-Motion Gear.* The Stephenson valve gear, through its variable lead for different points of cut-off, gives one of the best and most flexible steam distributions for locomotives. However, its undesirable and inaccessible location between frames and driving wheels and heavy revolving and suspended reciprocating wearing parts caused the writer, in 1903, to introduce the Walschaerts valve gear, a Belgian invention, in connection with the design of the Baltimore and Ohio Railroad Mallet articulated compound No. 2400.

114 The Walschaerts gear, as well as other outside valve gears now generally used, is accessibly located outside of the frames and driving wheels and driven from both the crosshead and an eccentric crank, but they all have the disadvantage of a constant lead and of being affected by the vertical displacement of the axle.

115 By eliminating the disadvantages of the outside valve gears now in use and adding certain improvements for the purpose of increasing the ratio of expansion and shortening the ratio of compression, the tractive effort can be increased at least 10 per cent at all points of cut-off and the fuel consumption reduced 5 per cent through ability to develop the same drawbar pull with a shorter cut-off. Such a change will add greatly to the efficiency of the steam locomotive, particularly when it is recalled that the average inside or outside valve gear slightly out of adjustment represents considerable loss in hauling power and in fuel economy. Tests made show that valves out of adjustment are responsible for from 8 to 21 per cent increase in fuel consumption per ton-mile as compared with valves properly set.

116 Where compound cylinders are used a steam expansion regulator should be incorporated with the motion gear to effect the automatic independent adjustment of the cut-off for each of the high- and low-pressure cylinders for the purpose of obtaining certain cylinder ratios, and at the same time bring the cut-off in harmony

at the center of the quadrant. By this means the ratio between the high- and low-pressure cylinders, which, for example, should properly be 1 to 3 at starting, can be brought to 1 to 4 at cut-off, thereby insuring easy exit of the exhaust steam from the low-pressure cylinder and at the same time automatically distributing the work properly between the two cylinders at speed. In this way a compound locomotive of the Mallet articulated type can be made to develop at least 55 per cent of its rated tractive power at a speed of from 8 to 10 miles per hour, when operating at 25 miles per hour, and there will be a gain in tractive power of about 15 per cent at 25, and of about 10 per cent at 30 miles per hour. In fact, a drop in the drawbar pull in a Mallet articulated compound locomotive on account of speed should not materially increase beyond that of a single-expansion engine.

117 *Cylinder Clearance.* The inauguration of the use of the inside-admission piston valve and of superheated steam has brought with it the wasteful effects of larger cylinder clearance, due principally to the use of a valve of too large diameter and an indifferent design of valve chest and ports in combination with the cylinder castings.

118 To somewhat overcome this trouble the piston valves were increased in length, with subsequent breakage of castings through the vertical ports, particularly as the result of water from condensation and unstayed flat surfaces.

119 The use of smaller-diameter piston valves located close to the cylinder and connected with properly designed expanding steam ports, will, in combination with improved material and workmanship, correct these generally existing deficiencies.

120 *Cylinder Back Pressure.* About 75 per cent of the cylinder back pressure is due to the use of the exhaust steam to produce draft for combustion, evaporation and superheat. Assuming that for every 100 hp. in steam used only 60 per cent is utilized in producing actual tractive power, then 40 per cent is wasted through the exhaust, of which 75 per cent is chargeable to steam and superheat generation.

121 Much remains to be done in the way of enlarging exhaust-steam openings from the cylinder to the atmosphere and in reducing existing sharp turns, cramped passages and obstructions to the free passage of steam through them; and also in the development of an exhaust stand and nozzle that will combine the advantages of the single and double types. It has been found that by enlarging a

5½-in.-diameter exhaust nozzle to 5¾ in. or about 9.3 per cent in area, fuel consumptions have decreased from 15 to 20 per cent, depending upon fuel and weather conditions, and that the locomotive efficiency has been increased from 10 to 15 per cent, depending upon cut-off and speed.

122 *Valves and Cylinders.* Inside-admission piston valves, although inherently deficient with respect to water- and compression-relieving capacity, have many advantages, particularly for superheated steam, and the application of double-ported valves for low-pressure cylinders has worked out satisfactorily.

123 Various tests and many years' experience have demonstrated through the better use of steam and the resulting reduction of jerking, pulling and stresses on valve stem and gear, unbalanced pressure, frictional contact, valve and bushing wear, leakage, and lubrication, the practical advantages of a minimum diameter and weight of valve with the circumference no greater than the length of a slide-valve port and with every inch of bushing port made effective and designed in conformity with the well-known principles governing the flow of gases so as to eliminate eddies and baffling in the steam flow between valve and cylinder.

124 In addition to reducing the weight of a valve by reducing its diameter, it can be further lightened by using a smaller spool, as experience has proven that with simple cylinders an area of opening through the valve body equal to one-half the area of a single exhaust-nozzle orifice is sufficient to obviate the hammering of the exhaust steam on the valve ends. With cross-compound cylinders the conditions are even more favorable, due to the receiver pressure. Furthermore, there is still a possibility of considerably reducing weight in bull-ring and follower designs, which will further reduce the stresses in valve rods and gears that have been found to increase with the speed, cut-off and weight of valve.

125 There is also considerable opportunity to improve packing rings by locking and putting them in absolute steam balance, preventing exhaust rings from collapsing under compression or being forced from grooves into ports between bridges, and stopping leakage of live steam to the exhaust side of the valve.

126 Extended rods and carriers for the front ends of both valves and pistons have also been found essential to the best results.

127 Two refined-gray-iron packing rings should be sufficient for all pistons, and two-piece one-ring piston and valve-rod packing of a suitable aluminum alloy should be satisfactory.

128 Wherever possible the center line of each cylinder, under normal working condition, should be in horizontal alignment with the centers of the driving axles.

129 All cylinders should be equipped with suitable bypass valves.

130 *Piston Speeds.* Frequent errors have been made in not properly proportioning the driver-wheel diameter and stroke of the piston. Slow speed and high ratios of expansion are factors particularly favorable to superheated steam, and piston speeds of from 700 to 1000 ft. per min. will insure the best results.

131 *General.* With superheated steam too much attention cannot be given to the foregoing and other details in valve and cylinder design and material, as the waste of steam, heat and power at these points has much to do with keeping down the thermal efficiency of the locomotive as a whole.

14 WASTE-HEAT DISTRIBUTION AND UTILIZATION

132 As a reasonable estimate would show that 40 per cent of the heat in the steam and in the products of combustion is exhausted from the stack, any considerable part of this heat that can be reclaimed for preheating boiler feedwater will add greatly to the overall efficiency of the locomotive and to the saving in fuel.

133 The principal means through which to accomplish this saving, in a practical way, are exhaust-steam heaters and flue-gas economizers, both of which can be readily adapted to a modern steam locomotive.

134 *Exhaust-Steam Heaters.* With the many steam-using auxiliaries such as those for air compressing, boiler feeding, valve-gear operating and electric lighting, which operate when the locomotive is standing, drifting or working, a combination open and closed type of feedwater heater and purifier for the utilization of the exhaust steam from these auxiliaries, supplemented if necessary by steam from the main engine's exhaust, should receive prompt consideration.

135 From actual service tests of closed types of heaters, made on modern superheated-steam locomotives, using a portion of the main-engine exhaust steam only, it has been found that a feedwater temperature approximating 240 deg. fahr., or within 15 deg. of the exhaust-steam temperature, can be obtained without interfering with the draft required for maximum steam and superheat generation.

136 *Flue-Gas Economizers.* High steam pressure and high superheat unquestionably save steam, and while comparatively little heat is required merely to superheat steam — probably not more than one-third as much as to preheat boiler feedwater, nevertheless, owing to the high rate of combustion and evaporation and in the process of superheating much heat is usually wasted, as the gases from which the steam receives its heat must be hotter than the steam itself.

137 The higher the steam pressure the less is the average difference in temperature between the gases of combustion and the contents of the boiler, therefore the slower the transmission of heat the greater the work of the economizer may be. Likewise the lower the efficiency of the boiler will be if it is not supplemented by an economizer.

138 An economizer will heat the feedwater to a higher temperature than an exhaust-steam heater and will recover most of the waste heat resulting from high steam pressure and high superheat, as it is able to recover low-temperature heat that has escaped from the boiler evaporating or superheater surfaces because the average temperature of the feedwater within the economizer — which should, if practicable, be brought up to the boiler evaporating temperature — is much lower than the temperature of the water in the boiler.

139 As locomotive-smokebox superheaters, now obsolete, have demonstrated that 50 deg. of superheat may be obtained from flue gases at 600 deg. fahr. there should be no difficulty in devising a locomotive economizer that will produce very effective results, in combination with high boiler pressures, superheat and draft, without baffling the boiler draft and evaporating capacity. In fact, with an average boiler efficiency of 60 per cent and an economizer efficiency of 50 per cent the possibility of recovering from 25 to 50 per cent of the stack-gas losses and increasing the thermal efficiency of the entire unit, is within the limits of possibility.

140 *General.* Another factor favoring the use of combination exhaust-steam heaters and flue-gas economizers is the opportunity to reduce the noise and nuisance now created by the steam exhausting from the main and auxiliary engines through the muffling of the exhausts.

15 FRICTION AND RESISTANCE

141 *Friction.* The only form of useful friction to which the steam locomotive is subject is that which occurs between the driv-

ing wheels and rails for adhesive purposes. All other friction due to oscillation, concussion, rolling, wheel flanges and treads, journals, cylinders, valves, valve gear, crossheads, center and side bearings, coupler side play and the like, absorbs a considerable percentage of the power developed by the steam.

142 Maximum machine efficiency, or ratio of drawbar to indicated horsepower, is usually obtained at speeds of from 25 to 50 miles per hour and ranges from 80 to 85 per cent, above which speeds, due to increased friction, it gradually decreases to about 70 per cent at 75 miles per hour. For example, with a locomotive developing about 2000 i.hp. at a speed of 30 miles per hour, about 325 hp. would be lost in internal or machine friction.

143 During the past ten years the increased rigid wheelbase and axle loads, greater lateral rigidity, larger cylinders, valves and revolving and sliding bearings, substitution of grease for oil lubrication, and greater number of frictional parts, have tended to increase the machine friction and consequently the horsepower, drawbar pull, and steam and fuel losses, all of which are factors that should receive proper consideration in new designs.

144 *Resistance.* Other than the resistances resulting from machine friction, the locomotive is subject to those due to grades, curves, weather, wind and head air, which latter is more particularly affected by the general design.

145 As the horsepower required to overcome front air pressure increases with the cube of the speed plus the resistance due to the "air in motion," reduction of transverse flat surfaces, smoothing off of projections, vestibuling of openings and use of general curves parallel to the natural flow of the air should be carefully considered in high-speed locomotives, particularly in view of the high fuel consumption and machine friction and the relatively small proportion of drawbar pull available for hauling trains at high velocities.

146 The possibilities in this direction can best be illustrated by a statement of the relative atmospheric resistance on bodies of different shapes, as follows:

Flat, abutting surfaces	100 per cent
Cone, single, apex ahead .... about	85 per cent
Cone, double, apices ahead and back .... about	25 per cent

147 While the complicated design of a steam locomotive, particularly as regards the application of its accessories, makes the use

of relatively smooth outside surfaces generally impracticable, still much has been done along this line on some of the European railroads that can be adopted by us to good advantage.

16 ACCELERATION

148 In order that the railway engineer may have before him some data as to what is being accomplished in the way of acceleration, which factor is fairly representative of the power, speed and traction that can be developed in a vehicle, it may be well to state what is being done by other mobile power. For example, an ordinary passenger automobile equipped with twelve 3-in. by 5-in. cylinders, having a piston displacement of 424 cu. in. and a gross weight of 5300 lb., will, on a level highway, in direct drive, accelerate from a standing start to a speed of 35 miles per hour in 14 sec., and to a speed of 60 miles per hour in 26 sec.

TABLE 6 FORCE REQUIRED IN LOCOMOTIVE ACCELERATION

From rest to given speed in miles per hour	Time allowed in seconds	Average force required per ton of engine and tender to overcome inertia only, lb.
10	60	16
20	60	32
30	60	48
40	60	64
50	60	80
60	60	96
70	60	112
80	60	128

149 In a steam locomotive the effects of inertia are distributed throughout the machine, due to the variable speed and action of the locomotive as a whole as well as of its reciprocating and revolving parts, and in order to indicate the force necessary to overcome this inertia and produce a particular speed in a given time, per ton of 2000 lb. of engine and tender, exclusive of the resistance due to grade, curvature and friction, the data in Table 6 may be of interest.

150 As the train resistance increases and the drawbar pull of the locomotive decreases due to speed, acceleration rapidly becomes a diminishing quantity. Therefore in order to expedite train move-

ment locomotives should be designed and adjusted so as to permit of the highest possible rate of acceleration in the shortest distance after starting, in order that the maximum desired running speeds can be reached in the minimum of time during which the greatest evaporating capacity of the boiler is available. In locomotives designed with trailer wheels a great deal of otherwise available adhesive power, particularly for starting and acceleration purposes, is being wasted and the utilization of this lost adhesive weight by the elimination of trailer wheels, or by the application of an independent means of power for their propulsion, would accomplish a great deal in the way of starting and accelerating trains to speeds of from 15 to 20 miles per hour.

17 DECELERATION

TABLE 7 DATA SHOWING RESULTS OF VARIOUS METHODS USED IN DECELERATING STEAM LOCOMOTIVES

Method of braking	Rails not sanded		Rails sanded	
	Distance to stop, ft.	Time to stop, sec.	Distance to stop, ft.	Time to stop, sec.
Driver and tender brakes	250	11	260	11
Driver brakes	410	18	...	...
Tender brakes	540	23	...	...
Driver and tender brakes and engine reversed	275	12	160	9
No brakes and engine reversed	450	20	280	12
No brakes and engine "plugged"	520	25	265	11

151 The deceleration or retardation of a steam locomotive is now universally produced by brake shoes brought against the treads of the driving and truck wheels by various means, the ideal method being where the maximum stopping effort is at all times under the positive control of the engineer. While the reversing of the engine with or without the use of steam effects varied degrees of stopping power without the use of brake shoes, this method is generally too slow and cumbersome for present-day requirements.

152 The principal factors involved in retardation by braking are:

a Adhesion between the driver and truck wheels and the rail

b Brake-shoe retarding efficiency
c Braking-power efficiency
d Quick and uniform application of maximum braking pressure on first application, with diminishing pressure as speed decreases, to prevent wheel sliding and flattening.

153 The locomotive test data given in Table 7 show the results of various methods for producing deceleration. In these tests the engine was braked at 70 per cent of its total working weight on drivers and the tender at 100 per cent of its light weight. All tests were made on a comparative basis on level track, from a speed of 30 miles per hour to a full stop.

154 With the best existing air-brake practice a Pacific-type passenger locomotive weighing about 80 tons on 80-in.-diameter driving wheels and from 200 to 220 tons total for engine and tender in working order, will, when braked to from 110 to 90 per cent of the total weight, make stops on straight level track under good rail conditions from a speed of 60 miles per hour in distances of from 1200 to 1600 ft. and in from 25 to 30 sec., respectively, whereas from a speed of 30 miles per hour, under otherwise identical conditions, stops can be made in distances of from 275 to 325 ft. in from 12 to 13 sec., respectively. When braked at 150 per cent the stopping distance from a speed of 60 miles per hour can be reduced to about 1000 ft.

155 Deceleration is as much a factor in expediting train movement as acceleration, particularly with long and heavy trains and grades, and improved brake-shoe design, material, flexibility and bearing area in combination with clasp types of brakes for all wheels would do much toward providing greater stopping control over large and high-speed steam locomotives and thereby avoid the necessity for resorting to the use of the engine-cylinder back pressure to produce adequate braking power without liability for skidding and flattening the driving wheels.

18 LUBRICATION

156 It is false economy to restrict the quantity of lubricants used or to employ inferior lubricants to an extent that results in excessive friction, wear and tear, and any saving thus effected is many times over expended in delays, repairs and fuel. At the same time probably no locomotive supplies are handled more wastefully

or ignorantly than oil and grease, due to the lubricant not being applied or used in the proper manner.

157 Difficulty of access to the various bearings to be lubricated and the necessity for frequent hand oiling have no doubt contributed to lubrication waste and trouble, but during the past 15 years the substitution of grease for oil on many bearings, the use of larger driving wheels and lower piston speeds, and greater accessibility to certain parts as obtains from the use of outside instead of inside valve gears and journals, have brought about a substantial improvement.

158 The principal lubricants now used are: valve oil for steam exposed valves and cylinders; grease for those bearings to which it has been found adaptable, such as driving-axle and crank- and cross-head-pin journals; and machinery oil for all other parts, particularly where there is no regular revolving motion, such as guide and shoe and wedge faces, and valve-gear bearings. While graphite is used to some extent in combination with valve and engine oils and grease, its tendency to clog oil holes and grooves has limited its application.

159 Grease and oil are made in summer and winter grades to somewhat counteract the higher coefficient of friction during the cold as compared with the warm weather, which materially increases the machine friction and resistance.

160 *Valve Oil.* The usual method of feeding valve oil is through a steam-condensing lubricator. However, this method gives an irregular feed of oil to engine valves and cylinders if no change is made in the adjustment of the sight feeds when the locomotive is at rest, working with a light or a full throttle, or drifting. With high steam pressures and superheat a suitable automatic force-feed lubricator, located near the steam chests, with individual feeds to engine valves and cylinders and adjusted to insure a positive and uniform feed of 50 per cent of the oil to each of the valves and cylinders at all times when the locomotive is moving, will unquestionably give better results.

161 Piston and valve rods equipped with a suitable aluminum-zinc-lead alloy metallic packing should not require lubricators or swabs except on roads where a high percentage of drifting obtains.

162 Superheat valve oil is unnecessary, as carbonization of oil is due to air admission to engine valve chests and cylinders when their temperature is greater than the flash point of the oil used and

is also aggravated by the induction of gas and cinders through the exhaust nozzle.

163 *Grease.* While a bath of oil is the ideal method for the lubrication of heavily loaded frictional surfaces, this method has not as yet been adapted to the locomotive. In order to overcome heating troubles occurring with direct feed, waste, curled hair, swab, pad and siphon methods of oil lubrication in combination with bearings subjected to from 400 to 5000 lb. pressure per sq. in. of projected area, the use of grease has been universally established for driving-axle and crank- and crosshead-pin journals.

164 The results of tests made to determine the respective coefficients of friction of oil- and grease-lubricated journals show the former to be about 0.02 and the latter about 0.03. Therefore, while the internal or machine friction of the modern locomotive has been considerably increased due to the use of solid lubricants in combination with relatively high bearing pressures, and the wear on these frictional surfaces has been materially increased, grease has nevertheless protected bearings that would otherwise have heated and its use will no doubt be continued until a satisfactory automatic force-feed method of oil lubrication is devised.

165 *Machinery Oil.* This is the ideal lubricant for wearing parts not subjected to excessive concentrated pressures and temperatures and should be employed wherever a better distribution of the work, proportion of parts, or method of application will permit of its use.

166 Machinery oil is now generally fed to the top, side or underneath parts of the bearings to be lubricated by means of direct, needle, plunger or siphon feeds, or wool-waste packing, but there is opportunity for much to be accomplished in the development of a more satisfactory and automatic means for its application.

19 INSULATION

167 The loss of heat through radiation justifies a considerable expenditure for its prevention and the most practical method for reducing this waste is to first design and locate the heat-transmitting parts so that they will be the least exposed to the surrounding atmosphere and to then make use of a good non-conducting lagging, properly applied.

168 With the available non-corrosive heat-insulating materials that can now be readily molded into sectional blocks to any form and size desired for ready application and removal, and which will

withstand the disintegrating effects of heat, vibrations and concussions incident to modern locomotive operation, there is no good reason why boilers, fireboxes, steam pipes, valve chests, cylinders and heads, air pumps, and other heat-radiating accessories or parts should be left exposed in the way they generally are, with the resultant steam and fuel losses.

20 SAFETY APPLIANCES

169 The Constitution of the United States provides in Section VIII that Congress shall have power to regulate commerce among the several states, and it accordingly enacted a law, effective January 1, 1898, which compelled common carriers engaged in interstate traffic to equip steam locomotives with power driving-wheel brakes; appliances for operating train-brake systems; automatic couplers; and grab-iron and hand holds. Federal and state laws since enacted cover self-cleaning ashpans, electric headlights and boiler and machinery factors of safety, inspection, testing and appurtenances.

170 Prior to the enactment of these laws and of the various rules and regulations by the interstate and intrastate commissions, the majority of the railroads had established the use of similar equipment and methods as well as additional measures and devices for greater safety in operation and maintenance such as improved design and stronger materials; fewer parts; better-fitted and locked bolts and nuts; reduction in the number of studs and plugs under pressure; automatic air signals; spark arresters; speed indicators and recorders; automatic bell ringers; non-telescoping appliances; metallic steam, oil and water connections; flexible staybolts; articulated engine and tender trucks and safety locomotive-control systems. Furthermore, the practice of using two each of safety valves, injectors, feedwater connections, air pumps, water and steam gages, and similar appliances had long been the general practice.

171 While the annual reports of the Interstate Commerce Commission on personal-injury accidents chargeable to locomotive equipment indicate that considerable remains to be done to improve safety with respect to boiler fireboxes, staybolts, flues, tubes, plugs, studs, blow-off cocks, water gages and grate shakers; injectors and connections; lubricators; squirt hose; reverse gears; main and side rods, and draft gear; a great deal in this direction has been accomplished during the past seven years through the coöperation

of the railroads and the locomotive and equipment builders with the Interstate Commerce Commission inspectors.

172 Without intelligent and proper upkeep and working on the part of the human element involved, no application of safety design, material or appurtenance can be expected to function without failure; and while it is the intention that such devices will provide for more automatic operation, there will always be the necessity for constant, vigilant and trained manual control to insure against liability for man or machine failure or accident.

21 SPECIAL APPLIANCES

173 The steam locomotive, other than the boiler, engine, frame, running gear and tender proper, is largely an assembly of special appliances for firing, combustion, superheating, steam distribution and utilization, feedwater delivery, lubrication, insulation, heating, lighting, safety, and labor saving, and of devices such as trucks, axles, wheels, tires, springs, bearings, brakes, draft gear and boiler fittings.

174 These appliances and devices are the result of highly specialized research, designing and experimenting by the railway-supply and -equipment companies, who in turn are largely responsible for the development of the steam locomotive during the past fifteen years. The fact that greater progress has not been made is due to lack of substantial encouragement, assistance and coöperation from the majority of the railroads, particularly in the matter of adequate reimbursement in prices paid to cover the cost of necessary improvements in material and manufacture; to enable progressive research, experimental and development work; and to furnish the educational, investigative and expert service that they are required to perform and should perform for the railroads.

175 With the cost for locomotive fuel and repairs averaging over 75 cents per mile run, or five times what it was ten years ago, the cost for the essential design, material, construction and equipment of the locomotive must now be placed on an engineering rather than on a purchasing-price basis in conformity with other high-class machinery if more efficient and economical operation and maintenance results are to be obtained, and reference to a few particular special appliances will make this self-evident.

176 *Tender Trucks.* An investigation made by the Interstate Commerce Commission of a wreck in which 17 passengers were killed and 139 passengers and 6 employees were injured, determined

the cause as due to the derailment of the forward truck of the engine tender on account of the latter being subjected to overturning and derailing forces which were aggravated by irregularities that existed in the track.

177 The present use of staggered instead of square rail joints in track laying results in considerable vibration and surging of tenders when first-class track surface and alignment are wanting. This derailing action necessitates the use of a flexible type of tender truck, such as will make it possible for each wheel to always follow and remain on the rail with which it is in contact without regard to any other wheel in the truck, if liability to derailment is to be avoided. While it is possible to safely operate tenders having rigid trucks where excellent track conditions are at all times maintained, it is impossible to do so when the nature of the sub-grade, ballast or weather conditions will not admit of such maintenance, and therefore only the most flexible type of tender truck should be used for the purpose of insuring proper safety in operation.

178 *Truck Wheels.* According to the reports of the Interstate Commerce Commission there were 954 derailments on the steam railroads during the year 1917 that were due to broken flanges and broken and burst wheels; these caused damage to railroad property amounting to $1,132,030, and resulted in the killing of 16 and the injury of 72 persons. While these reports apply to both locomotives and cars, still they indicate the urgency for improvement.

179 With increasing wheel loads and speeds and higher and longer braking pressures the chilled-iron and cast- and forged-steel wheels must not only be of the best design, material and construction to meet the most severe requirements with a proper degree of safety, but the weights should be reduced to an economical maintenance and operating basis. Chilled-iron and forged-steel wheels have become particularly notorious with respect to non-productive deadweight resulting from unsuitable or surplus metal, or both, and necessity will now demand an early betterment.

180 *Mechanical Stokers.* There are now about 4000 locomotives equipped with mechanical stokers, of which more than one-half have been applied during the past two years to Mallet, Santa Fé and Mikado types of locomotives burning coal at firing rates of 5000 lb. and over per hour.

181 The past reports indicate that stoker-fired locomotives burn from 10 to 40 per cent more coal than those hand-fired — which

includes the additional coal used for operating the stoker equipment — and that the cost for stoker repairs ranges from 2 to 4 cents per locomotive-mile. Also that failures occur due to broken stoker parts, foreign matter in coal and wet coal. The kind and preparation of fuel are also items of importance, particularly as relating to low-volatile bituminous and anthracite coal.

182 It is doubtful whether any considerable progress in efficiency or economy will be made in the stoker firing of locomotives in combination with the limitations now imposed by burning coal on grates or in retorts with forced draft, and this is a matter of the greatest concern in the economic development of the steam locomotive.

183 *Valve-Motion Gear.* Existing valve gears are notorious for irregularities in valve movements in long and short cut-offs, even when newly applied and adjusted. Ordinary wear and tear do not improve these deficiencies and much remains to be done to produce a valve-motion gear that will, through the production and maintenance of correct valve movements, greatly increase the locomotive starting power, hauling capacity and thermal efficiency.

184 *Superheaters.* While the existing, generally used superheater has made the modern steam locomotive possible, it is far from being perfected, particularly as regards lack of uniformity in superheat temperature and difficulties from entrained moisture, leakages at element and header connections, clogging of boiler flues and superheater elements, and air leaks around outside steam pipes where they pass through the front end. However, the separation of the superheating equipment from the boiler proper, so far as practicable without introducing separately fired equipment, in combination with other refinements in design and operation will make it possible for substantial improvements to be made.

185 *Air Compressors.* Compressed air is one of the most expensive mediums for producing power, particularly when the compressing is done by the single-stage system which is still in use on the majority of locomotives. As the steam is used at long cut-off and the heat of compression is dissipated and represents lost work, an average of from 70 to 85 lb. of saturated steam at 200 lb. pressure is required per 100 cu. ft. of free air compressed to from 100 to 130 lb. pressure.

186 For air pressures of 100 lb. and over a cross-compound steam and two-stage air compressor with intercooler between the air cylinders should be used. This will easily give an equivalent

compressed-air production on from one-third to one-fourth of the steam consumption, which result can be further improved by the use of superheated steam.

187 *Main-Driving-Axle Boxes.* These are the seat of one of the serious deficiencies in the locomotive of great power. As any change in the alignment of the main driving axle or an accumulation of lost motion therein immediately affects the movement of the directly or indirectly connected main and inside rods, valves and pistons, it is most important that this axle be kept in close adjustment at all times.

188 Increasing the length of driving boxes and the various means devised for applying and adjusting the crown bearings, hub plates and shoes and wedges have not yet produced the required result and considerable opportunity for improvement still remains.

189 *Lateral-Motion Devices.* Restricting the lateral movement over leading and trailing truck and driving wheels as well as in tender trucks has been responsible for many derailments and much wheel-flange and rail resistance and wear, particularly with modern designs of locomotives of long wheelbase and high center of gravity. Promising results have obtained from the development of constant-resistance lateral-motion devices, but further improvement is needed along these same lines to meet the more extended rigid-wheelbase conditions.

190 *Throttle Valves.* These should be removed from the boiler where they are now an obstruction to making boiler inspections and are inaccessible for inspection, adjustments and repairs.

22 POWER FOR ACCESSORIES

191 The steam locomotive must not only produce superheated steam for the development of drawbar pull, but also supply saturated steam to various accessories of its own and for train operation. For example, a modern passenger locomotive is required to supply power to operate locomotive and train air brakes and signals as well as train lighting, heating and ventilating equipment, and hot- and cold-water systems; in addition it must supply steam or compressed air for the operation of the stack blower; coal pusher, conveyor and distributor for mechanical stoker; drifting throttle or valve; power reverse gear; fire door; bell ringer; track sander- ashpan doors; ashpan blowing and thawing-out devices; water scoops; injectors; so-called smoke-prevention devices; heaters for injector feedwater and delivery connections; squirt hose; sight-

feed lubricators; wheel-flange oilers; and where fuel oil is used, for heating, atomizing and injecting oil into the firebox; also the steam that is wasted through pop valves and the energy lost by radiation and condensation, all of which represent heat that does not go to the engine cylinders for the development of drawbar pull. Therefore as high as 20 per cent of the fuel as fired for an average divisional run may be used for these accessories.

192 For example, it has been found from standing and road tests made by the Air Brake Association that the cost to furnish only the compressed air to operate such auxiliaries as power reverse gears, fire doors, bell ringers, sanders, water scoops and cylinder cocks, exclusive of coal pushers and ashpan doors, is from $200 to $600 per locomotive per annum, based on six hours' service of each locomotive per day. In arriving at these figures it was found that from 25 to 75 lb. of saturated steam at 200 lb. pressure was consumed, depending upon the type of compressor and the main reservoir pressure, to compress 100 cu. ft. of free air, and the cost for fuel was figured at $2 per ton on the tender. The figures also exclude such factors as the cost of handling coal on engines, water, and depreciation of boiler and compressor.

193 Not only has the use of compressed air been found to be most expensive for the working of these accessories, but the reserve supply for train braking has been frequently drawn upon for their operation. As power reverse gears, fire doors, water scoops, coal pushers, ashpan doors and like devices can be equipped for steam operation, such substitution offers possibilities for less drain on the boiler and much needed economy in the cost for this auxiliary power production. Moreover, as all of this power for accessories is produced by saturated steam, some means for substituting the use of superheated steam for those purposes where it is more suitable and economical should be given due consideration.

23 TIME SAVING

194 The principal time-saving factors other than speed reductions and stops necessary to take on and set off business and to [illegible] train-dispatching and operating requirements, may [illegible]

[illegible]

[illegible]

[illegible]

d Mechanical terminal delays
e Fueling
f Watering.

195 *Acceleration and Deceleration.* These factors have already been considered in preceding paragraphs, and it is only necessary to add that much time is to be gained in quickening the starting and stopping of locomotives. Any engineer who has noted the length of time usually taken to get a passenger, freight or switching locomotive, either light or loaded, under headway and to reduce the speed for a stop, will appreciate what this may amount to.

196 *Mechanical Road Delays.* These may be classed as due to engine, fuel, water and man causes.

197 With the adaptation of locomotives best suited for regional requirements and with proper improvements in design, material, construction, inspection, testing and upkeep, "engine causes" can practically be eliminated.

198 Through the installation of modern fuel-preparing facilities, provision for adequate tender capacity, adaptation of locomotives to utilize the most inferior and cheapest fuels available, use of simplified manual means of firing, and particularly by reducing the consumption required per boiler horsepower developed, the "fuel causes" can be substantially reduced.

199 The proper systems and time for washing out boilers and the supplying of suitable, treated if necessary, boiler water to adequate tender tanks will dispose of "water causes."

200 "Man causes" can best be avoided through the employment of competent men, the inauguration of proper systems for education and instruction, and by equipping locomotives so that they will require the least amount of arduous work in order that the engineer and fireman can devote their time to the observation of train rules and signals and to the proper regulation of the time-, fuel-, steam-, water- and labor-saving mechanical appliances that should be efficiently and economically utilized at all times during operation.

201 *Mechanical Terminal Delays.* These are due principally to sanding, ashpan and fire cleaning, fire building, boiler washing, firebox, flue and smokebox cleaning, inspection, testing, machinery cleaning and repairs.

202 Of these delays those due to ashpan, fire, firebox, flue and smokebox cleaning are the most prolonged and non-productive and can be reduced only by improved methods of firing, re-

duced fuel consumption per unit of work performed, and substitution of mechanical appliances for arduous labor so that upon arrival at terminals locomotives can be run directly into the enginehouse instead of being held outside for this class of work and delaying upkeep attention.

203 *Fueling.* Many facilities for fueling locomotives, either with coal or oil, are obsolete, inadequate and uneconomical. Fuel should be prepared ready for firing before being placed on tenders, and with modern facilities practically no time should be lost in supplying, either on the road or at terminals.

204 *Watering.* This operation is usually performed at the same time that fuel is taken, or otherwise water is supplied from track troughs when running, or from track cranes when making station or train-despatching stops or at terminals, and should involve little if any delay.

24 FUEL SAVING

205 The problem of locomotive fuel saving has never received more intelligent thought and attention from a supervising standpoint than during the past two years. This has been due to the war-time necessity for the conservation of both the fuel and the labor required for its production and to the fuel cost reflecting a constantly increasing percentage of the total expense for railroad operation.

206 While the furnishing of coal or oil of a proper kind and preparation by an intelligent, trained and careful fireman to a locomotive in good working order and properly operated should result in effective and economical performance, the vast difference in the amount of fuel actually used by different train despatchers, engineers, firemen and locomotives to produce the same ton-mile movement under like transportation conditions indicates the necessity for reducing the amount of fuel to be fired per ton-mile by effective mechanical means and methods instead of depending upon the directly involved and responsible human element for equivalent results.

207 There is no questioning the fact that avoidable low boiler and mean-effective cylinder pressures, saturated steam, indifferent boiler circulation, excessive firebox draft, clogged grates and boiler and superheater tubes, forced combustion, high smokebox temperatures, unnecessary non-adhesive weight and generally indifferent

steam generation, distribution and utilization factors, for which the engineer and fireman are not responsible, have more to do with high fuel and water rates than those factors within their control. Therefore the proper procedure, particularly in view of the relatively small increase in cost for the improved locomotive equipment as compared with the otherwise total locomotive cost and the annually reduced expense for its upkeep and operation, is to design and equip the modern steam locomotive so that it will, through its self-contained mechanical operation, more fully utilize the thermal heat value of the fuel fired and thereby not be so dependent upon manual control to bring the fuel used for equivalent productive work within the proper limitations.

208 Making initial capital and continual upkeep and operating expenditures in order to provide well-known inefficient and uneconomical mechanical means for handling, firing and wasting greater quantities of fuel than are within the easy range of one-man hand firing, in preference to diverting an equivalent amount of money for capacity-increasing and fuel- and water-saving appliances, represents a policy that is not at all consistent with existing and future labor and fuel costs if the railroads are to be continued on an investment basis.

25 LABOR SAVING

209 The labor now required for the upkeep, terminal handling and operation of the steam locomotive is divided into three classes, i.e., shop and enginehouse men; hostlers, cleaners and supply men; and enginemen.

210 The item of maintenance is distributed between general and running inspection, testing and repairs and is taken care of at the shops and enginehouses, respectively. During the past fifteen years a great deal of attention has been given in the planning of these facilities to provide labor-saving tools and machinery for dismantling, repairing and assembling locomotives and appurtenances, and there are today many conspicuous examples of modern railroad shops and enginehouses, even though many more are needed on railroads that have not given proper consideration to this important factor in their operation.

211 Great progress has been made in the establishing of adequate and suitable terminal handling, cleaning and supplying facilities which now include power-operated coal-, sand- and ash-

handling plants and turntables, high-capacity water cranes, hot-water boiler-washing and locomotive-cleaning systems, steam and compressed-air stack and flue blowers and similar appliances. The cleaning and dumping of fires, ashpans and front ends and the rebuilding of fires is, with the increasing size of locomotives and the use of inferior coal, becoming a matter of great concern, delay and expense in the terminal handling, particularly during congested traffic and cold-weather periods and a satisfactory solution of this problem still remains to be provided.

212 In the operation of locomotives the Hours of Service Law as enacted by Congress on March 4, 1907, established the general practice of pooling locomotives and crews, which system until that time had been adopted by only a few of the railroads. The divorcing of the engineers and firemen from regularly assigned locomotives, in combination with the increasing size of the latter, resulted in relieving the enginemen of work which was transferred to the enginehouse forces, such as: detailed inspection; adjustment of driving-box wedges and main- and side-rod brasses; repacking boiler-head fittings and journal cellars; filling grease cups; cleaning head, cab and marker lamps and various other equipment and parts; filling lubricators; looking after tools and supplies; hostlering at terminals; and similar detailed attention which in combination with the more extended use of power-operated brakes, reverse gears, ashpans, grates, stokers, coal pushers, water scoops, fire doors, bell ringers, cylinder cocks and like devices have practically eliminated arduous manual operation on steam locomotives of great power.

213 The mechanical requirements and status of the engineer and fireman on the large steam locomotive having been substantially changed through relief from long hours on the road, work at terminals and by means of these labor-saving devices on the road, there should now be a resulting higher standard of operation, efficiency and economy.

DISCUSSION

F. J. Cole (written). Notwithstanding the great improvements which have taken place in the last twenty years in locomotive design, much, as the author states, still remains to be done.

That the modern locomotive is a fairly efficient machine in spite of the fact that it is a complete power plant in a very small space, mounted on wheels, is shown by the fact that there are numerous instances where an indicated horsepower is produced with less than 2 lb. of coal per hour. Of course, under road conditions these extremes of economy usually cannot be obtained, because of the fluctuating demands for power, which necessitate working the locomotive a considerable part of the time under conditions which are not the most economical. The present locomotive boiler is not well adapted for pressures of 350 lb. and over, and before these high pressures can be used safely and economically for locomotives, much work must be done on designs of boilers suitable for these high pressures.

Many of the suggestions made by the author for the improvement of the locomotive have been under consideration for some time and many efforts have been made towards their realization. Item *h* of Par. 6, "lighter and properly balanced reciprocating and revolving parts," implies that much remains to be done in order properly to balance these parts. As a matter of fact, the methods now in use are satisfactory within their limitations, and since the excess weights introduced for the purpose of smoothing out the disturbances produced by reciprocating parts must produce increase or decrease of rail pressure, it necessarily follows that the goal of efforts in this direction is to make these parts as light as possible. It is well known that the hard riding of a locomotive does not always arise from improper counterbalance; it may be due to other causes, such as, worn driving box bearings, etc.

It is not always desirable to dispense with the use of trailing wheels because the boiler design can often be very much improved by their use, and it may be more desirable to have a free steaming engine than to sacrifice good qualities in an effort to utilize a greater percentage of adhesive weight. In calculating tractive power it is convenient to use 85 per cent of the boiler pressure and no useful purpose would be served by taking a figure as high as 92 per cent, although under the most favorable circumstances more than 85 per cent can be obtained in practice.

When more than four driving axles are used, some attention should be paid to the curves the locomotive has to negotiate, so as to reduce flange wear. Improvements can be made by the use of lateral motion driving boxes or conjugated leading trucks. These trucks control the lateral motion of the first driver.

Smokebox temperatures depend largely upon the intensity of draft and the length and diameter of flues. Modern long-tube locomotive boilers have very much lower smokebox temperatures than older engines, but this is not all gain because of the greater draft energy required when the ratio of length to area is excessive. It is doubtful whether increasing the water spaces in the average locomotive boiler would very materially increase its steam producing qualities. It is customary to talk of increasing contracted spaces in a more or less vague way, but much attention has been paid to this matter for many years, so it is not probable that very great improvements can be made in this direction.

The quality of steam has not been improved by the low domes necessitated by clearance limitations, which have been used on large modern locomotives. In some cases throttle valves have not been designed so that the steam is taken from the highest part of the dome. Therefore, some consideration should be given to these details, so that water is not carried through the throttle into the steam pipes. Tests made some years ago on locomotives with nominal height of dome, in especially hard service, showed steam of 98.5 per cent dryness, or a moisture content of 1.5 per cent and less. Putting the throttle valve outside the boiler does not seem to be the remedy for wet steam, except for special types of boilers.

Enlarging exhaust nozzles, consistent with the proper steaming qualities of boilers and properly designed cylinders without sharp turns, scant passages or obstructions, are to be commended. These matters have received considerable attention, therefore, no considerable gain can be effected by merely enlarging exhaust passages, since permitting the steam to expand unnecessarily merely causes it to lose some of its velocity.

There are many devices which have been applied to locomotives which show considerable economy, and many experiments with feed water heaters, economizers, etc., have been made. There is, however, a balance, which should be preserved between simplicity and complication, because adding devices designed for economy may result in increased maintenance charges, etc., which must be carefully considered, especially in relation to the cost of labor and the difficult labor problems of to-day. The increasing cost of coal, however, seems to warrant the use of many devices which were not considered economically desirable a few years ago.

The paper contains much excellent matter, which will repay careful reading.

HENRY B. OATLEY (written). The author has presented an extremely interesting paper and it is, in the writer's opinion, a complete résumé of improvements which have been suggested, developed and tried, for the most part, during the past generation. The advantages of very nearly all of these suggested improvements would probably not be questioned were their consideration to be based solely upon thermodynamics or the mechanical improvement of the locomotive.

The paper has been read with care because the topic discussed is of very great interest, and the writer has given considerable time and thought to quite a number of the improvements which have been suggested. In many of the progressive steps advocated by the author, the writer is in hearty accord, and particularly is this true of his advocacy of increasing the boiler pressure. At the Worcester meeting in June, 1918, many of these same points were brought forth and advocated. The writer at that time, as well as now, visualized a locomotive having somewhat different fundamentals of construction from those upon which the author has constructed a picture of the scientifically developed steam locomotive.

The author clearly indicates in his paper that the boiler on his 350-lb. pressure locomotive will have all of the main characteristics of the present form of locomotive boiler involving a relatively large shell of heavy plate. All through the paper there are references to indicate that no other general form of boiler has been considered. To the writer this seems, to say the least, an error of omission. It is difficult to believe that, with any appreciable increase in boiler pressure, the present form of boiler will prove satisfactory, either from an engineering or from a financial standpoint.

The expectation is fully warranted that the 350-lb.-pressure locomotive contemplated by the author will be successful. Its advent, however, will be but one step towards still higher boiler pressures. It is the writer's belief that in the near future pressures of 500 lb. per sq. in. and upwards, with suitably designed boilers, will be as safely and economically used as 350 lb. pressure in essentially the usual form of boiler suggested by the author of the paper.

So also is it apparent that the author does not visualize any considerable change in the cylinder arrangement nor in the method of transmitting power through two, three or four piston rods, main rods and main crank pins. The writer is firmly convinced that the present method of transmitting power will be superseded by a type of construction which will utilize a smaller amount of material and be more easily maintained and replaced; and by means of such construction there will be possible the elimination of considerable unbalanced reciprocating and rotating weight, resulting in less rapid deterioration of rail, roadbed and driving mechanism.

Another point which the author has apparently not considered is the marked increase in economy made possible through the utilization of condensing operation. It may not be feasible to think of condensing operation in the terms of high vacua now carried in marine and stationary installations, but the considerable reduction in the back pressure and in the utilization of the heat at the lower range of pressures is entirely reasonable, and has been accomplished in steam driven motor vehicles, both in this country and abroad. It is believed that this will be one of the features incorporated in the developed locomotive of the near future.

In Par. 193 reference is made to the use of superheated steam by many of the auxiliaries referred to in Par. 191. This is a feature that has already been provided for and has been under consideration by many railroad men for some time. Much may be expected from this development.

At various points in Par. 89, and following, a number of suggestions are made for changing the design, type and location of the superheater, as well as its allied parts. Some of these suggestions are undoubtedly worthy of careful consideration. However, in the N. Y. Railroad Club *Proceedings* for 1907, p. 781, the author urged slide valves, smoke-box superheaters and low degrees of superheat. In the present paper the scientifically developed locomotive is specified as having small-diameter piston valves (see Par. 119), 300 deg. of superheat (see Par. 6), and in Par. 184 provision is made for "the separation of the superheating equipment from the boiler proper," qualified by a clause barring a separately-fired superheater.

In view of the world-wide use to-day of superheaters which do not meet either the author's views of 1907 nor of 1919, are we not justified in going slowly in the matter of radical changes in the design of these parts?

Reference is made, in Par. 87, to the fact that the efficiency increased progressively "and in proportion to the amount of superheat up to the point at which the exhaust steam begins to show superheat," the inference being that above this point the increase in efficiency either stops or continues at a decreased rate. In Par. 89, 90 and 92, and at various other points, arguments are advanced for still further increasing the superheat above the usual present practice, and in Par. 90 the inference is made that superheat in the exhaust is not necessarily a sign of decreased efficiency. The reader is left with mixed feelings and is uncertain as to whether the author does or does not advocate increasing the degrees of superheat.

In Par. 94 references are made to particular troubles due to the boiler and superheater equipment generally used. It is felt that the paper lays emphasis greatly in excess of that warranted by the conditions existing on the American railroads, and probably on railroads of other countries. For example: It is claimed that extreme losses in steam pressure between boiler and steam chests exist to-day. It is believed that actual conditions demonstrate that, at any given output, locomotive operation, pressure loss has been decreased during the past ten years. Instances of large pressure losses may be found in modern locomotives, but these are when operating at capacities from 10 to 40 per cent higher than was possible before the introduction of superheated steam.

It would be quite possible to give many instances of the actual application of dozens of the details suggested in the paper, but as the time is limited only one instance will be cited.

In Par. 99, 101 and 105 is advocated the removal of the saturated steam delivery pipe, the superheater header and also the steam delivery pipes from the superheater to a point outside the boiler and smoke-box. In 1912 several engines for the Atchison, Topeka and Santa Fé Railway were equipped in a manner to meet these specifications. The net advantages of these features apparently were not pronounced as they have not since appeared in other locomotives, save in some few special types where any other construction would be obviously awkward and expensive.

The fact that many of the suggested improvements have not been perpetuated may perhaps be chargeable to faulty detail design, or possibly to a realization that the gains from such improvements did not provide a definite and adequate return on the investment. The writer believes that any improvements and betterments contemplated will stand or fall, depending upon whether or not they measure up to such a standard.

George Gibbs (written). The subject of the paper is of much importance and interest, but it is not clear how the electric locomotive is involved and it is regrettable that an inadequate reference to the latter should have been attempted in this paper. Electric traction is too big a subject to dispose of properly in a few paragraphs, as the author has done, and broad generalizations are always dangerous. The author's generalizations will, the writer believes, not bear vary with the particular installation, but there is no special difficulty in figuring results, economical or otherwise, with a fair degree of accuracy, provided the conditions are known. If one does not know all conditions it is, of course, dangerous to undertake interpretations.

As a part of Par. 16 "numerous objections" to electrification are listed. Presumably they also apply to "complete electrification of some portions of the large trans-continental trunk lines"; but because of the indefiniteness of the statement and because it is evident the author cannot be drawing all his conclusions from one example (the St. Paul Railway), as no statements have been made public regarding this one case, we must assume that he is talking about trunk line conditions generally. If this is so, the writer believes that the ordinary railway man or engineer who reads the paragraph in question would come to the conclusion that no railway electrification could be undertaken with any reasonable regard to either costs or reliability of service. The objections are of the most sweeping character, thus, *a*, *d* and *f* relate to the question of first or operating cost and the words "prohibitive" and "enormous" occur. Any one of such objections would be sufficient to kill any project.

What is meant by "prohibitive non-productive cost" in item *a*? All costs involved in an electric traction installation should be made for the purpose of producing a useful purpose; they are all figured in the net result, and this is what the railway man is after. Some of the objections (*b* and *d*), it would seem, must also apply to steam traction; i.e. different types of locomotives are developed for different services, and spare equipment must be provided for varying business conditions. These factors, of course, are also taken into account in determining the advisability and usefulness of electric traction.

As to reliability (items *c* and *g*), I know of no electric installation which has not proven more reliable than that of the displaced steam. It will be remembered that two years ago, in 1917, the winter was especially severe in the Eastern states. In the spring I happened to ask a high official of a large railway system how his electric installation (which is operating on a division) had gone through the winter. He said, "Fine; it is the only thing on the road that has run this winter. I wish we had no steam locomotives on the road." Of course this is an exaggerated statement, but it reflected the frame of mind of an official who had been through trouble. It is absurd to say, in the face of sufficient experience, that electric operation with all its complication and interdependent links is not more reliable than that of steam locomotive operation under almost any conditions.

If the author should contend that the electric locomotive is not yet at the end of its development stage, the writer would heartily endorse such statement. After 100 years of experience the steam locomotive is still in the process of evolution, as witnessed by the facts brought out in the author's paper. The writer considers it entirely creditable to engineers that they have been able to develop electric locomotives within the short period of, say, 10 years' time, which are reliable in service, have a moderate up-keep cost and which exceed in hauling capacity the most powerful steam locomotives.

What has been said is simply to discourage sweeping statements. If a comparison is to be drawn between steam and electric traction as a whole, it should be done with the same care and in the same detail for both.

Wm. H. Wood, in a written communication, asked the following questions:

1 How does the cost of upkeep of the locomotive superheater compare with the estimated gain in fuel economy?
2 How great is the weakness caused by the flexible stays in locomotive fireboxes?
3 What would be the increase in economy if the use of flexible firebox and combustion chamber were to increase the heating surface by 20 per cent?

Clement F. Street. In Par. 16 the author advances several objections to the electrification of trunk lines, among which is "First cost from five to ten times and operating cost from two to three times that of steam." This is a general statement, unsupported by figures, which is open to question.

Regarding rule-of-thumb methods, I am reminded of a little incident which occurred when our friend M. N. Forney was quite active in the Merchant Mechanics' Association. They were discussing the proportions for a locomotive boiler at one of their conventions at Saratoga when Mr. Forney said, "I have figured out the proper proportions of a locomotive boiler from every possible standpoint from which it can be figured and I have finally come to the conclusion that the only practical rule to follow is to make the boiler just as big as you can"; this is just what is being done today. Far be it from me to depreciate scientific methods, but when it comes to designing locomotive boilers, figures are forgotten and they are made as big as possible. The same is also true regarding locomotive frames.

I must take emphatic exception to Par. 174 in which Mr. Muhlfeld says, "The fact that greater progress has not been made is due to lack of substantial encouragement, assistance and coöperation from the majority of the railroads, particularly in the matter of adequate reimbursement in prices paid to cover the cost of necessary improvements in material and manufacture, to enable progressive research," etc. I have done considerable experimental work on railroads and never yet have I failed to have the most hearty coöperation of the railroad officials with whom I was working. They have backed me up in ways which I never would have expected them to do, and the coöperation, instead of being lacking has been more than any reasonable man could expect.

In Par. 181 the author says: "The past reports indicate that stoker-fired locomotives burn from 10 to 30 per cent more coal than those hand-fired." These figures are too low. A stoker-fired locomotive will burn, I would say, from 25 to 75 per cent more coal than it is possible to put in by hand and the locomotive will perform a correspondingly greater amount of work. The average fireman will burn 30 to 35 per cent more coal than the expert. The expert with ideal conditions gets all the heat units there are out of such an amount of coal as he can shovel. The locomotive stoker when properly handled and with ideal conditions will do the same as the expert, and in addition handle as much coal as any locomotive can burn.

He said he would be inclined to increase the figures given by the author in Par. 181 to an excess of coal fired by the stoker of 25 to 75 per cent over hand firing losses.

G. L. Fowler said that it had been found that trucks on locomotives were necessary, and he assumed that the author could not deny that the purpose of such trucks was to adjust the stresses on rails. Experience had demonstrated that a symmetrical wheel base was harder on the rails than an unsymmetrical one.

He doubted if the speed of 125 feet per minute, which the author had stated as the minimum for circulation within the boiler, could be maintained without artificial means. Although there was much agitation, he said, the circulation in the waterleg of the boiler was very slow. He could see no advantage in rapid circulation in a boiler either from a standpoint of heat transmission or lessening of internal stresses.

OTTO S. BEYER, JR., concurred with the author's statement in Par. 11 that electrification would not reduce the net cost below that possible in steam locomotive practice, and called attention to the possibilities of super-power stations located at coal sources.

In addition to the five items of Par. 21 for consideration in the design of a new locomotive, he would add greater speed in operation, which, he said, would affect those variable cost items of labor and fuel.

While it might be possible to load a locomotive to 92 per cent of its hauling capacity, as suggested in Par. 31, he thought that it was also important to consider the time of hauling as well as the tonnage hauled.

Concerning the coöperation of railroad officials he agreed with the author that this had been lacking in the past. He further pointed out that suggestions for improvements in locomotive design and operation have in the past come from sources entirely too limited when the great number of employees engaged in railway operation are considered. The trouble in his estimation lay in the fact that the management had not as yet devised satisfactory methods for releasing the latent resourcefulness of the great numbers of railway employees who are not specifically delegated to the task of designing and inventing. He felt that the ideal of service was not sufficiently emphasized in the methods which had been devised for operating our transportation systems. Not until this is the case will the rate of improvements in locomotive designing and operation increase.

EDWIN B. KATTE questioned the validity of the objections to electrification raised by the author in Par. 16. The fact that many of the most important railroads in the country are now operating a portion of their systems electrically, and, practically every other large railroad is considering electrifying some part of its property in the near future, would indicate that the cost was not prohibitive. So far as the non-interchangeability of equipment was concerned, he pointed out that steam locomotives were not often transformed or sold to other companies and that the question of System of electrification was not now an important item. Electrified steam railroads are not usually dependent upon a single power station as the author would have one believe, most railroads feeding the transmission lines of their systems from several power stations, and further, the power station is about the most reliable thing about a railroad. It is generally accepted that there is little complication of road and terminal tracks due to the working conditions and from 25 to 80 per cent increase in capacity when busy terminals or main lines are electrified. The figures for first and operating costs he considered high. The first cost is usually more for electric than for steam roads, but Mr. Katte had never heard of costs of ten to one. The operating costs, on the other hand, were invariably lower. It has been the experience of most roads, that the electric division operated more nearly 100 per cent of normal during snowstorms and blizzards than the steam divisions. Mr. Katte regretted that he was not permitted by the five minute rule to further discuss the subject.

James H. S. Bates, who asked about the electrification of the C. M. & St. P. R. R., was referred by the author to a report to the American Railway Engineering Association published in 1919. In answer to a question about the application of the steam turbine to locomotives, the author replied that such application had been considered but had never been put into practice.

No. 1732

OIL PIPE LINES

By S. A. Sulentic, Eldorado, Kan.
Member of the Society

In this paper the author compares the cost of pipe-line transportation with that of rail and canal transportation and shows the advantages of the oil engine as a means of economically transporting oil for long distances through pipes. He also derives simple formulæ which make it comparatively easy to make rapid calculations of the pressure, net horsepower and brake horsepower necessary for the transmission of any quantity of oil per day through a pipe of known diameter.

THE object of this paper is (*a*) to show the advantages of the oil engine as a means of transporting oil for long distances through pipe lines, (*b*) to compare the cost of pipe-line transportation with that of rail and canal transportation, and (*c*) to derive simple formulæ that will make it comparatively easy for any one to make quick calculations of the pressure, net horsepower and brake horsepower necessary for the transmission of any quantity of oil per day through a pipe of known diameter.

2 The cheapest method of overland transportation of oil is by pipe line. The cost in comparison to rail transportation is low, and yet when the rail cost is based on every-day practices, the amount seems very small. For instance, the management of an eastern railway, wishing to impress upon its clerical employees the importance of economy, posted the following notice: "For every lead pencil you waste we have to haul one ton of freight one mile."

3 The cost of a pencil has always been regarded as being insignificant, but when it is considered that it is equivalent to the cost of the above-mentioned haul, a similar comparison with pipe-line transportation should be interesting. The carrying of one ton of freight for one mile at the cost of a lead pencil is very cheap transportation. One or two cents per ton-mile is a low rate. Canal transportation, after allowing for the proper fixed and maintenance charges, may be lower than the rail charges by sixty per cent or more. Five or six cents per mile is not an uncommon charge.

Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers.

4 In order to make a very crude estimate of the cost of transporting oil by pipe line when using equipment of the highest economy, assume a single line operating under the following conditions at a load factor of 80 per cent for 300 days per year:

Size of line	8 in.
Length of line	33 miles
Pressure in line	700 lb. per sq. in.
Rate of discharge	900 bbl. per hr.

At this rate the discharge would be 21,600 bbl. per day or 6,480,000 bbl. per year of 300 days. Assuming 6.5 bbl. per ton, the yearly discharge would approximate 1,000,000 tons. The work equivalent of this discharge would be 33,000,000 ton-miles, calling for the continuous expenditure of 257 hp. as shown in Par. 13. Assuming the mechanical efficiency of the engine to be 75 per cent, the actual horsepower necessary to install would be 257/0.75 or 342.

5 The assumed costs would be as follows:

Line: 33 miles at $1.65 per ft	$287,500
Right of way at $0.25 per rod	2,640
Freight: 79 cars at $250	19,750
Haulage: 900 tons at $14.50	13,050
Laying pipe at $0.075 per ft	13,060
Burying pipe at $0.20 per ft	34,850
Engines, pumps, installed accessories	68,500
Pump stations, buildings and foundations	30,000
Tanks: Two 55,000-bbl. at $18,500 each	37,000
Tanks: Two 500-bbl. at $500 each	1,000
Telegraph line: 33 miles at $550	18,150
Superintendence	2,500
Incidentals	6,000
Total assumed costs	$534,000

6 The operating expense, including fixed charges based on the total assumed costs, would be as follows:

Interest at 6 per cent	$32,040
Depreciation at 5 per cent	26,700
Administration	10,000
Attendance at pump stations and lines	11,500
Repairs to equipment, lines, etc.	4,000
Fuel for pumping — 3000 bbl. at $2.65	7,950
Total operating expense	$92,190

Hence the cost of operation per ton-mile under the assumed conditions would be:

$$\frac{92{,}190}{33{,}000{,}000} = \$0.0028$$

or two and eight-tenths mills per ton-mile. As the relation between the cost of pipe-line transportation and rail transportation is in the ratio of 1 to 10, it is easily seen that the waste of a lead pencil in a pipe-line enterprise would be the equivalent of the cost of transporting 10 tons of oil one mile.

7 It should be noted, however, that almost all of the pipe-line costs are fixed and are mainly independent of the amount of oil pumped. As a result the transportation cost per ton-mile will vary almost inversely with the load factor of the line. If this hypothetical pipe line should be operated only one-tenth of the time assumed, the unit transportation cost would equal the rail cost. Furthermore, these figures are based on a life of 20 years (5 per cent amortization). A railroad would probably be used for various classes of freight as long as it existed, but a pipe line is of service only as long as oil is presented for transportation. If the pipe line in question were to become obsolete in ten years through the exhaustion of the oil fields or other causes, the ton-mile cost would be greatly increased.

8 These figures have not been presented as those of an average case but only offered as "food for thought." In further consideration of the subject, however, it will not be out of place to give a few figures and examples showing the relations existing between pressure, capacity, diameter, length of line and power required.

9 Disregarding viscosity, the general hydraulic formula for friction head in a pipe discharging an uniform volume is

$$F = k\frac{v^2L}{2gD} \qquad [1]$$

in which

F = friction head in feet of water = lb. per sq. in. $\div$ 0.433
k = friction coefficient for 38 gravity oil = 0.024
v = velocity of flow, ft. per sec.
G = acceleration of gravity = 32.2 ft. per sec.^{2}
L = length of line, ft.
D = diameter of line, ft.

The formula for pressure in the line may be stated as

$$P = 0.433\,k\frac{v^2L}{2gD} \qquad [2]$$

in which P = pressure in line in lb. per sq. in.

10 The discharge Q of the line, cu. ft. per sec. can be easily derived and stated as

$$Q = \frac{\pi D^2 v}{4} \qquad [3]$$

in which Q varies directly as v. Since P varies directly as v^2 in Formula [2] and Q varies directly as v in Formula [3], it follows that P varies directly as Q^2.

11 The net horsepower required for a pipe line may be most readily calculated if we note that the pressure per square foot is equal to the number of foot-pounds required to displace 1 cu. ft. of oil, or

$$\text{Hp.} = \frac{144\ PQ}{550} \qquad [4]$$

12 For 1000 bbl. per day against a pressure of 1000 lb. per sq. in., the net hp. necessary by Formula [4] would be

$$\text{Hp.} = \frac{144 \times 1000}{550} \times \frac{1000 \times 5.61}{24 \times 60 \times 60} = 17$$

where 5.61 = cu. ft. per bbl.

13 Assuming a pump efficiency of 85 per cent, the horsepower of the engine would be 17/0.85 or 20. Here we have a very simple rule for calculating the horsepower from bbl. per day and pressure in lb. per sq. in.: namely,

Net hp. = 17 × thousands of bbl. per day × $P/1000$

B.hp. = 20 × thousands of bbl. per day × $P/1000$

14 For example, for a discharge of 8000 bbl. per day against a pressure of 600 lb. per sq. in., the brake horsepower required would be 20 × 8 × 0.6, or 96. Since the horsepower varies directly as PQ and P varies directly as Q^2, it follows that hp. varies directly as Q^3. To sum up:

Velocity varies directly as the line discharge
Pressure varies directly as the square of the line discharge
Horsepower varies directly as the cube of the line discharge.

15 Another relation of value in making rapid pipe-line calculations is the ratio between the diameter D and length L with the discharge Q and the pressure P remaining constant.

16 From the friction formula [2] it will be seen that P varies directly as v^2L/D. Assuming P constant, L varies directly as D/v^2. For the same discharge, Formula [3], that is, with Q constant, v varies directly as $1/D^2$. Therefore v^2 would vary directly as

$1/D^4$. Substituting this value for v^2 above, it is seen that L varies directly as $D/(1/D^4)$, or in other words, L varies directly as D^5.

17 This means that for the same discharge and pressure, i.e., the same friction, the length of the line would vary directly as the fifth power of the diameter. Taking a diameter of 6 in. as unity, the fifth powers of several different diameters are as follows:

D =	6	8	10	12
D^5 =	1.0	4.2	12.8	32

18 An example will show how easily these relations are utilized. Let a proposed pipe line be made up as follows:

Two 6-in. lines, 8 miles long (Y-branch)
One 6-in. line, 6 miles long
One 8-in. line, 10 miles long
One 12-in. line, 12 miles long

With the particular oil to be transported the pressure per mile when pumping 10,000 bbl. a day through a 6-in. line will be assumed as 20 lb. per mile. What, now, will be the pressure for 17,000 bbl. per day through above line and what will be the net hp. and b.hp. required?

19 Since P varies as Q^2, the pressure for 17,000 bbl. will be $\left[\frac{17,000}{10,000}\right]^2$ or 2.89 times the pressure for 10,000 bbl. The pressure in the line would then be 2.89×20 or 57.8 lb. per mile.

20 Inasmuch as the velocity is halved when using two 6-in. lines for the 8-mile section, each line carrying an equal amount of oil, the equivalent length of 6-in. line for each branch would be only two miles, the length varying directly as the square of the velocity. The length of 6-in. line equivalent to the 8-in. pipe would be 10 miles divided by the ratio of the fifth powers of the diameters 6 in. and 8 in. or $10/4.2 = 2.38$ miles. In a like manner the length of 6-in. line equivalent to the 12 miles of 12-in. pipe would be 12/32 or 0.37 mile. In other words, the equivalent length of 6-in. pipe in the line under consideration would total 12.75 miles. Under a pressure on the line of 57.8 lb. per mile, the total pressure on the line would be 12.75×57.8 or 737 lb.

21 Making use of the simple formula for hp. already derived we have

$$\text{Net Horsepower} = 17 \times \frac{17,000}{1000} \times \frac{737}{1000} = 213$$

$$\text{Brake Horsepower} = 20 \times \frac{17,000}{1000} \times \frac{737}{1000} = 250$$

22 No doubt many other useful relations could be derived for the purpose of expediting pipe-line calculations, but those which have been presented are among the most useful.

23 In illustration of the foregoing the following data in regard to the 36-mile, 8-in. Alton pipe line operating between Carlton and Wood River, Mo., should prove of interest. This line, constructed in 1913, has four stations, in each of which are installed four units each consisting of a 100-hp. type F. H. De La Vergne oil engine direct-connected to a 6-in. by 18-in. National Transit Co. herring-bone-geared power pump with 8-in. suction and 6-in. discharge. The performance of one station equipment (three units) is given below.

Oil pumped during 10 days, bbl.	140,000
Oil pumped per day, average, bbl.	14,000
Pressure maintained in line, lb. per sq. in.	700
Brake horsepower, average	196
Pump efficiency, estimated, per cent	85
Fuel consumed by engines during 10 days, bbl.	65.8
Fuel consumed by engines per day, lb.	2020
Brake-horsepower-hours per day = 196 × 24	4704
Fuel consumption per b.hp-hr., lb.	0.43
Ft-lb. of work per day developed by the engines = 196 × 33,000 × 24 × 60	9,320,000,000
Ft-lb. of work per day in oil pumped = 9,320,000,000 × 0.85 (85 per cent efficiency)	7,900,000,000
B.t.u. in fuel consumed per day = 2020 × 18,000	36,000,000
Ft-lb. of work per 1,000,000 B.t.u.	217,000,000
Daily operating cost:	
Fuel oil: 6.58 bbl. at $1.50	$9.87
Lubricating oil: 2 gal. at $0.22	0.44
Cylinder oil: 1.6 gal. at $0.21	0.34
Attendance: Total salaries of 2 engineers, 2 assistant engineers, 1 chief engineer and 2 telegraph operators	41.50
	$52.15
Cost per b.hp-hr. ($52.15 ÷ 4704)	$0.011
Cost per bbl. of oil pumped ($52.15 ÷ 14,000)	0.0037
Bbl. of oil pumped per bbl. of fuel consumed (14,000 ÷ 6.58)	2130

24 In conclusion it may be said that the comparatively small amount of power involved in pipe-line transportation lends itself admirably to the efficient use of the oil engine as a prime mover. And unless some other form of power can show better results in the immediate future, the oil engine bids fair to hold its present superiority as a means for the transportation of oil.

DISCUSSION

Benjamin F. Tillson asked the author what friction coefficients would apply respectively to paraffin and asphalt base oils with densities varying from 38 to 14 deg. B. It seemed to him that in view of the fact that viscosity of the oils played a more important part than their specific gravities, it would be preferable to express a formula so as to include the factor of absolute viscosity of the oil in question. Did such a formula exist, he asked, and if so, what factors were properly applied in practice? What would be the corresponding factors for rifled pipe and in case water films bounded the stream of oil? Was the formula limited for stream line flow with velocity below the critical velocity above which turbulent flow results, and if so, what formula should be used to obtain such critical velocities in terms of viscosities?

Henry H. Suplee said that it was well known that certain gas pipe lines were being operated very economically by using gas power pumps deriving their fuel from the pipe line. The same would be true of an oil pipe line using oil engines for driving the pumps.

Albert C. Dickerman was interested in the paper from the point of view of transporting oil comparatively short distances within plants using oil fuel. He had been unable to locate any very dependable information regarding the most economical pipe line for such installations. He was also interested to know what insulating materials to use in oil and steam lines laid underground.

The Author. An attempt has been made to derive a friction constant for use with the Fanning formula which would be applicable to oils of various viscosities for each sized line. A similar effort was made to derive a constant C for use with the Chézy formula for flow of various viscosities.

It is the writer's belief that it is not practical to write a formula which will permit the flow to be expressed mathematically for all conditions. This flow is influenced by the velocity of the oil, size of the line, viscosity and gravity of the oil, temperature loss, roughness of the pipe, and turbulence of the flow above a certain critical velocity. It is apparent from tests that with the light oils a turbulent flow begins almost as soon as it does in the flow of water while with heavy oils equivalent to [illegible] deg B. California oil at 100 deg. Fahr. this critical point is not reached until the velocity has exceeded approximately three or four feet per second.

The writer has had no experience with rifled pipe.

The writer will be glad to furnish all necessary data of transporting oil short distances if Mr. Dickerson will furnish following information:

Amount of oil to be transported from base of supply to place of consumption.

Amount of oil to be delivered per hour or 24 hours

Viscosity, gravity and temperature of oil to be pumped.

The writer has used to great advantage for insulation of underground pipe lines the Johns-Manville underground system of insulation for both oil and steam lines.

The author wrote that prices of materials and rates of wages paid to pumping station attendants as used in his paper were out of date.

No. 1733

MANUAL ON AMERICAN STANDARD PIPE THREADS[1]

THE American Pipe Thread Standard, also known as the American Briggs Standard, was formulated by Mr. Robert Briggs prior to 1882.

2. Mr. Briggs for several years was superintendent of the Pascal Iron Works of Morris, Tasker & Company, Philadelphia, and later was Engineering Editor of the *Journal* of the Franklin Institute. After his death, a paper by Mr. Briggs containing detailed information regarding American pipe and pipe thread practice was read before the Institution of Civil Engineers of Great Britain. This is recorded in the Excerpt minutes, Volume LXXI, Session 1882–1883, Part 1.

3 While, in a general way, American manufacturers were threading practically to the Briggs Standard, in 1886 the manufacturers and The American Society of Mechanical Engineers jointly adopted it in detail, and master gages were made. The standard has since been in general use in the United States and Canada.

4 At various conferences later, American manufacturers and The American Society of Mechanical Engineers established additional sizes, certain details of gaging, tolerances and special appli-

---

[1] Approved by Council October 1919, and ordered printed. Presented at the Annual Meeting, December 1919, of The American Society of Mechanical Engineers as a progress report of the Joint Committee on International Standard for Pipe Threads. This Manual was prepared at the suggestion of the Committee and is endorsed by the Committee of Manufacturers on Standardization of Fittings and Valves and the National Screw Thread Commission. It has also been endorsed by the following eight societies and associations: Society of Automotive Engineers, The American Society of Refrigerating Engineers, The American Society of Heating and Ventilating Engineers, American Railway Engineering Association, Heating and Piping Contractors National Association, National Federation of Construction Industries, American Railroad Association, Section III — Mechanical, American Railroad Association, Signal Division.

This Standard has been formally presented by the A.S.M.E. to the American Engineering Standards Committee and has been approved by the Committee as an "American Standard."

cations of the standard, also the formulæ and dimensions were tabulated more completely than was originally done by Mr. Briggs.

OUTLINE OF STANDARD

5 The American Pipe Thread Standard establishes the following:

Outside diameter of pipe
Diameter of male thread
Diameter of female thread
Profile of thread
Pitch of thread
Length of thread
Taper of thread
Engagement (by hand) of male and female threads
Construction and use of gages
Tolerances
Use of taper threads
Use of straight threads.

TABLES OF DIMENSIONS

6 The dimensions of American Pipe Threads are expressed in "inches" to one-one hundred thousandth (0.00001) of an inch, and in "millimeters" to one-thousandth (0.001) of a millimeter.

7 While this is a greater degree of accuracy than is ordinarily used, the dimensions are so expressed in order to eliminate errors which might result from less accurate dimensions.

8 The relation between the inch and the meter used in calculating the dimensions in these tables is that established by law in the United States and on record in the Bureau of Standards, Department of Commerce and Labor, Washington, D. C. This is
1 meter = 39.37 inches exactly.

9 The metric equivalent of the inch resulting from this determination is 25.40005 millimeters = 1 inch.

OUTSIDE DIAMETER OF PIPE

10 The outside diameter of pipe is given in Column "G" of the table of dimensions. These diameters should be very closely adhered to by pipe manufacturers.

DIAMETER OF TAPER THREAD

11 The pitch diameters of the taper thread are determined by formulæ based on the outside diameter of pipe and the pitch of thread. These are as follows:

$A = G - (0.05G + 1.1)P$
$B = A + 0.0625F$
A = Pitch diameter of thread at end of pipe
B = Pitch diameter of thread at gaging notch
G = Outside diameter of pipe
F = Normal engagement by hand between male and female threads
P = Pitch of thread.

NOTE. — The above formulæ are not expressed in the same terms as the formula originally established by Mr. Briggs, because they are used to determine *pitch* diameters whereas the Briggs formula determined the *outside* diameter of the thread. However, both forms give identical results.

PROFILE

12 The angle between the sides of the thread is 60 deg. when measured in the axial plane, and the thread is perpendicular to the axis of the pipe, for taper or straight threads. (See Fig. 1.)

13 The crest and root are truncated an amount equal to $0.033P$. The depth of the thread, therefore, is $0.8P$. (See Fig. 1.)

NOTE. — While Mr. Briggs originally advocated a slightly *rounded* crest and root, the thread as applied in the manufacture of gages and thread tools has always been slightly *flattened* at the crest and root.

While the crests on commercially manufactured male and female threads would appear slightly rounded when examined with a microscope, for all practical purposes and when examined by eye they are sharp.

The roots of commercially manufactured threads are practically sharp when cut with new tools and slightly rounded when cut with worn tools.

PITCH

14 The pitch of the thread is the distance the axis will advance in one revolution. It is expressed in terms of the number of threads in one inch and the number of threads in 254 millimeters (254 mm. equals 10 inches).

LENGTH OF THREAD

15 The length of the taper male thread is determined by a formula based on the outside diameter of pipe and the pitch of the thread. This is as follows:

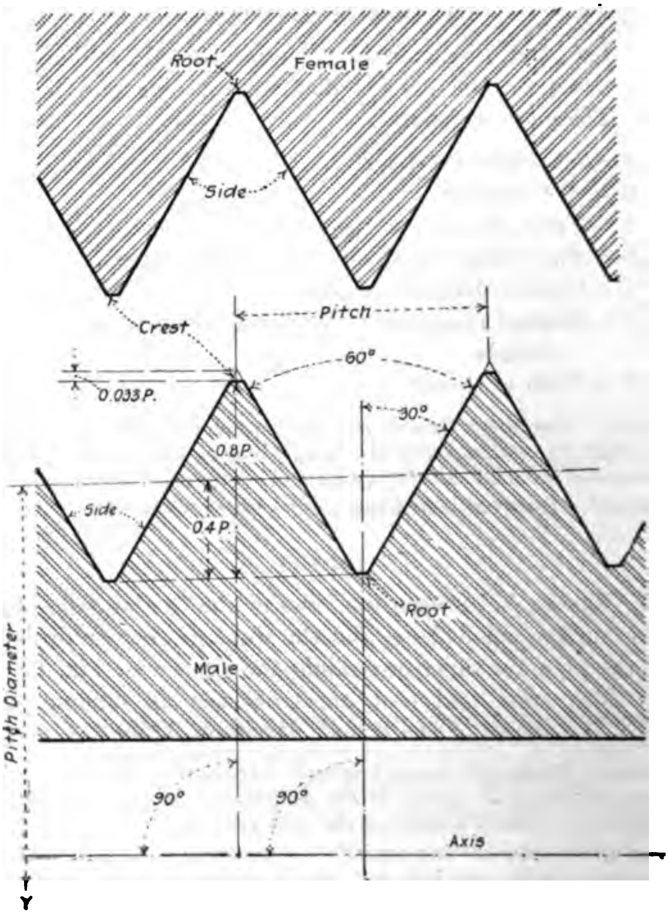


FIG. 1

$$E = (0.8G + 6.8)\,P$$

E = Length of effective thread
G = Outside diameter of Pipe
P = Pitch of thread.

NOTE.—The above formula is not expressed in the same terms as the one originally established by Mr. Briggs, because it determines directly the length of effective thread which includes two threads slightly imperfect on the crest, whereas the Briggs formula determined the length of perfect thread, the two threads imperfect on the crest not being included in the formula. However, both forms give identical results.

TAPER OF THREAD

16 The taper of the thread is 1 in 16 measured on the diameter.

ENGAGEMENT BETWEEN TAPER MALE AND FEMALE THREAD

17 The normal length of engagement between taper male and female threads when screwed together by hand is shown in Column "F" of Table 1.

18 This length is controlled by the construction and use of the gages.

GAGES

19 Gages to properly maintain interchangeability should consist of,

Master gages

Reference gages used for checking working gages

See Figs. 2, 3 and 4

Working gages used for checking the product

See Figs. 5 and 6.

MASTER GAGES

20 The master gage is a taper threaded plug gage. This should be accompanied by two taper threaded ring gages (to be known as checks) to afford ready comparisons.

21 The plug gage is made to the dimensions given in Table 1, as shown in Figs. 2, 3 and 4, and includes the gaging notch.

22 One ring (check) has a thickness equal to dimension F, is the same diameter at the small end as the small end of the plug gage, and is flush with the plug gage at the small end and at the gaging notch when screwed on tight by hand. (See Fig. 2.) This check will be fitted to its master.

23 The other ring (check) has a thickness equal to dimension E, but is threaded for a distance equal to E minus F. It is the same diameter at the large end as the large end of the plug gage. The distance equal to F is counterbored and unthreaded. (See Fig. 3.) This check will also be fitted to its master.

24 The roots of the threads are cut to a sharp V or may be undercut beyond the sharp V to facilitate making the thread. See Fig. 4. The crests are truncated an amount equal to $0.1P$, as illustrated in Fig. 4.

25 Master gages and their checks are primarily for the use of gage and thread tool manufacturers, and for very accurate reference in checking gages.

TABLE 1 AMERICAN TAPER PIPE THREADS (See Diagram on opposite page)

Nominal Size		A		B		E		F		G		Depth of Thread		Number of Threads	
Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Per Inch	Per 254 M.M.
1/8	3	.36351	9.233	.37476	9.519	.2638	6.700	.180	4.572	.405	10.287	.02963	.753	27	270
1/4	6	.47739	12.126	.48989	12.443	.4018	10.206	.200	5.080	.540	13.716	.04444	1.129	18	180
3/8	10	.61201	15.545	.62701	15.926	.4078	10.358	.240	6.096	.675	17.145	.04444	1.129	18	180
1/2	13	.75843	19.264	.77843	19.772	.5337	13.556	.320	8.128	.840	21.336	.05714	1.451	14	140
3/4	19	.96768	24.579	.98886	25.117	.5457	13.861	.339	8.611	1.050	26.670	.05714	1.451	14	140
1	25	1.21363	30.826	1.23863	31.461	.6828	17.343	.400	10.160	1.315	33.401	.06956	1.767	11½	115
1¼	32	1.55713	39.551	1.58338	40.218	.7068	17.953	.420	10.668	1.660	42.164	.06956	1.767	11½	115
1½	38	1.79609	45.621	1.82234	46.287	.7235	18.377	.420	10.668	1.900	48.260	.06956	1.767	11½	115
2	50	2.26902	57.633	2.29627	58.325	.7565	19.215	.436	11.074	2.375	60.325	.06956	1.767	11½	115
2½	64	2.71953	69.076	2.76216	70.159	1.1375	28.892	.682	17.323	2.875	73.025	.10000	2.540	8	80
3	76	3.34063	84.852	3.38850	86.068	1.2000	30.480	.766	19.456	3.500	88.900	.10000	2.540	8	80
3½	90	3.83750	97.473	3.88881	98.776	1.2500	31.750	.821	20.853	4.000	101.600	.10000	2.540	8	80
4	100	4.33438	110.093	4.38713	111.433	1.3000	33.020	.844	21.438	4.500	114.300	.10000	2.540	8	80
4½	113	4.83125	122.714	4.88594	124.103	1.3500	34.290	.875	22.225	5.000	127.000	.10000	2.540	8	80
5	125	5.39073	136.925	5.44929	138.412	1.4063	35.720	.937	23.800	5.563	141.300	.10000	2.540	8	80
6	150	6.44609	163.731	6.50597	165.252	1.5125	38.417	.958	24.333	6.625	168.275	.10000	2.540	8	80
7	175	7.43984	188.972	7.50234	190.560	1.6125	40.957	1.000	25.400	7.625	193.675	.10000	2.540	8	80
8	200	8.43359	214.214	8.50003	215.901	1.7125	43.497	1.063	27.000	8.625	219.075	.10000	2.540	8	80
9	225	9.42734	239.455	9.49797	241.249	1.8125	46.037	1.130	28.702	9.625	244.475	.10000	2.540	8	80
10	250	10.54531	267.851	10.62094	269.772	1.9250	48.895	1.210	30.734	10.750	273.050	.10000	2.540	8	80
11	275	11.53906	293.093	11.61938	295.133	2.0250	51.435	1.285	32.639	11.750	298.450	.10000	2.540	8	80
12	300	12.53281	318.334	12.61781	320.493	2.1250	53.975	1.360	34.544	12.750	323.851	.10000	2.540	8	80
14 O.D.	350	13.77500	349.886	13.87262	352.365	2.250	57.150	1.562	39.675	14.000	355.601	.10000	2.540	8	80
15 O.D.	375	14.76875	375.127	14.87419	377.805	2.350	59.690	1.687	42.850	15.000	381.001	.10000	2.540	8	80
16 O.D.	400	15.76250	400.368	15.87575	403.245	2.450	62.230	1.812	46.025	16.000	406.401	.10000	2.540	8	80
17 O.D.	425	16.75625	425.609	16.87500	428.626	2.550	64.770	1.900	48.260	17.000	431.801	.10000	2.540	8	80
18 O.D.	450	17.75000	450.851	17.87500	454.026	2.650	67.310	2.000	50.800	18.000	457.201	.10000	2.540	8	80
20 O.D.	500	19.73750	501.333	19.87031	504.707	2.850	72.390	2.125	53.975	20.000	508.001	.10000	2.540	8	80
22 O.D.	550	21.72500	551.816	21.86562	555.388	3.050	77.470	2.250	57.150	22.000	558.801	.10000	2.540	8	80
24 O.D.	600	23.71250	602.299	23.86094	606.069	3.250	82.550	2.375	60.325	24.000	609.601	.10000	2.540	8	80
26 O.D.	650	25.70000	652.781	25.85625	656.750	3.450	87.630	2.500	63.500	26.000	660.401	.10000	2.540	8	80
28 O.D.	700	27.68750	703.264	27.85156	707.431	3.650	92.710	2.625	66.675	28.000	711.201	.10000	2.540	8	80
30 O.D.	750	29.67500	753.746	29.84687	758.112	3.850	97.790	2.750	69.850	30.000	762.001	.10000	2.540	8	80

REFERENCE GAGES

26 Reference gages consist of one taper threaded plug gage and two taper threaded ring gages.

27 The plug gage is made to the dimensions given in Table 1, as shown in Figs. 2, 3 and 4, and includes the gaging notch.

28 One ring gage has a thickness equal to dimension *F*, is the same diameter at the small end as the small end of the plug gage, and is flush with the plug gage at the small end and at the gaging notch when screwed on tight by hand. (See Fig. 2.)

29 The other ring gage has a thickness equal to dimension *E*, but is threaded for a distance equal to *E* minus *F*. It is the same

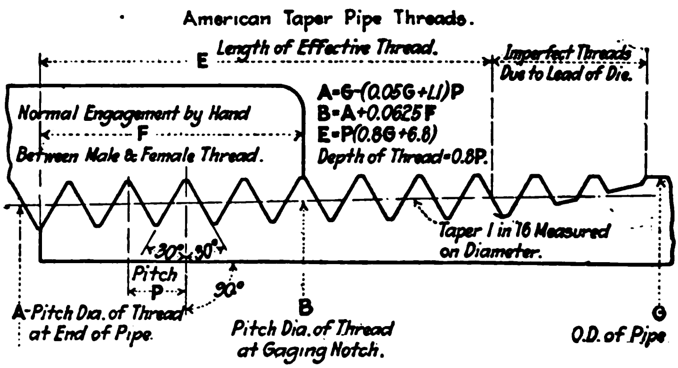


DIAGRAM REFERRED TO IN TABLE 1

diameter at the large end as the large end of the plug gage. The distance equal to *F* is counterbored and unthreaded. (See Fig. 3.)

30 The roots of the threads are cut to a sharp V or may be undercut beyond the sharp V to facilitate making the thread. The crests are truncated an amount equal to 0.1*P*. (See Fig. 4.)

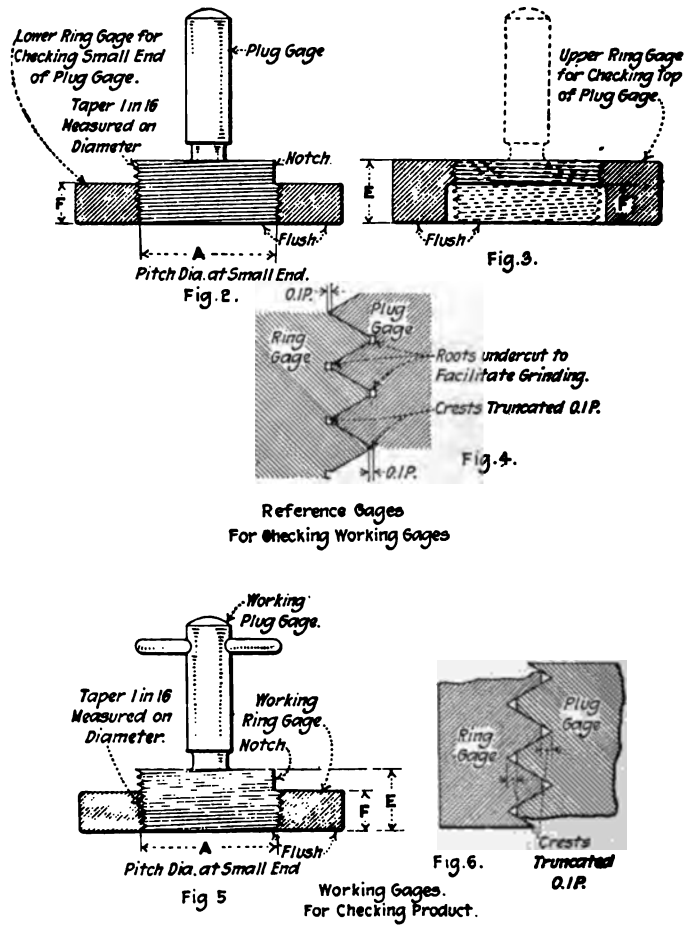
WORKING GAGES

31 Working gages consist of one taper threaded plug gage and one taper threaded ring gage.

32 The plug gage is made to the dimensions given in Table 1, as shown in Figs. 5 and 6, and includes the gaging notch.

33 The ring gage has a thickness equal to dimension *F*, and is the same diameter at the small end as the small end of the plug gage. (See Fig. 5.)

34 The crests are truncated an amount equal to $0.1P$. The roots are cut to a sharp V, or may be undercut beyond the sharp V to facilitate making the thread. (See Fig. 6.)



FIGS. 2–6

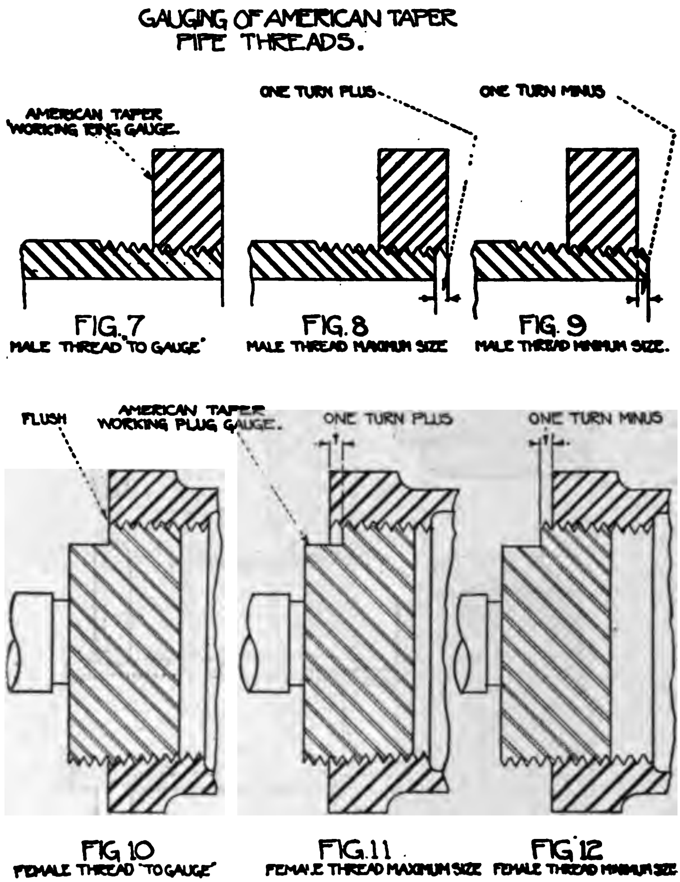
NOTE.—The object of truncating the crests on gages (**truncation 0.1*P***) is to insure that, when gaging a commercial thread cut with **a slightly dull tool,** the gage bears on the sides of the thread instead of on the **roots.**

GAGING FEMALE THREADS

35 The plug gage, Fig. 5, should screw tight by **hand into the** fitting or coupling until the notch is flush with the **face. When the**

thread is chamfered, the notch should be flush with the bottom of the chamfer.

36 This method of gaging is used either for taper female



FIGS. 7–12

threads or for straight threaded female couplings which screw together with taper male threads. (See Figs. 10 and 13.)

GAGING TAPER MALE THREADS

37 The ring gage, Fig. 5, should screw tight by hand on the pipe or male thread until the small end of the gage is flush with the end of the thread. (See Fig. 7.)

GAGE TOLERANCES

38 In the manufacture of gages, variations from basic dimensions are unavoidable. Furthermore, gages will wear in use. In order to fix the maximum allowable variations of gages, tolerances have been established.

39 *Master Gages.* Master gages should be made within the

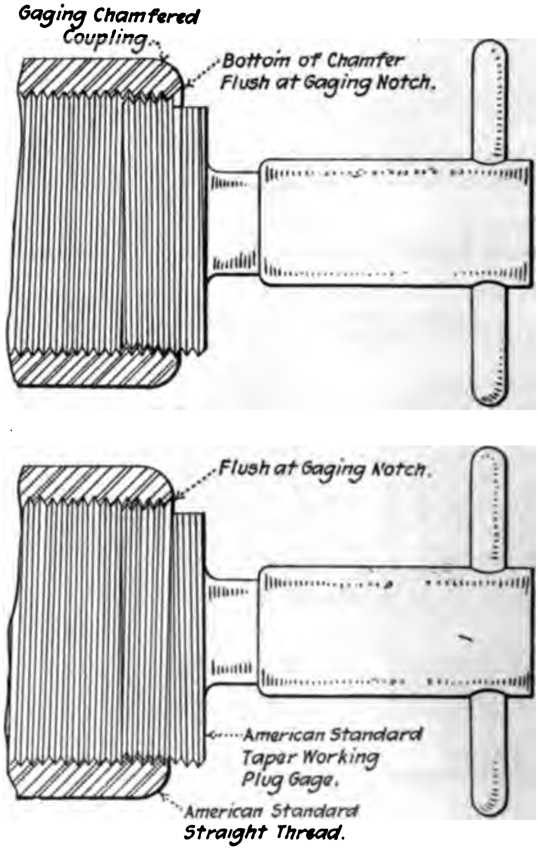


Fig. 13

See Figs. 11 and 12 for Tolerance. Taper Four Times Actual

narrowest possible limits of error, and checks should be fitted to their masters.

40 Each master gage should be accompanied by a record of all measurements, and a statement of the decimal part of a turn it varies plus or minus from the basic dimensions.

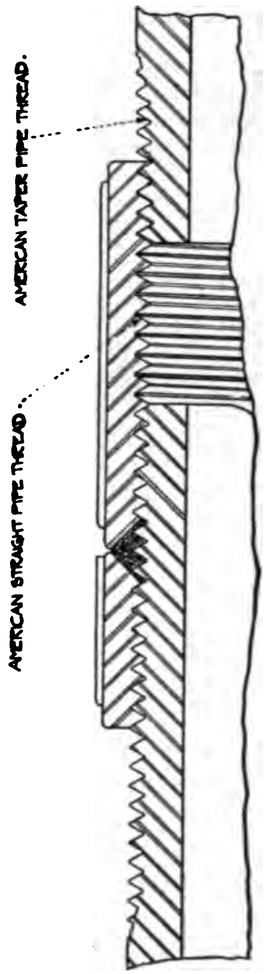


FIG. 14

41 *Reference Gages.* Column 1 of Table 8 gives the maximum allowable cumulation of all errors in the thread surface of reference gages, expressed in terms of diameter, and is illustrated in Fig. 16. No point in the thread surface of the gage should be outside of the zone of tolerance indicated by the shaded portion of the illustration.

NOTE. — This column is used when checking gages by measurement. If the errors in the gage are reported in terms of pitch, angle of thread, and diameter, Tables 10 and 11 may be used to determine the cumulation of these errors for

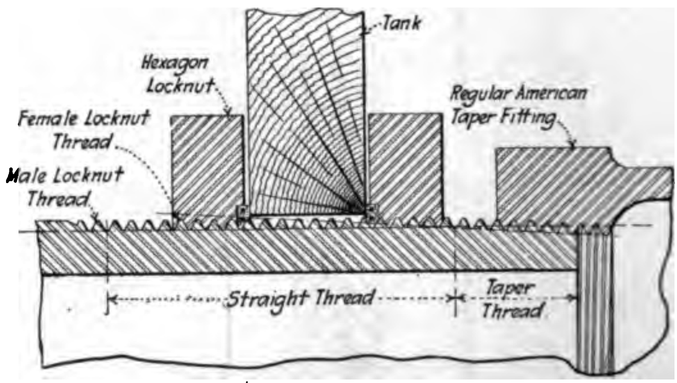


FIG. 15

comparison with Column 1. In Table 10 the results of errors in angle are expressed in terms of diameter. In Table 11 the results of errors in pitch are expressed in terms of diameter.

For example:

A ¾-in. — 14 plug pipe thread gage is reported as follows:

Pitch diameter, large end, 0.98881 in.

Pitch diameter, small end, 0.96775 in.

One-half included angle of thread, 29 deg. and 58 min.

Maximum error in lead, 0.00007 in.

The correct pitch diameter at large end is 0.98886 in. (See Table 1)

The error is 0.00005 in.

The correct pitch diameter at small end is 0.96768 in. (See Table 1)

The error is 0.00007 in.

2 min. error in angle equals 0.00006 in. (See Table 10)

0.00007 in. error in lead equals .00012 in. (See Table 11)

The cumulative error at large end in terms of diameter equals 0.00023 in.

The cumulative error at small end equals 0.00025 in.

The gage falls within the limits of the reference gage (0.00028 in. as given in Table 8).

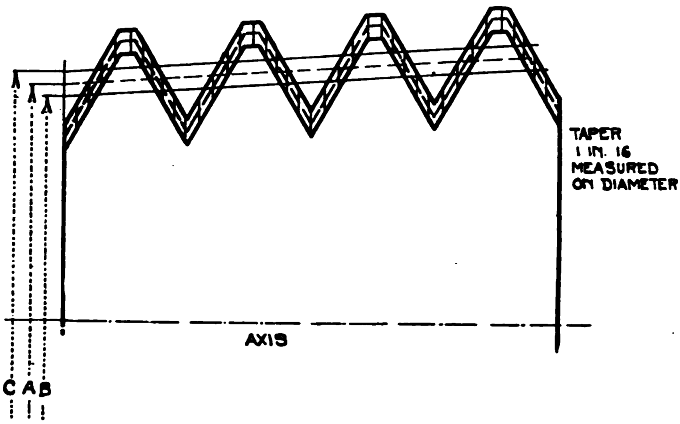


FIG. 16

NOTE CONCERNING FIG. 16. No point on the thread surface of the gage should be outside of the zone of tolerance indicated by the shaded portion of the illustration.

The dotted line indicates the outline of a perfect gage made exactly to the basic dimensions.

A = Basic pitch diameter at small end of gage
B = Minimum pitch diameter at small end of gage
C = Maximum pitch diameter at small end of gage

$$B = A - \begin{cases} \text{Column 1 from Table 8 for reference gages} \\ \text{Column 5 from Table 9 for new working gages} \end{cases}$$

$$C = A + \begin{cases} \text{Column 1 from Table 8 for reference gages} \\ \text{Column 5 from Table 9 for new working gages} \end{cases}$$

42 Column 2, Table 8, gives the equivalent of Column 1, expressed in terms of distance parallel to the axis, and represents the maximum distance which a reference ring gage of perfect thickness or a reference plug gage of perfect length from small end to gaging notch may vary from being flush at the gaging notch or at the small end, when referred to basic dimensions. It is equal to 16 times Column 1, owing to the basic taper of 1 in 16, measured on the diameter.

NOTE.—This column is used when checking reference gages by comparison with a master gage. The necessary allowance must be made for the error in the master.

AMERICAN STRAIGHT PIPE THREADS, FEMALE.

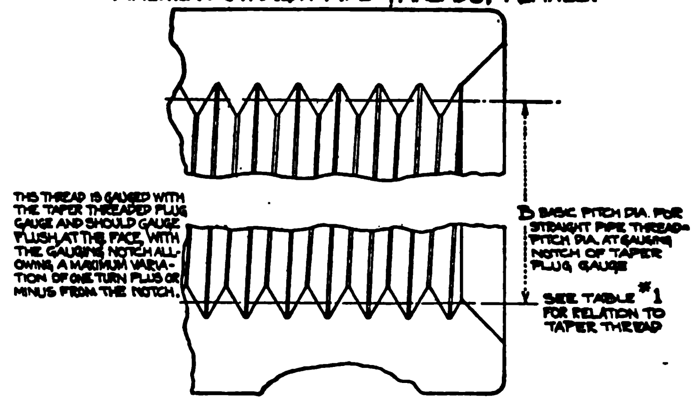


TABLE 2

		B		Depth of Thread		Number of Threads	
Inches	M.M.	Inches	M.M.	Inches	M.M.	Per Inch	Per 254 M.M.
1/8	3	.37476	9.519	.02963	.753	27	270
1/4	6	.48989	12.443	.04444	1.129	18	180
3/8	10	.62701	15.926	.04444	1.129	18	180
1/2	13	.77843	19.772	.05714	1.451	14	140
3/4	19	.98886	25.117	.05714	1.451	14	140
1	25	1.23863	31.461	.06956	1.767	11½	115
1¼	32	1.58338	40.218	.06956	1.767	11½	115
1½	38	1.82234	46.287	.06956	1.767	11½	115
2	50	2.29627	58.325	.06956	1.767	11½	115
2½	64	2.76216	70.159	.10000	2.540	8	80
3	76	3.38850	86.068	.10000	2.540	8	80
3½	90	3.88881	98.776	.10000	2.540	8	80
4	100	4.38713	111.433	.10000	2.540	8	80
4½	113	4.88594	124.103	.10000	2.540	8	80
5	125	5.44929	138.412	.10000	2.540	8	80
6	150	6.50597	165.252	.10000	2.540	8	80
7	175	7.50234	190.560	.10000	2.540	8	80
8	200	8.50003	215.901	.10000	2.540	8	80
9	225	9.49797	241.249	.10000	2.540	8	80
10	250	10.62094	269.772	.10000	2.540	8	80
11	275	11.61938	295.133	.10000	2.540	8	80
12	300	12.61781	320.493	.10000	2.540	8	80
14 O.D.	350	13.87262	352.365	.10000	2.540	8	80
15 O.D.	375	14.87419	377.805	.10000	2.540	8	80
16 O.D.	400	15.87575	403.245	.10000	2.540	8	80
17 O.D.	425	16.87500	428.626	.10000	2.540	8	80
18 O.D.	450	17.87500	454.026	.10000	2.540	8	80
20 O.D.	500	19.87031	504.707	.10000	2.540	8	80
22 O.D.	550	21.86562	555.388	.10000	2.540	8	80
24 O.D.	600	23.86094	606.069	.10000	2.540	8	80
26 O.D.	650	25.85625	656.750	.10000	2.540	8	80
28 O.D.	700	27.85156	707.431	.10000	2.540	8	80
30 O.D	750	29.84687	758.112	.10000	2.540	8	80

43 Column 3 gives the equivalent of Column 2, expressed in terms of the decimal part of a turn from the basic dimensions.

NOTE. — This column is also used when checking reference gages by comparison with a master gage. The necessary allowance must be made for the error in the master.

44 A tolerance of plus or minus 0.0002 in. (0.005 mm.) is allowed on the distance between the gaging notch and the small end of the reference plug gage, or on the thickness of the reference ring gage.

NOTE. — It is possible for reference plug and ring gages which come within all of the above tolerances to vary from being flush with each other at the small end, or at the gaging notch, when screwed together tight by hand. The maximum variation which might occur, expressed in terms of distance, is given in Column 4, and gages which come within these limits should be checked by measurement before being rejected.

45 *New Working Gages.* Column 5 of Table 9 gives the maximum allowable cumulation of all errors in the thread surface of new working gages, expressed in terms of diameter, and is also illustrated in Fig. 16. No point in the thread surface of the gage should be outside of the zone of tolerance indicated by the shaded portion of the illustration.

NOTE. — This column is used when checking gages by measurement.

46 Column 6 gives the equivalent of Column 5, expressed in terms of distance parallel to the axis, and represents the maximum distance which a new working ring gage of perfect thickness or a new working plug gage of perfect length from small end to gaging notch may vary from being flush at the gaging notch, or at the small end, when referred to basic dimensions. It is equal to 16 times Column 5, owing to the basic taper of 1 in 16, measured on the diameter.

NOTE. — This column is used when checking working gages by comparison with a gage the error of which is known. The necessary allowance must be made for this error.

47 Column 7 gives the equivalent of Column 6, expressed in terms of the decimal part of a turn from basic dimensions.

NOTE. — This column is also used when checking working gages by comparison with a gage the error of which is known. The necessary allowance must be made for this error.

48 A tolerance of plus or minus 0.0005 in. (0.0127 mm.) is allowed on the distance between the gaging notch and the small end of the working plug gage, or on the thickness of the working ring gage.

NOTE. — It is possible for working plug and ring gages which come within all of the above tolerances to vary from being flush with each other at the small

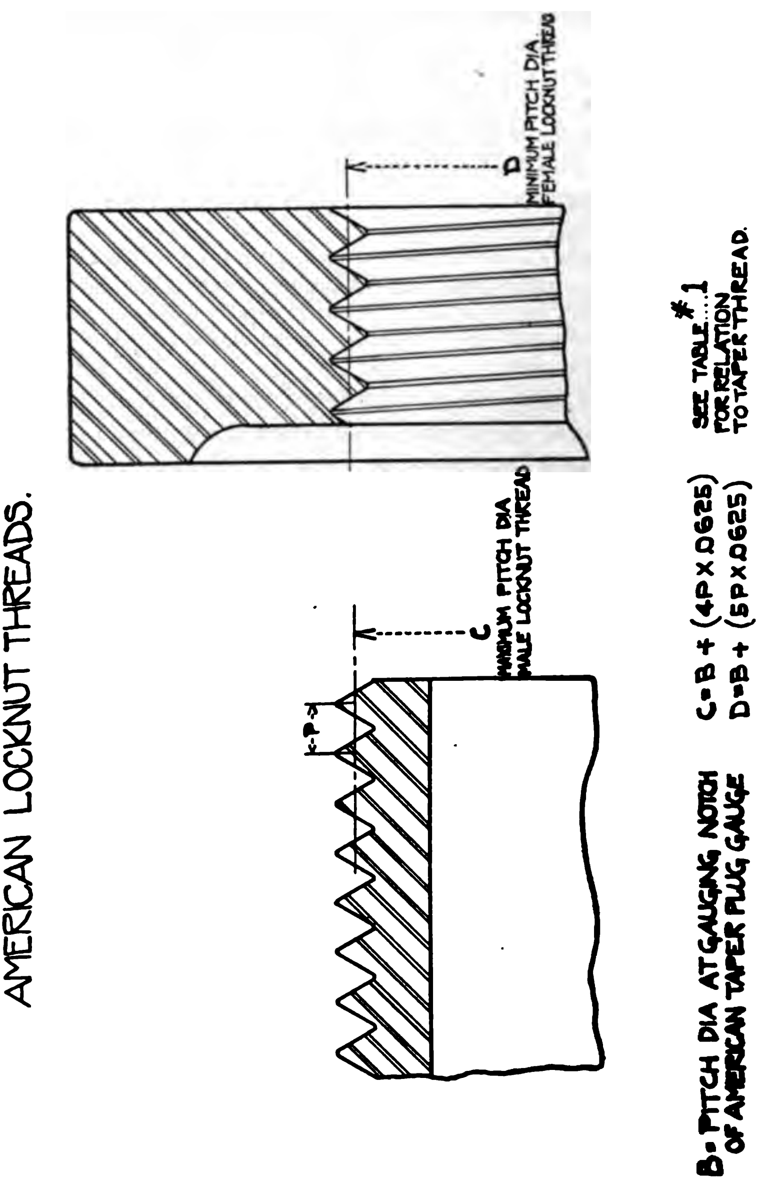

AMERICAN LOCKNUT THREADS.
D
MINIMUM PITCH DIA. FEMALE LOCKNUT THREAD
C
MAXIMUM PITCH DIA MALE LOCKNUT THREAD
P
B = PITCH DIA AT GAUGING NOTCH OF AMERICAN TAPER PLUG GAUGE
C = B + (4P x .0625)
D = B + (5P x .0625)
SEE TABLE #1 FOR RELATION TO TAPER THREAD.

TABLE 3

Nominal Size		C		D		Depth of Thread		Number of Threads	
Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Per Inch	Per 254 M.M.
1/8	3	.38402	9.754	.38633	9.813	.02963	.753	27	270
1/4	6	.50378	12.796	.50725	12.884	.04444	1.129	18	180
3/8	10	.64090	16.279	.64437	16.367	.04444	1.129	18	180
1/2	13	.79628	20.225	.80075	20.339	.05714	1.451	14	140
3/4	19	1.00672	25.571	1.01118	25.684	.05714	1.451	14	140
1	25	1.26037	32.013	1.26580	32.151	.06956	1.767	11½	115
1¼	32	1.60512	40.770	1.61055	40.908	.06956	1.767	11½	115
1½	38	1.84407	46.839	1.84951	46.978	.06956	1.767	11½	115
2	50	2.31801	58.877	2.32344	59.015	.06956	1.767	11½	115
2½	64	2.79341	70.953	2.80122	71.151	.10000	2.540	8	80
3	76	3.41975	86.862	3.42756	87.060	.10000	2.540	8	80
3½	90	3.92006	99.570	3.92787	99.768	.10000	2.540	8	80
4	100	4.41838	112.227	4.42619	112.425	.10000	2.540	8	80
4½	113	4.91719	124.897	4.92500	125.095	.10000	2.540	8	80
5	125	5.48054	139.206	5.48836	139.405	.10000	2.540	8	80
6	150	6.53722	166.046	6.54503	166.244	.10000	2.540	8	80
7	175	7.53359	191.354	7.54141	191.552	.10000	2.540	8	80
8	200	8.53128	216.695	8.53909	216.893	.10000	2.540	8	80
9	225	9.52922	242.043	9.53703	242.241	.10000	2.540	8	80
10	250	10.65219	270.566	10.66000	270.764	.10000	2.540	8	80
11	275	11.65063	295.927	11.65844	296.125	.10000	2.540	8	80
12	300	12.64906	321 287	12.65688	321.485	.10000	2.540	8	80

end or at the gaging notch, when screwed together tight by hand. The maximum variation which might occur, expressed in terms of distance, is given in Column 8, and gages which come within these limits should be checked by comparison with reference gages before being rejected.

It is also possible for working plug and ring gages which come within all of the above tolerances to vary from being flush at the small end or at the gaging notch, when screwed tight by hand on a reference gage which comes within the tolerances specified for reference gages. The maximum variation which might occur, expressed in terms of distance, is given in Column 9, and gages which come within these limits should be checked by measurement before being rejected.

49 *Worn Working Gages.* The maximum wear on working gages must not be more than the equivalent of one-half turn from the basic dimensions.

MANUFACTURING TOLERANCE

50 The maximum allowable variation in the commercial product is one turn plus or one turn minus from the gaging notch when using working gages. (See Figs. 8, 9, 10 and 12.) This is equivalent to a maximum allowable variation of 1½ turns from the basic dimensions, owing to the allowance of one-half turn on working gages.

AMERICAN TAPER PIPE THREADS

51 Taper male and female threads are recommended for threaded joints for any service.

AMERICAN STRAIGHT PIPE THREADS

52 *Female.* Straight threaded female wrought iron or wrought steel couplings of the weight known as "standard" may be used with taper threaded pipe for ordinary pressures, as they are sufficiently ductile to adjust themselves to the taper male thread when properly screwed together. For dimensions see Table 2.

53 For high pressures, only taper male and female threads should be used.

54 *Male.* Straight male threads are recognized only for special applications such as:

Long screws
Tank nipples.

LONG SCREWS

55 Long screws are used to a limited extent. This joint is not considered satisfactory when subjected to temperature or pressure.

In this application (see Fig. 14), the coupling has a straight thread and must make a joint with an American taper pipe thread.

56 In gaging, the American taper working plug gage is used, allowing the same tolerance from the notch as for a taper thread. (See Fig. 13.) The straight thread on the pipe enters the coupling freely by hand, the joint being made by a packing material between the locknut and the coupling. (See Fig. 14.)

57 It is necessary that the coupling be screwed on the straight male thread for the full length of the coupling and then back until it engages the taper male thread.

58 Owing to the long engagement of thread, imperfections in pitch affect the fit when the coupling is screwed on the pipe its full length. Refinements of manufacture and gaging to insure a properly interchangeable product are more costly than the commercial use warrants; therefore, the use of this type of joint is not recommended.

LOCKNUT THREADS

59 Occasional requirements make it advisable to have a straight thread of the largest diameter it is possible to cut on a pipe. This has been standardized and is known as Maximum Male and Minimum Female Locknut Threads. For dimensions, see Table 3. The "Tank Nipple" shown in Fig. 15 is an example of this thread. In this application an American standard taper thread is cut on the end of the pipe after having first cut the male locknut thread.

Respectfully submitted,

JOINT COMMITTEE ON INTERNATIONAL STANDARD FOR PIPE THREADS

EDWIN M. HERR, *Chairman* WILLIAM J. BALDWIN LAURENCE V. BENET GEORGE M. BOND STANLEY G. FLAGG, JR.	*Representing The American Society of Mechanical Engineers*
ALTEN S. MILLER W. CULLEN MORRIS HERMAN RUSSELL	*Representing The American Gas Association*

TABLE 4 STANDARD WROUGHT PIPE. TABLE OF DIMENSIONS

Nominal Size		Inside Diameter		External Diameter		Nominal Thickness		Transverse Areas				Length of Pipe per Sq. Ft. of External Surface	Length of Pipe per Sq. M. of External Surface	Nominal Weight per Ft. Threaded and Coupled	Nominal Weight per M. Threaded and Coupled
								Internal		Metal					
Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Sq. Inches	Sq. M.M.	Sq. Inches	Sq. M.M.	Feet	Meters	Pounds	Kilograms
1/8	3	.269	6.833	.405	10.287	.068	1.727	.057	36.77	.072	46.45	9.431	30.94	.245	.364
1/4	6	.364	9.246	.540	13.716	.088	2.235	.104	67.10	.125	80.64	7.073	23.20	.425	.632
3/8	10	.493	12.522	.675	17.145	.091	2.311	.191	123.23	.167	107.74	5.658	18.56	.568	.845
1/2	13	.622	15.799	.840	21.336	.109	2.769	.304	196.13	.250	161.29	4.547	14.91	.852	1.268
3/4	19	.824	20.930	1.050	26.670	.113	2.870	.533	343.87	.333	214.84	3.637	11.93	1.134	1.687
1	25	1.049	26.645	1.315	33.401	.133	3.378	.864	557.42	.494	318.71	2.904	9.53	1.684	2.506
1 1/4	32	1.380	35.052	1.660	42.164	.140	3.556	1.495	964.52	.669	431.61	2.301	7.55	2.281	3.394
1 1/2	38	1.610	40.894	1.900	48.260	.145	3.683	2.036	1313.55	.799	515.48	2.010	6.59	2.731	4.064
2	50	2.067	52.502	2.375	60.325	.154	3.912	3.355	2164.52	1.075	693.55	1.608	5.27	3.678	5.473
2 1/2	64	2.469	62.713	2.875	73.025	.203	5.156	4.788	3089.04	1.704	1099.36	1.328	4.36	5.810	8.660
3	76	3.068	77.927	3.500	88.900	.216	5.486	7.393	4769.69	2.228	1437.42	1.091	3.58	7.616	11.334
3 1/2	90	3.548	90.119	4.000	101.600	.226	5.740	9.886	6378.08	2.680	1729.03	.954	3.13	9.202	13.694
4	100	4.026	102.261	4.500	114.300	.237	6.020	12.730	8212.92	3.174	2047.74	.848	2.78	10.889	16.204
4 1/2	113	4.506	114.453	5.000	127.000	.247	6.274	15.947	10288.41	3.688	2379.35	.763	2.50	12.642	18.813
5	125	5.047	128.194	5.563	141.300	.258	6.553	20.006	12907.12	4.300	2774.20	.686	2.25	14.810	22.040
6	150	6.065	154.051	6.625	168.275	.280	7.112	28.891	18639.39	5.581	3600.65	.576	1.89	19.185	28.550
7	175	7.023	178.385	7.625	193.675	.301	7.645	38.738	24992.31	6.926	4468.39	.500	1.64	23.769	35.372
8	200	8.071	205.004	8.625	219.075	.277	7.036	51.161	33007.16	7.265	4687.10	.442	1.45	25.000	37.204
8	200	7.981	202.718	8.625	219.075	.322	8.179	50.027	32275.55	8.399	5418.72	.442	1.45	28.809	42.872
9	225	8.941	227.102	9.625	244.475	.342	8.687	62.786	40507.17	9.974	6434.85	.396	1.30	34.188	50.877
10	250	10.192	258.877	10.750	273.050	.279	7.087	81.585	52635.58	9.178	5921.30	.355	1.16	32.000	47.621
10	250	10.136	257.455	10.750	273.050	.307	7.798	80.691	52058.81	10.072	6498.08	.355	1.16	35.000	52.086
10	250	10.020	254.508	10.750	273.050	.365	9.271	78.855	50874.29	11.908	7682.59	.355	1.16	41.132	61.211
11	275	11.000	279.400	11.750	298.450	.375	9.525	95.033	61311.73	13.401	8645.82	.325	1.07	46.247	68.823
12	300	12.090	307.067	12.750	323.851	.330	8.382	114.800	74064.66	12.876	8307.11	.299	.98	45.000	66.967
12	300	12.000	304.801	12.750	323.851	.375	9.525	113.097	72965.95	14.579	9405.82	.299	.98	50.706	75.459

TABLE 5 EXTRA STRONG WROUGHT PIPE. TABLE OF DIMENSIONS

Nominal Size		Inside Diameter		External Diameter		Nominal Thickness		Transverse Areas				Length of Pipe per Sq. Ft. of External Surface	Length of Pipe per Sq. M. of External Surface	Nominal Weight per Ft. Plain Ends	Nominal Weight per M. Plain Ends
								Internal		Metal					
Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Sq. Inches	Sq. M.M.	Sq. Inches	Sq. M.M.	Feet	Meters	Pounds	Kilograms
1/8	3	.215	5.461	.405	10.287	.095	2.413	.036	23.22	.093	60.00	9.431	30.94	.314	.467
1/4	6	.302	7.671	.540	13.716	.119	3.023	.072	46.45	.157	101.29	7.073	23.20	.535	.796
3/8	10	.423	10.744	.675	17.145	.126	3.200	.141	90.97	.217	140.00	5.658	18.56	.738	1.10
1/2	13	.546	13.868	.840	21.336	.147	3.734	.234	150.97	.320	206.45	4.547	14.92	1.087	1.62
3/4	19	.742	18.847	1.050	26.670	.154	3.912	.433	279.35	.433	279.35	3.637	11.93	1.473	2.19
1	25	.957	24.308	1.315	33.401	.179	4.547	.719	463.87	.639	412.26	2.904	9.53	2.171	3.23
1 1/4	32	1.278	32.461	1.660	42.164	.191	4.851	1.283	827.74	.881	568.39	2.301	7.55	2.996	4.46
1 1/2	38	1.500	38.100	1.900	48.260	.200	5.080	1.767	1140.00	1.068	689.03	2.010	6.59	3.631	5.40
2	50	1.939	49.251	2.375	60.325	.218	5.537	2.953	1905.16	1.477	952.91	1.608	5.27	5.022	7.47
2 1/2	64	2.323	59.004	2.875	73.025	.276	7.010	4.238	2734.20	2.254	1454.20	1.328	4.36	7.661	11.40
3	76	2.900	73.660	3.500	88.900	.300	7.620	6.605	4261.30	3.016	1945.81	1.091	3.58	10.252	15.26
3 1/2	90	3.364	85.446	4.000	101.600	.318	8.077	8.888	5734.20	3.678	2372.91	.954	3.13	12.505	18.61
4	100	3.826	97.180	4.500	114.300	.337	8.560	11.497	7417.43	4.407	2843.23	.848	2.78	14.983	22.30
4 1/2	113	4.290	108.966	5.000	127.000	.355	9.017	14.455	9325.82	5.180	3341.94	.763	2.50	17.611	26.21
5	125	4.813	122.250	5.563	141.300	.375	9.525	18.194	11738.09	6.112	3943.23	.686	2.25	20.778	30.92
6	150	5.761	146.330	6.625	168.275	.432	10.973	26.067	16817.45	8.405	5422.59	.576	1.89	28.573	42.52
7	175	6.625	168.275	7.625	193.675	.500	12.700	34.472	22240.04	11.192	7220.66	.500	1.64	38.048	56.62
8	200	7.625	193.675	8.625	219.075	.500	12.700	45.663	29460.06	12.763	8234.21	.442	1.45	43.388	64.57
9	225	8.625	219.075	9.625	244.475	.500	12.700	58.426	37694.27	14.334	9247.76	.396	1.30	48.728	72.51
10	250	9.750	247.650	10.750	273.050	.500	12.700	74.662	48169.12	16.101	10387.76	.355	1.16	54.735	81.45
11	275	10.750	273.050	11.750	298.450	.500	12.700	90.763	58556.89	17.671	11400.67	.325	1.07	60.075	89.40
12	300	11.750	298.450	12.750	323.851	.500	12.700	108.434	69957.55	19.242	12414.22	.299	.98	65.415	97.35

TABLE 6 DOUBLE EXTRA STRONG WROUGHT PIPE. TABLE OF DIMENSIONS

Nominal Size		Inside Diameter		External Diameter		Nominal Thickness		Transverse Areas				Length of Pipe per Sq. Ft. of External Surface	Length of Pipe per Sq. M. of External Surface	Nominal Weight per Ft. Plain Ends	Nominal Weight per M. Plain Ends
								Internal		Metal					
Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Sq. Inches	Sq. M.M.	Sq. Inches	Sq. M.M.	Feet	Meters	Pounds	Kilograms
½	13	.252	6.401	.840	21.336	.294	7.468	.050	32.26	.504	325.16	4.547	14.92	1.714	2.551
¾	19	.434	11.024	1.050	26.670	.308	7.823	.148	95.48	.718	463.23	3.637	11 93	2.440	3.631
1	25	.599	15.215	1.315	33.401	.358	9.093	.282	181.93	1.076	694.19	2.904	9.53	3 659	5.445
1¼	32	.896	22.758	1.660	42.164	.382	9.703	.630	406.45	1 534	989.68	2.301	7.55	5.214	7.759
1½	38	1.100	27.940	1.900	48.260	.400	10.160	.950	612.90	1.885	1216.13	2.010	6.59	6.408	9.536
2	50	1.503	38.176	2.375	60.325	.436	11.074	1.774	1144.52	2.656	1713.55	1.608	5.27	9.029	13.436
2½	64	1.771	44.983	2.875	73.025	.552	14.021	2.464	1589.68	4.028	2598.71	1.328	4.36	13.695	20.380
3	76	2.300	58.420	3.500	88.900	.600	15.240	4.155	2680.65	5.466	3526.46	1.091	3.58	18.583	27.654
3½	90	2.728	69.291	4.000	101.600	.636	16.154	5.845	3770.97	6.721	4336.14	.954	3.13	22.850	34.004
4	100	3.152	80.061	4.500	114.300	.674	17.120	7.803	5034.20	8.101	5226.46	.848	2.78	27.541	40.985
4½	113	3.580	90.932	5.000	127.000	.710	18.034	10.066	6494.21	9.569	6173.56	.763	2.50	32.530	48.410
5	125	4.063	103.200	5.563	141.300	.750	19.050	12.966	8365.18	11.340	7316.14	.686	2.25	38.552	57.371
6	150	4.897	124.384	6.625	168.275	.864	21.946	18.835	12151.64	15.637	10088.41	.576	1.89	53.160	79.110
7	175	5.875	149.225	7.625	193.675	.875	22.225	27.109	17489.71	18.555	11970.99	.500	1.64	63.079	93.872
8	200	6.875	174.625	8.625	219.075	.875	22.225	37.122	23940.72	21.304	13744 54	.442	1.45	72.424	107.778

TABLE 7 LARGE O. D. PIPE

Nominal Size		Outside Diameter		INSIDE DIAMETER																	
				¼ Inch Thick	6.4 m/m Thick	5/16 Inch Thick	7.9 m/m Thick	⅜ Inch Thick	9.5 m/m Thick	7/16 Inch Thick	11.1 m/m Thick	½ Inch Thick	12.7 m/m Thick	9/16 Inch Thick	14.3 m/m Thick	⅝ Inch Thick	15.9 m/m Thick	¾ Inch Thick	19.0 m/m Thick	1 Inch Thick	25.4 m/m Thick
Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.
14	350	14	355.6	13½	342.9	13⅜	339.7	13¼	336.6	13⅛	333.4	13	330.2	12⅞	327.0	12¾	323.8	12½	317.5	12	304.8
15	375	15	381.0	14½	368.3	14⅜	365.1	14¼	362.0	14⅛	358.8	14	355.6	13⅞	352.4	13¾	349.2	13½	342.9	13	330.2
16	400	16	406.4	15½	393.7	15⅜	390.5	15¼	387.4	15⅛	384.2	15	381.0	14⅞	377.8	14¾	374.6	14½	368.3	14	355.6
17	425	17	431.8	16½	419.1	16⅜	415.9	16¼	412.8	16⅛	409.6	16	406.4	15⅞	403.2	15¾	400.0	15½	393.7	15	381.0
18	450	18	457.2	17½	444.5	17⅜	441.3	17¼	438.2	17⅛	435.0	17	431.8	16⅞	428.6	16¾	425.4	16½	419.1	16	406.4
20	500	20	508.0	......	......	19⅜	492.1	19¼	489.0	19⅛	485.8	19	482.6	18⅞	479.4	18¾	476.2	18½	469.9	18	457.2
22	550	22	558.8	......	......	21⅜	542.9	21¼	539.8	21⅛	536.6	21	533.4	20⅞	530.2	20¾	527.0	20½	520.7	......	......
24	600	24	609.6	......	......	......	......	23¼	590.6	23⅛	587.4	23	584.2	22⅞	581.0	22¾	577.8	22½	571.5	......	......
26	650	26	360.4	......	......	......	......	25¼	641.4	25⅛	638.2	25	635.0	24⅞	631.8	24¾	628.6	24½	622.3	......	......
28	700	28	711.2	......	......	......	......	......	......	27⅛	689.0	27	685.8	26⅞	682.6	26¾	679.4	26½	673.1	......	......
30	750	30	762.0	......	......	......	......	......	......	29⅛	739.8	29	736.6	28⅞	733.4	28¾	730.2	28½	723.9	......	......

TABLE 8 TOLERANCES FOR REFERENCE GAGES

Nominal Size		1 (Total cumulative tolerance on diameter. (See Fig. 16.)		2 (Equivalent longitudinal variation (16 × Col. 1.)		3 Equivalent Angular variation expressed as decimal part of one turn.	4 *	
Inches	M.M.	Inches	M.M.	Inches	M.M.		Inches	M.M.
1/8	3	.00020	.0050	.0032	.081	.086	.0068	.173
1/4	6	.00022	.0056	.0035	.089	.063	.0074	.188
3/8	10	.00024	.0061	.0038	.097	.068	.0080	.203
1/2	13	.00026	.0066	.0042	.107	.059	.0088	.224
3/4	19	.00028	.0071	.0045	.114	.063	.0094	.239
1	25	.00030	.0076	.0048	.122	.055	.0100	.254
1 1/4	32	.00032	.0081	.0051	.130	.059	.0106	.269
1 1/2	38	.00034	.0086	.0054	.137	.062	.0112	.284
2	50	.00036	.0091	.0058	.147	.067	.0120	.305
2 1/2	64	.00038	.0097	.0061	.155	.050	.0126	.320
3	76	.00038	.0097	.0061	.155	.050	.0126	.320
3 1/2	90	.00041	.0104	.0066	.168	.053	.0136	.345
4	100	.00043	.0109	.0069	.175	.055	.0142	.361
4 1/2	113	.00045	.0114	.0072	.183	.058	.0148	.376
5	125	.00047	.0119	.0075	.191	.060	.0154	.391
6	150	.00051	.0130	.0082	.208	.065	.0168	.427
7	175	.00055	.0140	.0088	.224	.070	.0180	.457
8	200	.00059	.0150	.0094	.239	.075	.0192	.488
9	225	.00063	.0160	.0101	.257	.080	.0206	.523
10	250	.00066	.0168	.0106	.269	.085	.0216	.549
12	300	.00074	.0188	.0118	.300	.095	.0240	.610
14	350	.00082	.0208	.0131	.333	.105	.0266	.676
16	400	.00090	.0229	.0144	.366	.115	.0292	.742
18	450	.00098	.0249	.0157	.399	.125	.0318	.808
20	500	.00106	.0269	.0170	.432	.135	.0344	.874
22	550	.00113	.0287	.0181	.460	.145	.0366	.930
24	600	.00121	.0307	.0194	.493	.155	.0392	.996
26	650	.00129	.0328	.0206	.523	.165	.0416	1.057
28	700	.00137	.0348	.0219	.556	.175	.0442	1.123
30	750	.00144	.0366	.0230	.584	.185	.0464	1.179

*Maximum amount it is possible for plug and ring gauges to vary from being flush at small end or at gauging notch when screwed together tight by hand. (2 times Column 2 + .0004".)

TABLE 9 TOLERANCES FOR WORKING GAGES

Nominal Size		5 Total cumulative tolerance on diameter. (See Fig. 16.)		6 Equivalent longitudinal variation. (16 times Column 5.)		7 ‡	8 *		9 †	
Inches	M.M.	Inches	M.M.	Inches	M.M.		Inches	M.M.	Inches	M.M.
1/8	3	.00040	.0102	.0064	.163	.172	.0138	.351	.0103	.262
1/4	6	.00044	.0112	.0070	.178	.126	.0150	.381	.0112	.284
3/8	10	.00048	.0122	.0077	.196	.136	.0164	.417	.0122	.310
1/2	13	.00052	.0132	.0083	.211	.118	.0176	.447	.0132	.335
3/4	19	.00056	.0142	.0090	.229	.126	.0190	.483	.0142	.361
1	25	.00060	.0152	.0096	.244	.110	.0202	.513	.0151	.384
1 1/4	32	.00064	.0163	.0102	.259	.118	.0214	.544	.0160	.406
1 1/2	38	.00068	.0173	.0109	.277	.124	.0228	.579	.0170	.432
2	50	.00072	.0183	.0115	.292	.134	.0240	.610	.0180	.457
2 1/2	64	.00076	.0193	.0122	.310	.100	.0254	.645	.0190	.483
3	76	.00076	.0193	.0122	.310	.100	.0254	.645	.0190	.483
3 1/2	90	.00082	.0208	.0131	.333	.105	.0272	.691	.0204	.518
4	100	.00086	.0218	.0138	.351	.110	.0286	.726	.0214	.544
4 1/2	113	.00090	.0229	.0144	.366	.115	.0298	.757	.0223	.566
5	125	.00094	.0239	.0150	.381	.120	.0310	.787	.0232	.589
6	150	.00102	.0260	.0163	.414	.130	.0336	.853	.0252	.640
7	175	.00110	.0330	.0176	.447	.140	.0362	.919	.0271	.688
8	200	.00118	.0348	.0189	.480	.150	.0388	.986	.0290	.737
9	225	.00126	.0370	.0202	.513	.160	.0414	1.052	.0310	.787
10	250	.00132	.0432	.0211	.536	.170	.0432	1.097	.0324	.823
12	300	.00148	.0472	.0237	.602	.190	.0484	1.229	.0362	.919
14	350	.00164	.0518	.0262	.665	.210	.0534	1.356	.0400	1.016
16	400	.00180	.0554	.0288	.732	.230	.0586	1.488	.0439	1.115
18	450	.00196	.0594	.0314	.798	.250	.0638	1.621	.0478	1.214
20	500	.00212	.0683	.0339	.861	.270	.0688	1.748	.0516	1.311
22	550	.00226	.0719	.0362	.919	.290	.0734	1.864	.0550	1.397
24	600	.00242	.0759	.0387	.983	.310	.0784	1.991	.0588	1.494
26	650	.00258	.0800	.0413	1.049	.330	.0836	2.123	.0626	1.590
28	700	.00274	.0841	.0438	1.113	.350	.0886	2.250	.0664	1.687
30	750	.00288	.0732	.0461	1.171	.370	.0932	2.367	.0698	1.773

‡Equivalent angular variation expressed as a decimal part of one turn.

*Maximum amount it is possible for new working plug and ring gauges which come within the specified tolerances to vary from being flush at the small end or at the gauging notch when screwed together tight by hand. (2 times Column 6 + .0010.)

†Maximum amount it is possible for new working plug or ring gauges which come within specified tolerances to vary from being flush at the small end or at the gauging notch when screwed on reference gauge tight by hand. $\frac{\text{(Column 4 + Column 8.)}}{2}$

TABLE 10 CORRECTIONS IN DIAMETER FOR ERRORS IN ANGLE

A = Error in half included angle of thread expressed in minutes.

$$\text{Correction in diameter} = \frac{1.323P \sin A}{\sin (60° \times A)}$$

Correction in diameter

A	8 thds.		11½ thds.		14 thds.		18 thds.		27 thds.	
	Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.	Inches	M.M.
1′	.000056	.0014	.000039	.0010	.000032	.0008	.000025	.0006	.000017	.0004
2′	.000112	.0028	.000078	.0020	.000064	.0016	.000050	.0013	.000033	.0008
3′	.000168	.0043	.000117	.0030	.000096	.0024	.000075	.0019	.000050	.0013
4′	.000224	.0057	.000156	.0040	.000128	.0033	.000099	.0025	.000066	.0017
5′	.000279	.0071	.000194	.0049	.000160	.0041	.000124	.0031	.000083	.0021
6′	.000335	.0085	.000233	.0059	.000192	.0049	.000149	.0038	.000099	.0025
7′	.000391	.0099	.000272	.0069	.000223	.0057	.000174	.0044	.000116	.0029
8′	.000447	.0114	.000311	.0079	.000255	.0065	.000199	.0051	.000132	.0034
9′	.000503	.0128	.000350	.0089	.000287	.0073	.000223	.0057	.000149	.0038
10′	.000558	.0142	.000388	.0099	.000319	.0081	.000248	.0063	.000165	.0042
11′	.000614	.0156	.000427	.0108	.000351	.0089	.000273	.0069	.000182	.0046
12′	.000670	.0170	.000466	.0118	.000383	.0097	.000298	.0076	.000198	.0050
13′	.000725	.0184	.000505	.0128	.000415	.0105	.000322	.0082	.000215	.0055
14′	.000781	.0198	.000543	.0138	.000446	.0113	.000347	.0088	.000231	.0059
15′	.000837	.0213	.000582	.0148	.000478	.0121	.000372	.0095	.000248	.0063
16′	.000892	.0227	.000621	.0158	.000510	.0130	.000397	.0101	.000264	.0067
17′	.000948	.0241	.000660	.0168	.000542	.0138	.000421	.0107	.000281	.0071
18′	.001004	.0255	.000698	.0177	.000574	.0146	.000446	.0113	.000297	.0075
19′	.001059	.0269	.000737	.0187	.000605	.0154	.000471	.0120	.000314	.0080
20′	.001115	.0283	.000776	.0197	.000637	.0162	.000495	.0126	.000330	.0084
21′	.001170	.0297	.000814	.0207	.000669	.0170	.000520	.0132	.000347	.0088
22′	.001226	.0311	.000853	.0217	.000700	.0178	.000545	.0138	.000363	.0092
23′	.001281	.0325	.000891	.0226	.000732	.0186	.000570	.0145	.000380	.0097
24′	.001337	.0340	.000930	.0236	.000764	.0194	.000594	.0151	.000396	.0101
25′	.001392	.0354	.000969	.0246	.000796	.0202	.000619	.0157	.000413	.0105
26′	.001448	.0368	.001007	.0256	.000827	.0210	.000643	.0163	.000429	.0109
27′	.001503	.0382	.001046	.0266	.000859	.0218	.000668	.0170	.000445	.0113
28′	.001559	.0396	.001084	.0275	.000891	.0226	.000693	.0176	.000462	.0117
29′	.001614	.0410	.001123	.0285	.000922	.0234	.000717	.0182	.000478	.0121
30′	.001669	.0424	.001161	.0295	.000954	.0242	.000742	.0188	.000495	.0126
45′	.002498	.0634	.001738	.0441	.001427	.0362	.001110	.0282	.000740	.0188
60′	.003322	.0844	.002311	.0587	.001909	.0482	.001477	.0375	.000984	.0250

TABLE 11 CORRECTION IN DIAMETER FOR ERRORS IN LEAD

E = error in lead
Correction—1.732 E.

Error in Lead	CORRECTION IN DIAMETER									
	.00000	.00001	.00002	.00003	.00004	.00005	.00006	.00007	.00008	.00009
.00000	.00000	.00002	.00003	.00005	.00007	.00009	.00010	.00012	.00014	.00016
.00010	.00017	.00019	.00021	.00023	.00024	.00026	.00028	.00029	.00031	.00033
.00020	.00035	.00036	.00038	.00040	.00042	.00043	.00045	.00047	.00048	.00050
.00030	.00052	.00054	.00055	.00057	.00059	.00061	.00062	.00064	.00066	.00068
.00040	.00069	.00071	.00073	.00074	.00076	.00078	.00080	.00081	.00083	.00085
.00050	.00087	.00088	.00090	.00092	.00094	.00095	.00097	.00099	.00100	.00102
.00060	.00104	.00106	.00107	.00109	.00111	.00113	.00114	.00116	.00118	.00120
.00070	.00121	.00123	.00125	.00126	.00128	.00130	.00132	.00133	.00135	.00137
.00080	.00139	.00140	.00142	.00144	.00145	.00147	.00149	.00151	.00152	.00154
.00090	.00156	.00158	.00159	.00161	.00163	.00165	.00166	.00168	.00170	.00171
.00100	.00173	.00175	.00177	.00178	.00180	.00182	.00184	.00185	.00187	.00189
.00110	.00191	.00192	.00194	.00196	.00197	.00199	.00201	.00203	.00204	.00206
.00120	.00208	.00210	.00211	.00213	.00215	.00217	.00218	.00220	.00222	.00223
.00130	.00225	.00227	.00229	.00230	.00232	.00234	.00236	.00237	.00239	.00241
.00140	.00242	.00244	.00246	.00248	.00249	.00251	.00253	.00255	.00256	.00258
.00150	.00260	.00262	.00263	.00265	.00267	.00268	.00270	.00272	.00274	.00275
.00160	.00277	.00279	.00281	.00282	.00284	.00286	.00288	.00289	.00291	.00293
.00170	.00294	.00296	.00298	.00300	.00301	.00303	.00305	.00307	.00308	.00310
.00180	.00312	.00313	.00315	.00317	.00319	.00320	.00322	.00324	.00326	.00327
.00190	.00329	.00331	.00333	.00334	.00336	.00338	.00339	.00341	.00343	.00345
.00200	.00346	.00348	.00350	.00352	.00353	.00355	.00357	.00359	.00360	.00362

No. 1734

REPORT[1]

OF THE

JOINT CONFERENCE COMMITTEE

THE AMERICAN SOCIETY OF CIVIL ENGINEERS
THE AMERICAN INSTITUTE OF MINING AND METALLURGICAL ENGINEERS
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, AND
THE AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS.

AN informal meeting of the Conferees of the Development Committees of the four Societies was held in New York on July 2, 1919, at which the scope of the work was considered, and a Sub-Committee on Procedure, consisting of the chairmen of the Conferees from each of the four Societies, was appointed to arrange the program for and the time and place of the first formal meeting. The Conferees subsequently adopted the name, "The Joint Conference Committee."

2 The Joint Conference Committee convened at the Hotel Montclair, Montclair, New Jersey, August 13, 14 and 15, and in New York, September 15, 16 and 17, 1919.

3 The Committee has divided its report into two parts, that covering the public activities and that covering the society activities of common interest to the four Societies.

1 COÖPERATION IN PROFESSIONAL ACTIVITIES

4 It was the unanimous opinion of those present at all conferences that one of the most important matters on which joint action can be taken is the formation of a single comprehensive organization to secure united action of the engineering and allied technical professions in matters of common interest to them.

5 In presenting its report upon a proposed form of organization which shall provide a voice to speak for the ideals of the profession, a hand to enforce unity of action and render the maximum of national, social and political service, the Committee presents for consideration the following plan. The Committee emphasizes that the plan can only be valuable and enduring as the motive dominating it is patriotic, broad-visioned and unselfish, but firmly believes that all efforts toward bringing a united profession to the service of the Nation, the State and the City will bring to its members the public honor, esteem and recognition which their qualifications deserve, now relatively unacknowledged and uncompensated.

[1] Received by the Council, January 24, 1920, and ordered printed.

6 In preparing this plan the Committee has recognized that there exists in Engineering Council a tool which is engraving an honorable record on the pages of professional history, but its limitations are well known and its poverty is chronic. If desired, Engineering Council can be molded into this organization by making it more democratic and founding it on direct representation of all engineers, rather than appointment as at present.

7 The great object is to provide an effective body, widely and truly representative, modestly yet adequately financed, which will be neither autocratic nor aristocratic, which will at all times stand as the representative and defender of the profession in matters affecting its honor, welfare and common interest.

8 The mandate for a vehicle to provide for coöperation and solidarity among engineers has been unmistakably expressed by the membership of the four Societies, through their several Development Committees. In obedience thereto the Joint Conference Committee has constructed a plan for an organization designed to perform this function by providing an opportunity to use the strength of every existing technical organization in the country, but without taking from them any of their present privileges, or in any way interfering with their respective spheres of usefulness.

9 Every new activity, to be effective, means work and needs money. A movement such as proposed, if undertaken without sufficient funds, would not only prove barren of results, but by failure would bring ridicule upon the profession. In advance of its organization no definite budget can be prepared. The revenues proposed are moderate, and are based on the experience of Engineering Council, which, while called upon to occupy an ever-widening field, has been continually handicapped by limited resources.

10 In submitting the proposed fundamentals to govern a single comprehensive organization by which the engineering and allied technical professions may become more active in their service to the public and themselves, the Joint Conference Committee would point out the following as among some of the objects to be attained by such an organization:

a To render the maximum of service to the nation through unity of action

b To give the engineers of the country a more potent voice in public affairs

c To secure greater recognition of the services of the engineer, and to provide for his advancement

d To promote *esprit de corps* among the members of the profession

e To provide the machinery for prompt and united action on matters affecting the profession, among which are:

Licensing and registration of engineers
National Service Committee
National Department of Public Works
Conservation of national resources
Publicity
Classification and compensation of engineers
General employment bureau
Engineering education
International affiliation of engineers
Industrial relations.

11 In the greater vision resulting from the world war it is apparent that no one alone can successfully solve the problems with which he is confronted. He needs the coöperation of his fellows. National problems demand united effort. The Joint Gonference Committee believes that the greatest value of the proposed organization is in the united effort for the service of the nation, from which effort will result the greatest service to the individual.

12 The general plan offers a definite method of organization for public service. The Committee presents a means whereby the expressed wish of engineers throughout the land may be formulated in organization which shall make for the common weal.

13 The Committee submits the plan with the confident belief that it will be accepted by the four Societies, that they will make the sacrifices necessary and assume the responsibility of leadership, supporting the movement by virtue of their position in the engineering profession. A national movement by local societies is sure to come. The four National Societies can take the initiative and continue their leadership in American engineering by prompt action, or by inaction lose the prestige they now hold.

NATIONAL ENGINEERING ORGANIZATION[1]

ORGANIZATION AND PURPOSE

14 In order that the engineering and allied technical professions may become a more active national force in economic, industrial and civic affairs, and in order that united action may be facilitated, it is desirable that there be coöperation through a single comprehensive organization. The purpose of such organization should be to further the public welfare wherever technical knowledge and training are involved, and to consider all matters of common concern to these professions. It should embrace:

COMPONENT PARTS OF ORGANIZATION

15 *First.* Local Affiliations, preferably under the auspices of local engineering societies or clubs, as follows:

a "Local Associations" or "Sections" of the national engineering or technical societies;

b Local engineering societies; and

c Other local engineers and members of allied technical professions and associates.

16 *Second.* A National Council, consisting of representatives of national engineering and technical societies and of representatives of local, state or regional affiliations or organizations.

17 The formation of State Councils, composed of representatives of the local affiliations within the State or otherwise representative of the majority of the engineers and members of allied technical professions in the State, is desirable as conducive to coöperation and to further the objects of the National Council with which such State Councils should harmonize; and such State Councils are recommended wherever and whenever the local conditions warrant.

[1] The Joint Conference Committee does not deem it desirable to select definite names for either the organization or its component parts, but believes that this should be left to the action of this organization when formed.

DELIMITATION OF AUTHORITY

18 Local Affiliations, State Councils (where formed) and the National Council, shall take action on local, state and national matters respectively, and they shall be autonomous with respect thereto. It shall, however, be the duty of the National Council to interest itself in the activities of Local Affiliations and State Councils if such activities are of national scope or affect the interest of the engineering and allied technical professions in other parts of the country; provided, that nothing herein stated shall be construed as preventing the discussion by any Local Affiliation or State Council or by the National Council of any matters of interest to engineers and members of allied technical professions.

LOCAL AFFILIATIONS

PURPOSE

19 Local Affiliations are created to consider local matters of public welfare with which the engineering and allied technical professions are concerned, as well as other matters of common interest to these professions, in order that united action may be made possible in local matters.

CONSTITUTION

20 Each Local Affiliation shall submit its constitution and by-laws and all subsequent modifications thereof to the Executive Board of the National Council for approval.

STATE COUNCILS

PURPOSE

21 State Councils may be created to consider state matters of public welfare with which the engineering and allied technical professions are concerned, as well as other matters of common interest to these professions, in order that united action may be made possible in state affairs.

CONSTITUTION

22 Each State Council shall submit its constitution and by-laws and all subsequent modifications thereof to the Executive Board of the National Council for approval.

NATIONAL COUNCIL

PURPOSE

23 The National Council is created to consider national matters of public welfare with which the engineering and allied technical professions are concerned, as well as other matters of common interest to these professions, in order that united action may be possible in matters of national scope.

BASIS OF REPRESENTATION

24 Each local, state or regional affiliation or organization whose membership is not otherwise represented than through the national engineering or tech-

nical societies, shall be entitled to one representative to the National Council for a membership of from 100 to 1000 inclusive, and one additional representative for every additional 1000 members or major fraction thereof.

25 Each national engineering or technical society shall be entitled to one representative for a membership of from 200 to 2000 inclusive, and an additional representative for every additional 2000 members or major fraction thereof.

MEETINGS

26 The National Council shall hold a stated annual or biennial meeting. Other meetings *may* be called by the Executive Board upon its own initiative, and *shall* be called by it upon the written request of 25 delegates to the National Council, provided that all notices of special meetings shall be mailed not less than 60 days prior to the date thereof.

TERMS OF DELEGATES

27 Delegates shall serve for terms of four years; provided that arrangements shall be made by the Executive Board so that an approximately equal number of delegates will be elected each year as provided in the by-laws.

NATIONAL HEADQUARTERS AND SECRETARY

28 The National Council shall maintain National Headquarters with a permanent Secretary appointed by and holding office during the pleasure of the Executive Board. He shall not be a member of the Executive Board.

ELECTED OFFICERS

29 The following officers shall be elected by the National Council: A President to hold office for two years, who shall be ineligible for immediate re-election; two Vice-Presidents to hold office for four years, one to be elected every two years, and a Treasurer to hold office for two years.

EXECUTIVE BOARD

30 There shall be an Executive Board which shall direct the activities of the National Council in accordance with its adopted policies. It shall have such other functions as may be assigned to it by the National Council.

31 The Executive Board shall consist of the four officers elected by the National Council and one representative for each national society of 2000 members or less, with one additional representative from each national society for every additional 2000 members or major fraction thereof, to be selected from the membership of the National Council in each case by the national society concerned, and of a number of representatives selected from the membership of the National Council by the representatives therein of the local, state or regional affiliations or organizations, the said number to bear the same ratio to the representation of the national societies as the total membership of the local, state or regional affiliations or organizations bears to that of the national societies;

provided that the numerical basis of representation shall be so changed from time to time that the membership of the Executive Board shall not exceed thirty (30).

32 The President and Secretary of the National Council shall be the Chairman and Secretary, respectively, of the Executive Board.

ELECTORAL DISTRICTS

33 The National Council shall divide the country into such districts or regions, as may be desirable to provide for the election of the district members of the Executive Board. In establishing these districts due regard shall be had to geographical conditions and membership.

ADMISSION TO NATIONAL COUNCIL

34 Any national engineering or technical society or any local, state or regional affiliation or organization desiring representation on the National Council shall submit a written request to the Secretary of the National Council which shall be accompanied by such data regarding the aims and status of the organization as the by-laws may provide.

35 The Secretary shall refer this request to the Executive Board which shall submit it to the delegates, together with its recommendations, for a letter ballot.

36 The applicant shall be admitted by a majority vote of the National Council, provided that not more than 25 per cent thereof shall vote in the negative.

UNEXPIRED TERMS

37 Vacancies in the offices of the President, the Vice-Presidents, the Treasurer and in the Executive Board and delegates shall be filled as soon as feasible, by the agencies originally selecting the incumbents. Officers and delegates thus chosen shall serve for the unexpired terms.

FINANCES

38 For the purpose of financing the National Council and its Executive Board (not for the use of local affiliations and State Councils) the following assessments shall be made.

39 Each national society represented on the National Council shall contribute annually one dollar and fifty cents ($1.50) per member.

40 Each local, state or regional affiliation or organization represented on the National Council shall contribute annually one dollar ($1.00) per member.

2 COÖPERATION IN SOCIETY ACTIVITIES

41 Careful consideration was given to many points of common interest, and in the time available the Committee feels warranted in taking unanimous action on some of these topics, as follows:

COÖPERATION IN PROFESSIONAL ACTIVITIES

42 The Committee recommends that the growing coöperation in the activities of the Societies be fostered. The unified operation of the libraries, the organization of Engineering Council, American Engineering Standards Committee, Engineering Foundation and Employment Service, are conspicuous examples of such common action. Joint committees for definite purposes should be established, and standing or other committees directing similar activities in the several societies would profit by conference. In this way common action may be taken or similar policies may be adopted when advantageous, thus avoiding duplication of effort. Comprehensive means should be provided for united action in other larger professional and technical matters.

JOINT MEETINGS

43 The Committee is of the opinion that periodic Joint meetings of the four National Societies in various localities would be beneficial in the development of social intercourse between the members and for debating matters of common interest. Furthermore such large conferences would bring the engineer forcibly to the attention of the general public.

INDUSTRIAL RELATIONS

44 The question of industrial relations is one of the most pressing issues of the present day. It is a vital factor in the economic future of the country. It is a matter with which the engineer is peculiarly fitted to deal. In view of the great importance of the subject, and of the unique relation of the engineer to industry, the Joint Conference Committee recommends that Industrial Relations should be a major subject for the consideration of the engineering and allied technical professions. It is also recommended that support be given to educational movements, such as the Industrial Service of the Y. M. C. A. in colleges, the Americanization movement, and similar activities.

PERSONAL SERVICE BUREAUS

45 It is recommended that the Engineering Societies Employment Bureau be extended by providing for coöperation with the existing organizations of engineers maintaining employment service, and that upon the formation of the contemplated national organization the service be made available to the members of all constituent associations and branch agencies be established in communities where the local engineers desire to coöperate.

STUDENT ORGANIZATIONS

46 The Committee recommends that it be the duty of local associations and sections of the National Societies to promote and assist general student engineering societies in neighboring universities and technical schools, to provide carefully selected speakers, and to maintain personal contact with such societies. In view of the desirability of more intimate coöperation among engineers, consideration of the affiliation of the student branches of the national Societies into general student engineering associations is recommended.

THE AMERICAN ENGINEER IN FOREIGN SERVICE

47 It is the opinion of the Committee that American engineers in foreign countries should be urged to assemble for purposes of better acquaintance and mutual help, and to fraternize with the engineers of the country in which they reside, for the promotion of world-wide coöperation.

STANDARDS

48 The Committee is unanimously of the opinion that the co-relation of standards by the four Societies and others working with them, along the lines contemplated by the American Engineering Standards Committee, should be approved in principle; but in view of the fact that this matter is under consideration by the Boards of Direction of those Societies and the matter is well advanced, this Committee does not feel warranted in making a further recommendation.

ARBITRATION AND EXPERT TESTIMONY

49 The Committee is of the opinion that the recommendation of the Committee on Development of the American Society of Civil Engineers in regard to arbitration and expert testimony, namely, that a committee of the American Bar Association be asked to coöperate with a joint committee of the four Societies in order to develop better practice in these matters, should be referred to Engineering Council.

LICENSING AND REGISTRATION OF ENGINEERS

50 The Committee of Engineering Council that has had under consideration the licensing and registration of engineers has nearly completed its draft of a standard form for a uniform law on these matters. Since it is undesirable to duplicate this work, the Joint Conference Committee recommends that action be deferred pending the receipt of that report.

PATENT LAW

51 Since the Patents Committee of Engineering Council, after due consideration, has approved the report of the Patents Committee of the National Research Council, in which definite recommendations are made in regard to the present patent system, it is deemed inexpedient for the Joint Conference Committee to take action on the subject of Patents and the Patent Law.

CODE OF ETHICS

52 The Joint Conference Committee, after discussion of the question of a code of ethics, reached the conclusion that the matter would require extensive study and careful consideration in order to secure joint action by the representatives of the four Societies, and therefore deemed it unwise to attempt to make a report on this matter at this time.

CONCLUSIONS

53 The Joint Conference Committee was confronted with a multiplicity of matters inviting consideration, but in the two and a half months of its existence it has been able to consider only the more important ones. The vital matter is the general organization of engineers for public activities, a general plan for which is now submitted for the approval of the Societies.

54 When the four Societies shall have approved the plan, the Committee holds itself in readiness to work out the details by which it may be put into effect. The Committee has therefore adjourned to meet after this matter has been considered by the four Societies.

George G. Anderson Gardner S. Williams Richard L. Humphrey, *Chairman*	Conferees of the Committee on Development, The American Society of Civil Engineers.
Arthur S. Dwight Philip N. Moore John V. W. Reynders, *Chairman*	Conferees of the Committee on Development, The American Institute of Mining and Metallurgical Engineers.
Edwin S. Carman Dexter S. Kimball Louis C. Marburg, *Chairman*	Conferees of the Committee on Aims and Organization, The American Society of Mechanical Engineers.
Lewis T. Robinson Charles F. Scott Calvert Townley, *Chairman*	Conferees of the Committee on Development, The American Institute of Electrical Engineers.

Note. — In addition to the above the following members have been in attendance at meetings of the Committee, but were unable to participate in the final drafting of the report: Joseph W. Richards and Allen H. Rogers, the American Institute of Mining and Metallurgical Engineers, and Charles E. Lord, The American Society of Mechanical Engineers.

There was also present by invitation Comfort A. Adams, Engineering Council.

New York, N. Y., September 17, 1919.

No. 1735

NECROLOGY[1]

LUCAS NICHOLAAS ALTA

Lucas Nicholaas Alta, a member of the Society since 1896, died at Watergraafsmeer, Holland, December 2, 1919. Mr. Alta was born in Holland in 1859 and received his education at the School of Technology, Amsterdam. Following his graduation, he obtained drawing-room and shop experience in Amsterdam and was for a time assistant engineer on board a steamship. From 1882 to 1896 he was connected with the W. C. & K. DeWitt Engineering Works in Holland as chief erector and draftsman. From 1896 until the time of his death he was mechanical engineer and a member of the firm of L. N. Alta & Co., of Amsterdam.

DANIEL ASHWORTH

Daniel Ashworth, who died November 8, 1919, was born in Lancashire, England, in 1841. At the age of nine he came to this country with his father settling in Pittsburgh, Pa., where he attended the public schools. He entered the mechanical field as a glass mold maker and later became master mechanic of glass works in Pittsburgh and Boston. In 1872 he became the manager of the Hemingray Glass Company, Covington, Ky., a position which he held for ten years, during which time he made a thorough study of steam. For several years he was general superintendent of the Lane & Bodley Co. of Cincinnati. In 1885 he returned to Pittsburgh as consulting engineer and steam expert.

Mr. Ashworth retired from the engineering profession in 1906, having been honored by President Roosevelt with the appointment as United States Pension Agent of Pennsylvania; later he was reappointed by President Taft. He became a life member of the Society in 1885. He was also a member of the Engineers Society of Western Pennsylvania and was prominent in fraternal, civic and political affairs throughout western Pennsylvania.

WILLIAM FRANKLIN AUSTIN

William F. Austin, a member of the Society since 1905, died on July 12, 1919. Mr. Austin was born on October 2, 1864, in East Greenbush, N. Y. He received his early education in the public schools of Rensselaer and later supplemented this by a correspondence course in mechanical engineering. He served his apprenticeship with the Central Bridge Company, Buffalo, from 1882 to 1884, when he became connected with the Hilton Bridge Company, Albany, first as foreman of their drafting room and later of the erection department. In 1894, Mr. Austin became superintendent of the Albany plant of the American Bridge Company, and in 1905 superintendent of the Eddystone plant of the Belmont Iron Works, where he was located for about eight years, when illness compelled his giving up active business life.

[1] Those members who died while in military or naval service have asterisks affixed to their names.

ABRAM T. BALDWIN

Abram T. Baldwin was born on September 26, 1870, in Yonkers, N. Y., and was educated in the public schools and Cornell University, graduating from the latter with the class of 1893.

Upon graduation Mr. Baldwin first worked in the Wm. A. Sweet Rolling Mills, Syracuse, N. Y., and served his apprenticeship in practically every branch of the industry. In January 1895 he became connected with the Solvay Process Co., also in Syracuse, where he worked through the various departments, being assistant manager of the soda ash department at the time he left Syracuse to enter the coke department of the same company in Detroit. In May 1910 the Precision Instrument Co. was organized, in which Mr. Baldwin was very much interested. It was not unti 1911, however, that he gave this company his full attention and at that time he became its treasurer; in 1913 he was made president and general manager of the firm, which position he was holding at the time of his death.

When the United States entered the war, Mr. Baldwin was asked to take up the manufacture of air-speed indicators for the Science and Research Division of the Bureau of Aircraft. His work in this connection was so arduous that it resulted in his physical breakdown and sudden death from heart collapse on January 8 in Boston.

Mr. Baldwin was greatly interested in the combustion of coal and efficiency of boiler operation. He was a member of the Detroit Engineering Society, the National Association of Stationary Engineers, the American Gas Institute and the Detroit Board of Commerce. He became a junior member of the Society in 1899 and in 1902 a life member.

CHARLES MILTON BALDWIN *

Charles M. Baldwin, Lieutenant, U. S. Navy, was born on March 26, 1876, in Mt. Vernon, Tex. He received his early education in Rockport, Tex., and later attended the Southwestern University at Georgetown.

From 1893 to 1897 he worked as apprentice machinist with the International and Great Northern Railroad Shops, when he became third assistant marine engineer of the Mallory Steamship Line. He served as second and first assistant, respectively, with the Red D and Southern Pacific lines. From 1900 to 1903 he was engaged as erecting and operating engineer with the J. G. White Co., San Juan, Porto Rico.

In 1903 Lieutenant Baldwin entered the service of the U. S. Navy as machinist, serving in this capacity for about four years, when he was promoted to the grade of warrant machinist, with a chief engineer's unlimited ocean license. He was a graduate of the State Steam Engineering School of Boston, Mass., as operating engineer.

When the United States entered the war, he offered his services and was commissioned as a lieutenant in the Navy. He died at sea on March 12, 1919.

Lieutenant Baldwin was considered an authority on operating turbines and express-type boilers. He was a member of the National Marine Engineers' Beneficial Association and belonged to a number of fraternal organizations. He became an associate-member of the Society in 1918.

J. SELLERS BANCROFT

J. Sellers Bancroft, a member of the Society since 1880, manager from 1909 to 1911, and vice-president from 1915 to 1917, died at his home in Philadelphia on January 29 after an illness of several months.

Mr. Bancroft was born in Providence, R. I., on September 12, 1843. His father, Edward Bancroft, a distinguished engineer, who made many inventions of machine tools and shafting appliances, founded the firm of Bancroft and Sellers, now Wm. Sellers & Co., Inc., Philadelphia. As a sidelight on the growth of the machine-tool industry in this country, it is interesting to note that Edward Bancroft built the first metal planer in the United States, the bed and table being chipped and filed by hand.

J. Sellers Bancroft was educated in Philadelphia, being graduated from the high school of that city in 1861. In February of that year he entered the employ of Wm. Sellers & Co., being apprenticed to his uncle, Wm. Sellers. In 1863, at the age of twenty, he was made gang foreman, and in 1866 general foreman. He was admitted to the firm in 1873, and when the business was incorporated in 1887 he was made general manager.

While with the Sellers Co. he was granted many patents on machine tools, injectors, shafting appliances, power cranes and their interlocking electrical devices.

For forty-one years he remained with this company, leaving in 1902 at the age of fifty-nine to become general manager and mechanical engineer of the Lanston Monotype Machine Co., Philadelphia. In the following sixteen years of his life he did his greatest engineering work; the perfection of the monotype and the special machines for making the molds and matrices used with it for casting and composing type in automatically justified lines. The organization of the monotype factory exemplifies his work as an executive and a leader of men.

A keen lover of books, it was a great joy to him that his work did much to raise the quality and reduce the cost of printing not only in the United States but also throughout the world.

At the time of his death he was vice-president and treasurer of the Monotype Co., as well as its general manager and mechanical engineer.

More than 100 patents testify to the scope of Mr. Bancroft's engineering work and the range of his creative ability. To those who worked with him on the solution of any of the many problems that engaged his attention, his thoroughness, the result of his never-failing perseverance and patience will be an ever-inspiring memory.

JAMES ALEXANDER BARR

James A. Barr was born on December 17, 1890, at Newburgh, N. Y. He was educated in the public schools of that city and was graduated from the Newburgh Academy in 1908.

The following autumn he entered the General Electric Works, Schenectady, where he took a mechanical course in connection with Union University. Leaving there in about eighteen months, he was employed by the Alberger Pump Co., Newburgh, N. Y., as draftsman. He was next connected with the Ridgway Dynamo & Engine Co., Ridgway, Pa., and following that he was with the Landis Machine Co., Waynsboro, Pa. He then went to New York City with the Standard Galvanizing Company. For a short period he was in Lawrence, Mass., as assistant engineer in a large pump-manufacturing concern, resigning to take a position with the Kennedy Stroh Corporation, Pittsburgh, Pa., afterwards taken over by the Edgewater Steel Co., Oakmont, Pa., where his accuracy and ingenuity won him rapid promotion.

Mr. Barr died at Oakmont, Pa., on December 11, 1918. He became a junior member of the Society in 1916.

EDWARD PAYSON BATES

Edward Payson Bates, a life member of the Society, died on August 3, 1919, in Butte, Mont. Mr. Bates was born on March 3, 1844, in Savannah, Ga. His parents were of New England stock; his father, Levi Whitcomb Bates, was a descendant of Clement Bates, who came to this country from England with his brother Joseph. They, in turn, were descendants of John Bate (or Bates) of Lydd, England, who was a mayor of the town and justice in the court. His mother was Ruth Ann Bailey. She was born in Meredith, Delaware County, N. Y. Her father, Timothy Bailey, was a skillful mechanic and inventor, having invented the knitting frame which made underclothes and made radical changes in the spinning jenny which tended to save time and decrease the cost of producing cloth. This knitting frame was first used in Cohoes, N. Y., and is in use there to this day.

Mr. Bates was a born mechanic, his ideas running in that direction from the time he could handle a hammer and saw, which was very early in his experience. At the age of seventeen he entered the machine shop of Hobart B. Bigelow, in New Haven, who was afterward one of the beloved governors of the State of Connecticut. He remained there only a short time when he moved to New York State in the vicinity of Albany and entered a machine shop. After two years he could do all the ordinary work skillfully and correctly. He received a journeyman's wages for the second year of his experience, during that time learning to assemble a locomotive and operate it on the road. His next move was to New York City where he learned how to build a marine engine and install it in a ship. He studied at nights with a professor connected with Cooper Institute, was granted his marine license and for several years went to sea as an engineer.

During the Civil War he saw service on a transport which was bearing home the wounded from Libby Prison. Soon after this, Willis Warner, widely known in engineering circles, his childhood friend, induced him to enter his employ to learn how to erect steam-heating apparatus, with the idea of opening a branch business in Syracuse, N. Y. That business he was thereafter engaged in, the character changing much from time to time. He added to it ventilation, hot-water heating, factory equipment, power plants, sprinkler work, and various contracts of a similar nature. He was in Mr. Warren's employ for several years when the latter's sudden and unexpected death threw him on his own resources. He was enabled by fortuitous circumstances to buy out the business and it was known thereafter as Bates & Johnson Co. Upon taking over the business he increased the number of branches until at one time there were offices of the concern in eleven cities. As conditions changed and trade unions made drastic rules he found it advisable to cut off business conducted at a distance and limit himself to two offices — the principal one at Syracuse and a branch at Utica. At the beginning of 1917 Mr. Bates incorporated the business under the laws of New York State and was the new company's first president, Mrs. Bates being one of the directors. The company proposed to carry on the same line of business which Mr. Bates, himself, organized and maintained.

Mr. Bates was a member of the Society of Naval Architects and Marine Engineers, charter and life member of the Technology Club of Syracuse, a life member of the Mayflower Society of Massachusetts, life director of the American Bible Society, life member of the Archæological Institute of America, member of the Robert Fulton Memorial Association, life member of the Bates Association and director of the Syracuse Museum of Fine Arts. In 1910, Mr. Bates was appointed official delegate of the Society of Naval Architects and Marine Engineers to the Fiftieth Anniversary of the foundation of the Institution of Naval Architects, which was held in London.

In the middle of March, 1919, Mr. and Mrs. Bates started on an extended trip that took them southward to New Orleans by steamer, through Texas into Southern California, north to Alaska, then through Yellowstone Park; a singularly happy and delightful trip which was interrupted in Butte, Mont., by his peaceful death.

WILLIAM CLYDE BAXTER

William C. Baxter was born in Thayer, Kan., in November, 1889. He was graduated from the Kansas State Agricultural College in 1913, receiving his B.S. in mechanical engineering.

Upon graduation he became associated with the Wichita Pipe Line Co., in special engineering work and field testing. From January 1914 to May 1914 he was with the Wichita Natural Gas Co., working on gas sampling and analysis. For a short time he was connected with the Quapaw Gas Co. in orifice-meter experimental work. In June 1915, Mr. Baxter returned to the Wichita Pipe Line Co. to do special research and experimental work. At the time of his death in November 1918, he was superintendent of the meter department and school of instruction for the company.

Mr. Baxter became a junior member of the Society in 1916.

CARL BEHN

Carl Behn was born in Hamburg, Germany, on August 2, 1858. He received his early education in Karlsruhe, Germany, and his technical training as mechanical engineer at the Polytechnic Institute in that city. After completing his education he traveled extensively in Europe and came to the United States about 1878 when he entered the employ of the Howard Iron Works, Buffalo, N. Y., later becoming superintendent of the company.

In 1887 he acquired an interest in the Buffalo Refrigerating Machine Co., which was founded by his father in 1881. In 1904 the main office of the company was moved to Harrison, N. J., and later to New York City. Mr. Behn at this time became the active head of the business.

Mr. Behn was one of the pioneers in building refrigerating machinery in this country and his training, experience and keen judgment made him an authority in refrigerating engineering.

Mr. Behn became a member of the Society in 1903. He died on November 3, 1918.

WILLIAM A. BOLE

William A. Bole, Assistant to Vice-President H. T. Herr of the Westinghouse Electric & Manufacturing Co., Machine Works, died at his home in Pittsburgh on June 16. Mr. Bole was born on July 12, 1859, in Pittsburgh, Pa. He received his early education in the schools of Allegheny, later attending the University of Western Pennsylvania, now the University of Pittsburgh.

In 1878 he left school and was employed by his father who at that time was a manufacturer of steam engines in Pittsburgh. Here he gained considerable information regarding the manufacture of the mechanical parts of steam engines. After serving his father for three years he entered the employ of the Westinghouse Machine Co. in 1882 in the capacity of cost and time clerk. In 1883 he became foreman of the company and the following year was appointed Superintendent of Works and Purchasing Agent and in 1900 Manager of Works. In 1906 he was advanced to the position of Consulting Engineer and in 1908 became General Manager. In 1914 he was appointed Vice-President in Charge of Manufacture for the Machine Works and Trafford City Foundry. He was a veteran of thirty-seven years with the company.

Mr. Bole was a member of the Engineers' Society of Western Pennsylvania, of which he was president for one year, and of the American Foundrymen's Association. He became a member of our Society in 1887.

STEPHEN P. BROWN

Stephen P. Brown, who, until June 1919, was vice-president and general manager of Ford, Bacon & Davis, Engineers, New York, met his death on Saturday, December 6, 1919, by drowning at Sebec Lake, Dover, Me., when he broke through the ice on which he and his nine-year-old son were crossing to a landing. His delay in seeing his son to safety cost him his own life.

Mr. Brown was born in Dover, Me., on April 29, 1877. He was educated in the town schools, Foxcroft Academy, Hotchkiss School and the Massachusetts Institute of Technology from which he was graduated in 1900. In that year he began his career as junior partner in the firm of Collier & Brown, consulting engineers, Atlanta, Ga. He remained with the firm until 1904 when he was made inspector of the Bridgeport Station Works of the N. Y., N. H. & H. R. R. From 1905 to 1909 he was with the United Engineering & Contracting Co., holding in succession positions of increasing importance, finally as general superintendent of all work west of Fifth Avenue on the Pennsylvania Railroad cross-town tunnels under New York. For the next three years he was chief engineer of the Tidewater Building Co., building a section of the Fourth Avenue Rapid Transit four-track subway in Brooklyn, N. Y.

From 1912 to 1917 Mr. Brown was chief engineer of the Mount Royal Tunnel & Terminal Co., Ltd., and managing engineer for Mackenzie, Mann & Co., Ltd., having charge of both design and construction of the terminal development in Montreal for the Canadian Northern Railway. From 1918 to June, 1919, Mr. Brown was vice-president and general manager of Ford, Bacon & Davis, Engineers, New York City. He left in June to take a long vacation in the Maine woods, where he met his tragic end.

Mr. Brown was a member of the American Society of Civil Engineers, the

American Railway Engineering Association, the Engineering Institution of Canada, the Institution of Civil Engineers (British), and of the Engineers' and the University Club of New York. He became a member of the Society in 1909.

ROLLA C. CARPENTER

Prof. Rolla C. Carpenter, who has been a prominent member of the faculty of Sibley College, Cornell University, since 1890, died at his home at Ithaca, N. Y., on January 19.

The following account of Professor Carpenter's life is taken from that prepared only a few months ago for the *Sibley Journal of Engineering* at the time when he was about to retire from active service at the University, to devote his attention to engineering and consulting work, particularly along the lines of coke manufacture and the recovery of by-products incident to that industry. For many years Professor Carpenter was one of the most actively participating members of The American Society of Mechanical Engineers, a frequent attendant at meetings and one whom a very large number who came to the meetings always wanted to see and talk with.

Professor Carpenter was born near Orion, Michigan, June 26, 1852. His father, Charles K. Carpenter, owned an extensive farm at this place and was also vice-president of a railroad running between Detroit and Bay City, which now forms part of the Michigan Central system.

He graduated from Michigan Agricultural College in 1873 and received the degree of Civil Engineer from the University of Michigan in 1875. He was then engaged as an instructor in the Michigan Agricultural College, at the same time doing graduate work, and received the degree of Master of Science in 1876. In 1878 he was elected professor of mathematics and civil engineering at the Michigan Agricultural College, which position he held until 1890. During part of this period he spent his vacations, which then came in the winter months, studying at other institutions. Part of this time was spent at the Massachusetts Institute of Technology, where he studied under Professors Peabody and Lanza, and part was spent at Cornell, where he received the degree of Master of Mechanical Engineering in 1888. He was greatly assisted in the preparation of his thesis for the M.M.E. degree by his connection with the Lansing Iron and Engine Company of Lansing, Michigan, as consulting engineer. This connection placed at his disposal the facilities of a large and up-to-date manufacturing plant which offered opportunities not then enjoyed by any of the technical schools. This thesis, which is now on file in the University library and which was reported upon by Dr. Thurston in a paper read before The American Society of Mechanical Engineers, was on the subject of Internal Friction in Non-Condensing Engines and, as shown by Dr. Thurston's discussion, played an important part in the entire revision of the ideas which then prevailed concerning steam-engine friction.

In 1890 Professor Carpenter was elected Associate Professor of Engineering at Cornell University and the laboratory work was organized as a separate department under his direction. In 1895 he was elected professor of experimental engineering, which position he held up to the time of his death.

Professor Carpenter's experience in the several leading educational institutions as well as his intimate contact with various industrial enterprises peculiarly fitted him for the work of building up a course of instruction in experimental engineering which has done much for the upbuilding of the reputation of Sibley College and which is regarded by many alumni as furnishing a most valuable part of their education.

This system has been copied with some modifications in many other colleges and technical schools and has no doubt had a pronounced influence upon the methods of teaching other sciences.

Professor Carpenter published his "Notes on Mechanical Laboratory Practice" in 1891. This was the basis of his later text book on "Experimental Engineering" which has been the leading manual in this country on the subject. The first edition of his book on heating and ventilation was published in 1895 entitled "Heating and Ventilating Buildings." This book has gone through six revisions and has had an extensive circulation. It contains much original material from the author's own experience and is much quoted by later writers on heating and ventilating. Professor Carpenter is also joint author with Professor Diederichs of a textbook on "Gas Engines." In addition to these books, he has made many contributions to engineering literature through various societies and publications, among which may be mentioned The American Society of Mechanical Engineers, the American Society of Civil Engineers, and the American Society of Heating and Ventilating Engineers.

Professor Carpenter held membership in eight of the leading engineering societies of America. He was vice-president of The American Society of Mechanical Engineers from 1908 to 1911 and served on various committees of this Society, perhaps the most important of which is the Boiler Code Committee. He was President of the American Society of Heating and Ventilating Engineers in 1898, was vice-president of the American Society of Automobile Engineers in 1910-12, and has taken an active interest in the student branch of The American Society of Mechanical Engineers at Cornell.

Professor Carpenter engaged in a diversified field of investigation and research, including problems relating to power plants, gas engines, cement manufacture, coke manufacture, railway management, heating and ventilating, etc. He was one of the leading patent experts in the country and was employed by many of the leading law firms in various parts of the United States. He invented a number of pieces of laboratory apparatus, such as the Carpenter coal calorimeter, which was for many years a standard for testing the heating value of coal, the throttling and separating steam calorimeters now extensively used, a friction testing machine which may be found in most of the large laboratories and an inertia governor for the steam engine.

Professor Carpenter was honored by appointment to various positions of distinction. He was judge of machinery and transportation at the Chicago Exposition in 1893, at the Buffalo Exposition in 1901, and at the Jamestown Exposition in 1907. He was a member of the commission appointed by the Academy of Science in 1915 at the request of the President of the United States to investigate the slides at the Panama Canal and to make such recommendations as in the judgment of the commission would improve the conditions and lessen the possibilities of slides in the future. He received the degree of Doctor of Laws in 1907 from the Michigan Agricultural College.

Professor Carpenter's kindly manner and genial disposition made it easy for even the most timid to approach him and he was never too busy to be considerate of anyone who sought his council and advice. His large and varied experience, coupled with good judgment and his extensive knowledge of the engineering profession and of human nature, made his counsel and advice exceedingly valuable to his colleagues as well as to students and the world at large.

WILLIAM FRAZIER CARPENTER

William Frazier Carpenter was born in 1857 near the place now known as Sailors' Snug Harbor, Staten Island, N. Y. He attended school at Tivoli, N. Y., and then entered Stevens Institute of Technology from which he was graduated in 1876.

His first position was with the Ramapo Iron Works, Hillburn, N. Y., and from there he went to Pittsburgh where he started the Kent Construction Co. and later the Pittsburgh Testing Laboratory. He then became connected with the Westinghouse Electric & Manufacturing Co., being superintendent of the East Pittsburgh works, manager of the Newark works and finally special representative in which capacity he was closely associated with the financing and engineering work of a number of large electric railway lines, central power plants, hydro-electric developments, etc. Later as vice-president and general manager of the St. Lawrence Power Co., he was responsible for the financing and building of what at that time was one of the largest hydro-electric developments in the country. As a consulting engineer, he later built a power house and dam in the Spokane River for the Spokane & Inland Empire Railway Co. He then became vice-president of the Pittsburgh Testing Laboratory, the same concern, which as a boy of twenty-one, he had started in Pittsburgh. He was holding this position at the time of his death, October 10, 1919.

Mr. Carpenter (who, in 1918, had his name changed from Zimmerman) was one of the original members of the Engineers' Club of New York. He became a member of the Society in 1884.

IRVING EMORY CENTER

Irving E. Center was born at Bridgewater, Mass., on March 9, 1891. He was graduated from the high school in Kingston, Mass., and then entered the University of Maine, from which he received his B. S. in mechanical engineering in 1912.

In the same year he entered the employ of the General Electric Co. at Lynn, Mass., as a student engineer in the turbine research department. He was later advanced to the position of assistant engineer in charge of testing and of the original design and construction of many pieces of novel machinery. In November, 1918, Mr. Center became connected with the Newburgh Shipyards, Newburgh, N. Y., and was with this concern at the time of his death, February 12, 1919.

Mr. Center became a junior member of the Society in 1916.

JAMES WEBSTER CULLEN *

James W. Cullen was born in Edgewater, a suburb of Chicago, Ill., on March 15, 1888. He obtained his early education in the public schools of Chicago and Hamilton, Ohio, later attending the Miami and Ohio State Universities.

He served his apprenticeship in the Niles Tool Works, Hamilton, and was then made assistant to the engineer in charge of power, light and heat.

He was the founder and vice-president of the Cullen & Vaughan Construction Co., Hamilton, which engaged in mill work and factory-building construction. His liking for the mechanical trades later led him to accept a position with the Midvale Steel Co., in whose plant at Nicetown he spent some time in acquiring a knowledge of steel production, when he was put in charge of the Pittsburgh sales office of the company. The last position which Mr. Cullen occupied was in the Sales Department of the Elwell-Parker Co., which he resigned to enter the Service as a first lieutenant in the Signal Corps of the Army, serving both in France and Italy. His death occurred very suddenly on June 28, 1919, shortly after his return to this country.

Mr. Cullen was a member of the Chamber of Commerce of Pittsburgh and of several clubs and fraternal organizations. He became an associate-member of the Society in 1916.

GEORGE RHODES CULLINGWORTH

George R. Cullingworth, vice-president of the Garvin Machine Co., New York, died on December 15, 1919. Mr. Cullingworth was born on August 25, 1837, in Manayunk, Pa., then a suburb of Philadelphia. He attended public school there until about sixteen years of age, when he entered business with his father in Philadelphia, where he served his apprenticeship.

From 1857 to 1870 he was connected for varying periods with the following firms: Bennett, Dougherty & Sons, Philadelphia; Colt's Armory, Hartford; the Starr Armory, Yonkers; the Trenton Arms Co., Trenton, and the E. E. Garvin Co., New York. In 1870 he formed a partnership with Mr. Sargent in the Hydraulic Machine Co., New York. During this period he worked on the erection of the elevated railroad in New York City, inventing the ticket chopper which is now in use. For twenty years Mr. Cullingworth was associated with the Ingersoll Rock Drill Co., New York, as vice-president and as president. He sold his interest in this firm to become connected with the Garvin Machine Co., with which concern he spent the last twenty-two years.

Mr. Cullingworth was a member of the General Society of Mechanics and Tradesmen of the City of New York. He became a member of our Society in 1884.

MARSHALL TEN BROECK DAVIDSON

Marshall T. Davidson was born on February 17, 1837 in Albany, N. Y. He was educated in the public schools of Hudson, N. Y., and the Hudson Academy. Later he attended the Albany Academy and Polytechnic Institute.

In 1857 he entered a marine-engine and machine shop in New York City and a short while afterwards went to sea as assistant engineer on ocean steamers running north from San Francisco to the various settlements on Puget Sound. In 1862 he accepted the position of second assistant engineer in the Naval service and later was promoted to the position of chief engineer on an Army transport which was being built at Wilmington, Del. He superintended the construction and installation of machinery on this vessel and upon its completion received a commission as chief engineer in the Revenue Cutter Service, with like duties on two vessels being built for this department.

At the close of the Civil War Mr. Davidson resigned from the service and started in business as contracting mechanical engineer, furnishing and installing the steam heating and elevator equipment at the main Post Office building, New York City.

In 1878 he started the construction of the Davidson steam pump and pumping engines, which machines have been extensively used in the Navy Department on battleships, destroyers, etc. He also installed under contract with the city of New York pumping engines for the Ridgewood, Milburn and Mt. Prospect Pumping Stations and various other small pumping stations throughout Long Island. At the time of his death, April 10, 1919, he was president of the M. T. Davidson Co., Brooklyn, manufacturers of steam pumps, pumping engines, condensers, evaporators, etc.

Mr. Davidson was a member of the Society of Naval Architects and Marine Engineers, the Naval Order of the United States, the Associate Society U. S. Grant Post, No. 327, the Columbia County Association and a life member of the Navy League. He became a member of our Society in 1886.

JOSEPH FRANZ DIEPENBROCK

Joseph F. Diepenbrock was born in Germany on August 28, 1874. He was educated in the public and high schools of Essen and attended the Fachschule at Hagen, taking the engineering course. He served a two-years' apprenticeship in pattern and machine shops and having completed the fundamental studies at this school, he entered the College Charlottenburg. He spent about four years there, the latter part of this period as assistant to Prof. A. Riedler.

In 1901 Mr. Diepenbrock came to the United States where he became associated with the Dickson Manufacturing Co., Scranton, Pa., as designing draftsman. In 1902 he became chief draftsman for the Allis-Chalmers Co., in Chicago; later he was transferred to Milwaukee as mechanical engineer of the pumping-engine department and on the work of standardization of all vertical and horizontal compound and triple pumping engines.

In 1910 he took charge of the drawing room of the H. R. Worthington Co. Works at Harrison, N. J. After seven years of intense and interesting activity Mr. Diepenbrock resigned from the company to become associated with Col. H. A. Allen in Chicago on engineering construction work for the City of Chicago. During the War he worked temporarily on war work with J. E. Russel & Co., the Tuthill Spring Co. and the Arnold Engineering Co. While in the service of the latter concern he assisted in some very important work in connection with Fort Sheridan and so earnestly did he devote himself to this task that his health failed and after a five-months' illness he died on April 7, 1919. Mr. Diepenbrock became a member of the Society in 1912.

WILLIAM W. DINGEE

William W. Dingee was born on January 5, 1831, in Byberry, Pa. He was educated in the Maryland Institute and also attended the School of Mechanical Arts connected with the Institute.

For thirty years Mr. Dingee was the proprietor and superintendent of a foundry and machine business, where he designed, built and operated many machines for general and special purposes. For many years he was associated with the J. I. Case Threshing Co., Racine, Wis., as machine designer and constructor, and while with this concern was the inventor of many improvements for threshing machines.

Mr. Dingee retired from active business in 1906 and the last years of his life were spent in Chicago where he died on May 25, 1919. He was one of the early members of the society, having joined in 1886.

GEORGE DINKEL

George Dinkel, a member of the Society since 1890, was killed in an automobile accident near Havana, Cuba, on January 21.

Mr. Dinkel was born on November 28, 1866. He was graduated from Stevens Institute of Technology, Hoboken, N. J., in 1888, when he became associated with the American Sugar Refining Company as assistant manager at the Jersey City plant. At the time of his death he was chief engineer of the company, having been in its employ for 31 years.

He attained distinction in his profession, and was well known in the sugar industry. He had been granted a number of patents for important inventions, especially in the line of machinery for the refining of sugar.

Mr. Dinkel was also a member of the Engineers' Club of New York, and a member of the Board of Trustees of Stevens Institute of Technology.

WILLIAM P. DONOVAN

William P. Donovan, general superintendent of the gasoline department of the Gypsy Oil Co., Tulsa, Okla., died in Philadelphia on May 30, 1919. Mr. Donovan was born on January 7, 1875, in Eagle Rock, Pa., and received his early education in the home schools. His father instructed him in telegraphy and later he attended a business school at Oil City, Pa. While studying there he found time to work as telegraph operator at Smoky Station of the National Transit Co.

In 1892 he entered the employ of the Crescent Pipe Line Co., where he assisted in the construction of an oil-pipe line from Oakdale to Marcus Hook, Pa. He was later promoted to the position of stationary engineer in charge of trunk-line stations on the same pipe line. When the Gulf Refining Co. was organized in 1903, Mr. Donovan became superintendent of distribution at Bayonne, N. J., having also supervision of all additions to the plant. In 1913 he was transferred to the gasoline department of the Gypsy Oil Co., an affiliated concern. Here he designed and constructed plants in Oklahoma, Texas, Louisiana and Mexico for the production of gasoline from casinghead gas. Mr. Donovan was actively interested in the Mid-Continent Section organization. He became a member of the Society in 1916.

HOWARD DUFF

Howard Duff was born in Jamaica, B. W. I., on October 30, 1879. He received his technical education at Drexel Institute, Philadelphia, where h received the degree of E. E. in 1906. His first position was in the construction department of the Philadelphia Electric Co. In 1907 he became connected with the Brooklyn Rapid Transit Co., Brooklyn, and was with this company for a number of years working in various capacities. As assistant of the testing bureau he was in charge of all steam testing and in addition superintended the economic operation of the various power stations.

For a short period Mr. Duff was associated with the Westinghouse Electric & Manufacturing Co., when he was called to Allentown, Pa., as superintendent of power for the Lehigh Valley Transit Co. After a year he resigned to become superintendent of the power division of the Lehigh Navigation Electric Co., with control of nine stations, which position Mr. Duff was holding at the time of his death, October 24, 1918.

Mr. Duff became an associate member of the Society in 1915.

HARRY DREW EGBERT

Harry Drew Egbert was born at Bay Head, N. J., on August 24, 1886, and died of pneumonia on March 23, 1919. He was educated in the Jersey City public schools and in the Newark Academy. In the latter institution he was prepared for college and entered Columbia University in 1904 as a sophomore. He had spent the previous year with his father, Prof. J. C. Egbert of Columbia, in Rome where the latter was serving as professor of classical literature in the American School of Classical Studies.

In 1907 Mr. Egbert was graduated from Columbia University, taking high rank in his class and being elected to Phi Beta Kappa. He was also elected the most faithful and deserving student of his class. He then entered the Schools of Applied Science and spent three years in study for the degree of mechanical engineer which he received in 1910. He received the Illig medal for extraordinary proficiency in his studies. In the summer during his scientific study, he attended the course in surveying at Camp Columbia and was an apprentice in different shops where he became familiar with the practical side of his profession.

Upon his graduation he became a member of the office force of the Guaranty Construction Co. After two years he was asked to join the staff of the Research Corporation and in connection with this organization, he became an expert on electrical precipitation. He wrote a number of articles on this subject.

Mr. Egbert was a member of the Phi Beta Kappa and Sigma Xi societies. He became a junior member of our Society in 1915 and was secretary of the New York Section, the Executive Committee of which on April 19, 1919, adopted the following resolution:

"*Resolved:* that there be entered upon the minutes the sincere regret of the Committee and their appreciation of his splendid achievements as an engineer and his untiring effort at all times to advance the interest of the Profession, the Society and this Committee."

CHARLES EKSTRAND

Charles Ekstrand was born on August 19, 1863, in Kalmar, Sweden. He received his early education in Swedish schools and served his apprenticeship as marine engineer in Oscarshamn. He was engaged in practical marine engineering until 1890 when he obtained his U. S. license as chief engineer on ocean-going steamers. In that year he became assistant chief engineer of the Brooklyn Cooperage Co. and Palmer Docks, Brooklyn, and nine years later was made chief engineer. During this period Mr. Ekstrand took a course in the scientific department of Cooper Union, New York City, receiving his B. S. in 1897 and his M. E. in 1898.

In 1906 Mr. Ekstrand left the Brooklyn concern to become superintendent for the Lowell M. Palmer's Works, York, Pa., where he remained until 1911 when he became connected as constructing and consulting engineer with the Warner Sugar Refining and Warner Quinlan Asphalt companies, New York. He was holding this position at the time of his death, September 28, 1919.

During the War Mr. Ekstrand served as lieutenant in the United States Naval Reserve Force. He was an associate member of the American Institute of Electrical Engineers. He became a member of our Society in 1898.

WILLIAM T. ENGLISH

William T. English, president of the W. T. English Co., Boston, died on December 22, 1919. Mr. English was born on July 21, 1855, in Baltimore, Md. The family moved to Boston when he was a child and he was educated in the schools of that city. He served his apprenticeship as patternmaker with the Boston Machine Co., and then entered the employ of the Boston Blower Works. In about four years he was advanced to the position of works superintendent. His leisure time at this period was devoted to the study of mechanical engineering.

In 1895 Mr. English became connected with the Walworth Construction & Supply Co., as construction engineer, resigning in 1903 to become president of the Walworth-English Flett Co., with which concern he was associated for fourteen years. For the last two years Mr. English was engaged in business for himself under the name of the William T. English Co., Boston.

Mr. English belonged to several fraternal organizations. He became a member of the Society in 1902.

WALTER ERLENKOTTER

Walter Erlenkotter was born on August 22, 1887, in Hoboken, N. J. He received his early education in the Hoboken Academy and entered Stevens Institute of Technology in 1904 from which he received his M. E. degree in 1908.

In the fall of that year Mr. Erlenkotter entered the apprenticeship course of the Allis-Chalmers Co. in their foundry and electrical-erecting shops, West Allis, Wis. The following year he became connected with the Owens Engineering Co., Brooklyn, N. Y., as an assistant in design and construction. In 1910 he accepted a position with J. G. White & Co., New York City, where his work dealt with the design and construction of steam generating plants, electrical stations, water-gas plants, inspection of material and shipment of electric cars. From 1911 to 1916 Mr. Erlenkotter was fuel-engineering chemist in charge of the Central Testing Laboratory, Bureau of Contract Supervision, New York City. Here he had charge of physical tests, sampling, coal analysis, the design of new apparatus and its installation, etc. In the latter part of 1916 he became chief engineer for the Secaw Chemical Co., Irvington, N. J., and in January 1917 he became vice-president of the company, which position he held at the time of his death, May 14, 1919.

Mr. Erlenkotter was a member of a number of fraternal organizations. He became a junior member of the Society in 1912 and in 1916 was promoted to the grade of associate-member.

JAMES R. FLETCHER

James R. Fletcher, factory manager of the P. & F. Corbin Co., manufacturers of hardware, New Britain, Conn., died at his home in that city on September 13, 1919. Mr. Fletcher was born on February 21, 1867, in Jersey City, N. J., where he was educated and learned the trade of machinist. He served his apprenticeship with the New Haven Clock Company in tool making and die sinking. He was formerly associated with the American Cash Register Company, the National Cash Register Company, Sargent & Co., the New Britain Hardware Company, of which he was in charge for seven years, and the Yale & Towne Manufacturing Co., of which he was assistant superintendent.

In 1906 Mr. Fletcher became associated with the P. & F. Corbin Co. as manufacturing expert to assist the general superintendent. Later he was made factory manager, a position created to give him the wider range warranted by his exceptional ability.

He was a member of a number of fraternal organizations. He became a member of the Society in 1906.

JAMES J. FLYNN

James J. Flynn was born in April, 1893, in New York City. He was educated in the public schools of the city and was graduated from the Mechanics' Institute. In 1908 he started work for the Western Electric Co., taking their regular students' course and spending six months in their manufacturing department.

He was then transferred to the tool and gage-inspection department where he was located for about five years; he was then placed on construction work of telephone equipment on the road. After seven years with this concern, Mr. Flynn became associated in 1915 with the Remington Arms Bridgeport Works as assistant low-tension engineer. The following year he was made foreman inspector of the barrel department and in 1918 he became chief inspector of ordnance for the U. S. Government in the same company.

Mr. Flynn became a junior member of the Society in 1918. He died on February 11, 1919.

ALDEN WALKER GALLUP

Alden W. Gallup, works engineer of the Hinde & Dauch Paper Co., died on December 17, 1918, in Sandusky, Ohio. Mr. Gallup was born on October 29, 1890, in Sandusky, Ohio. He was prepared for college in St. John's School, Manlius, N. Y., and then entered Cornell University from which he was graduated in 1913 with the degree of M. E. His first and only business connection was with the Hinde & Dauch Paper Co., Sandusky, Ohio, where he acted in the capacity of works engineer. Mr. Gallup became a junior member of the Society in 1917.

HENRY LAURENCE GANTT

On November 23, 1919, H. L. Gantt died at his home in Montclair, N. J., after a brief illness. In his death the engineering profession has lost one of its foremost engineers, a pioneer in industrial management and a keen student of the human element in its relations to industry. He was associated with Frederick W. Taylor in his early work at the Midvale and Bethlehem Steel Companies and with this as a basis, and his personal ability as an organizer, he later established and successfully conducted his own consulting practice as an industrial engineer. His career was marked by original and thoughtful work, progressive in viewpoint, and effective in its accomplishment. In later years he developed a broad conception of industry as a national problem in which he regarded it essential that the man at the top should have the same close scrutiny and careful direction that has in the past been given his co-workers in the lower ranks. To these views he gave expression in a characteristically original manner in many addresses and written articles.

Henry Laurence Gantt was born May 20, 1861, in Calvert County, Maryland. He was graduated from Johns Hopkins as a Bachelor of Arts when only 19 years of age and taught for three years at the McDonough School in Baltimore County, which he had attended as a boy. In 1884 he received the degree of Mechanical Engineer from Stevens Institute of Technology.

Following his connection with Frederick W. Taylor, he conducted his consulting practice until his death and had as clients more than a score of prominent manufacturing plants. Among these were the Westinghouse Electric and Manufacturing Co., American Locomotive Co., factories of Remington Typewriter Co., Bethlehem Foundry & Machine Co., Saco-Lowell Shops, Ingersoll-Rand Co., Amoskeag Mfg. Co., Cheney Brothers (silk manufacturers), Brighton Mills, Sayles Bleacheries and Corticelli Silk Mills.

Mr. Gantt was active in The American Society of Mechanical Engineers and served on the Council as Manager, 1908–11, and as Vice-President, 1913–15; on the Meetings Committee, 1912-17; and on the Executive Committee of this Council, 1913–14.

His papers contributed to the Society and which enrich its TRANSACTIONS comprise the following list, of which his early paper on the bonus system brought prominently to the front new features of management developed during his early work with Mr. Taylor: Steel Castings; Recent Progress in the Manufacture of Steel Castings; Bonus System of Rewarding Labor; Graphical Daily Balance in Manufacture; Training of Workmen in Habits of Industry and Coöperation; Mechanical Engineer and the Textile Industry; Modifying Systems of Management; Measuring Efficiency; The Relation Between Production and Costs; Productive Capacity a Measure of Value of an Industrial Property; Expenses and Costs; Efficiency and Democracy.

During the German trip of the Society in 1913 Mr. Gantt was a frequent speaker at the various functions and at the Leipsic meeting of the Verein deutscher Ingenieure, held jointly with our Society, aroused much interest by his discussion of Scientific Management.

Mr. Gantt was also the author of three books: Work, Wages and Profits: Their Influence on Cost of Living; Industrial Leadership (Yale Lectures); and Organizing for Work. In addition, he contributed many articles to the technical and daily press.

There grew up around Mr. Gantt several groups having as their central thought the principles which he enunciated. Among these was a circle of friends who regarded service to the community as the essential element in industry and which it was believed could be attained by concentrating on the production of goods rather than on the production of profits. It was considered that industry required a "new machine" rather than a disrupting social movement, and the movement was thus given this unique designation.

During the war Mr. Gantt acted in a consulting capacity, first for the Ordnance Department by invitation of General Crozier, who recommended for the entire War Department the use of Gantt's production charts, employed by Mr. Gantt in his work and which had been used in the Ordnance Department. These charts are well known among engineers and constitute an important development in recording the progress of work. When General Crozier was retired as Chief of Ordnance a change in plans was made, but the charts were later used by the U. S. Shipping Board and Emergency Fleet Corporation in routing ships and in following up constructive work.

Mr. Gantt's vision of a better spirit in industry is well expressed by the following quotations from his latest book, Organizing for Work: "The community needs service first, regardless of who gets the profits, because its life depends upon the service it gets. The business man says profits are more important to him than the service he renders; that the wheels of business shall not turn, whether the community gets the service or not, unless he can have his full measure of profit. He has forgotten that his business system had its foundation in service, and as far as the community is concerned has no reason for existence except the service it can render."

GEORGE K. GARVIN

George K. Garvin, president of the Garvin Machine Co., New York, died at his home in Garden City, Long Island, on February 20, 1919. Mr. Garvin was born on May 2, 1859, in Hartford, Conn. He was educated in the public schools of New York and Jersey City, later attending Hasbrouck Institute. At the age of sixteen he entered his father's business, then known as Smith & Garvin, and upon his father's death became president of the concern. He became a member of the Society in 1909.

JOHN ARTHUR HALL

John A. Hall was born in Pittsburgh, Pa., on September 16, 1877. He was graduated from Sheffield Scientific School, Yale University, class of 1897. He spent the next year in graduate work in Yale when he became connected with the Carnegie Steel Co., Pittsburgh, as chemist.

In 1902 he accepted the position of superintendent of the Alice Furnaces of the Tennessee Coal, Iron & Railroad Co., Birmingham, Ala., and was later associated as chemical engineer with the Edison Portland Cement Co., Stewardsville, N. J. From 1910 to 1915 Mr. Hall was with the Ransome Concrete Machinery Co., Dunnellen, N. J., as superintendent in charge of production, designing, purchasing, etc. He was at one time located in New York City as a manufacturers' agent and more lately has been engaged in business in Elizabeth, N. J., under the firm name of the Hall Machine Co., general machinists.

Mr. Hall became a member of the Society in 1916. He died on October 1, 1919, of injuries received in an automobile accident.

RICHARD HAMMOND

Richard Hammond was born on January 27, 1849, in Thurles, Tipperary County, Ireland, and received his early education there in the parochial schools, coming to this country when about thirteen years of age.

He served his apprenticeship in the boiler-making trade with Starbuck Brothers, Troy, N. Y., and became a journeyman boiler maker in charge of the outside work for the company with which he was connected for several years. In 1872 he transferred to the oil fields in Pennsylvania and started business at Titusville, Pa. In 1875 he entered into partnership with Mr. John Cofield in Franklin, Pa., as manufacturers of oil-well boilers and engines and a short while later purchased Mr. Cofield's interests, extending the business into tank building and refinery construction.

In 1880 he removed to Buffalo where he designed a plant for the manufacture of steam boilers. His methods and ideas were considerably in advance of the times and it was not long before his plant, known as the Lake Erie Boiler Works, had obtained an unusual reputation for the manufacture of heavy marine boilers. In 1890 he organized the Lake Erie Engineering Works of which he became president. The shops were equipped with the most modern tools especially designed and constructed according to Mr. Hammond's own specifications by the Niles Tool Works. He continued at the head of both concerns up to within a couple of years ago, when he disposed of his interests and retired from active business life. He died on October 9, 1919.

Mr. Hammond was an early member of the Engineers' Club of New York. He became a member of the Society in 1890 and devoted a great deal of his time and practical boiler-manufacturing experience to the Boiler Code Committee of the Society of which he was a member.

FRANCIS L. HAND

Francis L. Hand was born at Dennisville, Cape May County, N. J., on February 18, 1838. His family moved to Philadelphia in 1844 where he attended the public schools. Coming of a family which had followed the sea for generations he began "steamboating" in 1854 on the *Delaware* on which he remained for four years when he received his first engineer's license. He then became assistant engineer on the *Kennebec*, running on the outside line between Philadelphia and New York.

When the war came in 1861, he shipped on the steamer *Union* as engineer and took part in the Dupont-Sherman Expedition on October 29, 1861. The vessel went ashore on Bogue Island during a storm, was totally wrecked and all on board were taken prisoners of war by the Confederates, and sent to Salisbury, N. C. After ten months, Mr. Hand was paroled and returned to his home.

In 1865 he became chief engineer of the new steamer *Star of the Union* and superintended fitting her out during her construction. Mr. Hand afterwards became chief engineer in the Southern Mail Steamship Co. of Philadelphia and was with them until 1873 when he was appointed U. S. inspector of steam vessels.

In 1886 he accepted the position of general superintendent of the Philadelphia Water Department. He filled that position until 1900 and was then appointed chief engineer of the Water Bureau, remaining until 1906 when he retired from active life. Mr. Hand died on September 11, 1919, in his eighty-second year.

Mr. Hand was a member of the Engineers' Club of Philadelphia and of the American Water Works Association. He became a member of the Society in 1888.

HENRY S. HAYWARD *

Henry S. Hayward, Lieutenant, U. S. Naval Reserve Force, was born on December 25, 1876, in Elizabeth, N. J. He was a 1900 graduate of Stevens Institute of Technology.

His first position was in the drafting room of the Pennsylvania Railroad, Jersey City, N. J. He was next associated with the Franklin Air Compressor Co., Franklin, Pa., as chief engineer, his work consisting of designing and supervising the construction of their power plant. From 1902 to 1905 he was in business for himself as a mechanical engineer, handling mechanical mill supplies.

In 1906 he became assistant to the vice-president of the Franklin Railway Supply Co., manufacturing and promoting special mechanical devices for locomotives. Prior to his enlistment in 1918 he was president of the Roybel Packing Co., N. Y.

He was commissioned a lieutenant (j.g.) in the Naval Reserve Force and was stationed at the Charleston Navy Yard, Charleston, S. C. He was north on a twenty-day furlough when he contracted pneumonia and died on March 31, 1919.

Lieutenant Hayward became a member of the Society in 1900.

JAMES NISBET HAZLEHURST *

Major James N. Hazlehurst, one of the South's foremost engineers and a prominent member of the engineering profession of Atlanta, Ga., died in France February 9, 1919. He was born at Sparta, Ga., January 15, 1864, and received his early education and engineering training at the University of the South, Sewanee, Tenn.

In 1885 Major Hazlehurst was president of the Chattanooga Rolling Mill, later becoming chief engineer and president of a railroad in Florida. In 1895 he was engineer for the city of St. Augustine, Fla., during which time he built the water works and paved the majority of the streets of that city. In the year 1889 he was elected chief engineer of the Board of Public Works of Tampa, Fla., and it was at this time he wrote his book Towers and Tanks. In 1900 he became chief engineer of the Miles Salt Co., New Orleans, La., later going to Mobile, Ala., in 1901 as chief engineer for the Board of Public Works in that city for a period of five years.

In 1906 Major Hazlehurst moved to Atlanta, Ga., opening a consulting engineer's office, becoming prominently identified with the engineering activities of that city, taking a leading part in the organization of the Affiliated Technical Societies of Atlanta, a movement which stands as the first of its kind in the United States. He served as chairman of the committee on Engineering and Good Roads of the Atlanta Chamber of Commerce, was a member of the Commission which selected the site of Camp Gordon as a military cantonment and was a member of the National Council of Defense.

Major Hazlehurst was a member of the American Society of Civil Engineers, the American Water Works Association, and the American Public Health Association. He became a member of our Society in 1916.

When the United States entered the War he immediately offered his services to the Government and was commissioned a Major of Engineers, June 20, 1917, and assigned to the staff of Maj.-Gen. Leonard Wood, Commander of the Department of the Southeast, as officer in charge of the water-supply section for the camps located in the Southeastern states. Early in September, 1917, he was ordered overseas as the principal assistant to the officer in charge of the water-supply section of the American Expeditionary Forces at Chaumont. He was later attached to the Third British Army as an observer in the Valley of the Somme, being transferred to Tours in the same capacity as at Chaumont. In October, 1918, he was ordered to the southern part of France as Engineer Officer in charge of construction and water-supply officer for Base No. 6, which included the length from Riviera to the Italian frontier. Three months later he was ordered to Paris as a member of the American Commission to negotiate peace. His assignment was in the economical and financial section of the Belgian Commission, whose duty it was to study and check the consequential damage due to the German occupancy.

Major Hazlehurst's studies were mines, metallurgy, railways and industrial and general buildings. He was made Director of the Division having to do with damages to structures in Belgium. His death made it impossible for him to complete this work.

CARL ANTHONY HEILMANN *

Carl Anthony Heilmann, Captain, 5th Engineers, U. S. A., died July 12, 1919, as a result of an accident near Camp Humphreys, Alexandria, Va., when an army motor truck plunged over the side of a bridge.

Captain Heilmann was born in Brooklyn, N. Y., January, 1886. He was a graduate of Manual Training High School, and also studied at Worcester Polytechnic Institute for two years, graduating in 1908 as a mechanical engineer from Purdue University, where he completed the studies begun in Worcester. He served his apprenticeship in various machine shops during summer vacations. He was associated with the Ideal Portland Cement Company, Portland, Colo., as draftsman; with the Monett Electric Light Company, Monett, Mo., where he had charge of the electrical equipment and construction; with the Green Fuel Economizer Company in the capacity of sales engineer; with the American Radiator Company as engineering representative, Philadelphia, Pa.; and with Warren Webster and Company, as district manager, Washington, D. C.

Captain Heilmann had been stationed at Camp Humphreys after eight months of arduous service in France, with the First Division, having returned to the United States in order to make a lecture tour of the country. He entered the service shortly after the declaration of war and was commissioned as first lieutenant, Engineer Corps, at Fort Myer, Va., in June, 1917. He was promoted to Captain soon after his return to the United States. Captain Heilmann was elected to membership in the Society in 1913.

HENRY WILLIAM HENES

Henry W. Henes, of the Acme Steel Goods Co., Chicago, died on January 19, 1919. Mr. Henes was born in January, 1887, in New York City, and was educated in the city schools. Later he entered Columbia University from which he was graduated in 1909 with the degree of M. E.

His first position was as structural-steel estimator and inspector with A. Bolter's Sons, Chicago, where he remained for about three years, when he became associated with the Dueth-Henes Corporation, manufacturing agents in the same city, as secretary and treasurer. His next connection was with the Stromberg Motor Devices Co., also in Chicago, first as assistant purchasing agent and later as sales engineer. In July 1918 he became associated with the Acme Steel Goods Co. in charge of the manufacture of nailless box-strapping tools. Mr. Henes was a member of several social clubs in Chicago. He became a junior member of the Society in 1909.

GEORGE SHERWOOD HODGINS

George S. Hodgins, editor of *Railway and Locomotive Engineering*, died at his home in New York City on January 18, 1919. He was born in 1859 in Toronto, Ont., and was a graduate of the Upper Canada College and the school of practical science, University of Toronto.

He served his apprenticeship with the Canadian Locomotive & Engine Co., Kingston, Ont., in machine-shop work, locomotive erecting, etc. In 1882 he was appointed draftsman in the same company and was directly responsible for locomotive design. After some experience in a division master mechanic's office on the Canadian Pacific Railway, he was advanced to various positions on the road and of late was locomotive inspector on the entire system. In 1889 he was recalled to the Canadian Locomotive Works as mechanical engineer, where he had charge of all engineering and designing work. Later he entered the service of the Pressed Steel Car Co., Pittsburgh, Pa., as general inspector of the output of that extensive plant, and was also for some years inspector of the Richmond Locomotive Works.

During these earlier years he had contributed to a number of railroad publications. In 1900 he entered the field of practical journalism as editor of *The Railroad Digest.* In 1902 he joined the staff of *Railway and Locomotive Engineering* as associate editor and in 1908 became managing editor, which position he held until 1911, when he was called by the Canadian Government to make a comprehensive report on the shops, appliances, tools and equipment necessary for the Trans-Continental Railroad. On the completion of that work in 1915 Mr. Hodgins joined the staff of the Railway Periodicals Company as managing editor of the *Railway Master Mechanic* and *The Railway Engineering and Maintenance of Way.* In 1916 he returned to *Railway and Locomotive Engineering* and remained on the staff as editor until his death.

He contributed many articles to popular science magazines and as a writer on engineering and technical subjects his style was marked by an exact and comprehensive lucidity.

Mr. Hodgins became a member of the Society in 1908.

ELIAS QUEREAU HORTON *

Elias Q. Horton, Lieutenant (j.g.), U. S. Naval Reserve Force, died of influenza-pneumonia on March 4, 1919. Lieutenant Horton was born in Ossining, N. Y., on July 28, 1884. He was educated in the public schools of New York City, Stevens Preparatory School and Stevens Institute of Technology, from which he was graduated in 1905 with the degree of M.E.

Upon graduation he became associated with the Otis Elevator Co., New York, and gained his shop experience in their factory at Yonkers. He was then transferred to the estimating department and in 1908 he was made assistant chief estimator. In 1910 he was made assistant manager in Oklahoma City, in charge of all sales in the state. His advancement was continuous and rapid until, at the time of his enlistment in April 1917, he was in charge of the Washington office of the company.

He enlisted in the Naval Reserve Force and was assigned to active duty as coxswain in Newport, R. I. In June of that year he was sent to Annapolis for special training, and at the termination of the course was commissioned as ensign. He was ordered to duty on the U.S.S. *South Carolina*, where he served as junior watch officer, division officer and communicating officer, respectively. He was then promoted to the rank of lieutenant (j.g.). Shortly before his death he was ordered to the Receiving Ship *Philadelphia*, where he was to have been discharged and to have rejoined the staff of the Otis Elevator Co.

Lieutenant Horton was a member of the Sons of the Colonial Wars, the Sons of the Revolution and of a number of clubs. He became a junior member of the Society in 1912.

RICHARD HUMPHREY HUGHES

Richard H. Hughes, vice-president and general manager of the Crescent Portland Cement Co., Wampum, Pa., and a member of the Board of Directors of the Portland Cement Association, died on April 12, 1919.

Mr. Hughes was born on September 16, 1862, in Lima, Ohio. He received his education in the schools of that city and also served there his apprenticeship in railroad engineering. In 1901 he became connected with the Portland Cement Co. and took charge of the remodeling of their plant. During the year 1908 and 1909, in coöperation with Mr. W. B. Ruggles of the Ruggles-Coles Engineering Co., he had entire charge of the designing and building of the company's 3000-barrel plant and upon its completion, of its operation, alteration and additions.

Mr. Hughes belonged to a number of social and fraternal organizations. He became an associate-member of the Society in 1914.

HENRY LOCKETT HUTSON

Henry Lockett Hutson was born at Americus, Ga., on December 30, 1876, He was the son of Charles Woodward Hutson and his wife, Mary Jane Lockett. His father was a college professor, occupying chairs in various southern colleges, including the University of Mississippi at Oxford, and the A. & M. College of Texas at Bryan.

Mr. Hutson received his preliminary education in the public and private schools in the towns where his family resided, and his college training at the A. & M. College of Texas, where he was graduated in the mechanical engineering course in 1896, one of the first three in his class. He exhibited unusual talent for mechanical work in his early childhood, and was a great student of engineering subjects up to the time of his death.

In 1898 he volunteered as a private in the First Regiment of the U. S. Volunteer Infantry, commanded by Colonel Riche. In 1898, after being mustered out, he entered the employ of Henry R. Worthington at the Brooklyn shops as a student apprentice and received the usual thorough training in practical hydraulic engineering given by the Worthington Company. In 1901 he entered the employ of A. M. Lockett & Co., Ltd., of New Orleans, as mechanical engineer. By reason of his ability and great loyalty to the interests of the company he was later promoted to the position of chief engineer.

In addition to engineering skill he possessed unusual talent in business management, and in addition to his work as chief engineer he was the sales manager and secretary of the Lockett Company. He supervised the designing and construction of perhaps a greater number of low-lift centrifugal pumping plants of large capacity than any other one engineer in this country, and by reason of this broad experience he was regarded as an authority on this class of work by other engineers in the Southwest.

Mr. Hutson became a member of the Society in 1906 and at the time of his death was Chairman of the New Orleans Section. He was also an active member of the Louisiana Engineering Society. He died on January 10, 1919.

CHARLES EDWARD JOHNSON

Charles E. Johnson was born in December 1883, in East Atchison, Mo. He received his early education in the public schools of De Witt, Neb., and the high school of Lawrence, Kan. Later he attended the University of Kansas from which he was graduated in 1910 with the degree of B. S. He was also a graduate of the International Correspondence School.

Upon graduation he worked for two years as draftsman for the Santa Fe Railway Co. at Topeka, Kan. He was next connected with the Lawrence Paper Mill, Lawrence, Kan., as chief engineer of their power plant. The years 1912 to 1915 he spent in the employ of the Board of Education, Kansas City, Kan., as an instructor of mechanical drawing in both the day and evening central high school. From there he went to the high school at Ottawa, Kan., as a teacher of mechanical drawing and manual training. In the fall of 1916 he accepted a position with the Diamond Gasoline Co., spending his time partly in the Kansas City, Mo., offices of the company and partly in the oil fields of Oklahoma. He received several promotions during his service there and at the time of his death, January 5, 1919, he held the position of chief engineer.

Mr. Johnson became an associate-member of the Society in 1918.

JOSEPH ESREY JOHNSON, JR.

Joseph E. Johnson, Jr., was born on July 10, 1870, in Longdale, Va. He was prepared under a tutor for Haverford College from which he was graduated in 1888. For two years he worked in the drafting room of the Baldwin Locomotive Works and then returned home for a year's study, receiving his M. E. degree from Haverford in 1891. That autumn Mr. Johnson entered a post-graduate course at Cornell University and in 1892 received his master's degree in mechanical engineering.

After some valuable experience as assistant to the president of the Cranberry Iron & Coal Co., in North Carolina, he entered in 1893 the employ of John E. Sweet of the Straight Line Engine Co. and went from there as assistant to engineer of tests with the Solvay Process Co. In 1894 he became assistant to the superintendent of the Ames Iron Works, engine builders in Oswego, N. Y. In 1895 he was offered and accepted the position of engineer and assistant manager of the Longdale Iron Co., where he remained (with the exception of a few months as engineer officer in the Navy in 1898) until 1899. From 1899 to 1901 he was with the Carnegie Steel Co. at the Edgar Thompson & Duquesne Works, part of the time as master mechanic at the Duquesne blast furnaces.

In 1901 he returned to the Longdale Iron Co. and remained there until 1906 when he became general manager of the Princess Furnace Co., Glen Wilton, Va., whose plant he practically rebuilt. In 1909 he was appointed general superintendent of the southern furnaces and coke ovens of the Republican Iron & Steel Co., Birmingham, Ala. In 1910 he became manager of the blast furnace and chemical plant of the Lake Superior Iron & Chemical Co., Ashland, Wis., which he largely rebuilt.

In 1913 Mr. Johnson opened offices in New York City as consulting and metallurgical engineer. In the interest of New York capital he made a trip to China in 1916.

He was the author of Blast Furnace Construction in America which was published early in 1917; the following year saw the publication of its companion volume Blast Furnace Principles, Operation and Products.

Mr. Johnson was a director of the American Institute of Mining and Metallurgical Engineers, the American Electrochemical Society, the American Iron and Steel Institute, the British Iron and Steel Institute and the American Society of Testing Materials. He became a member of our Society in 1896. He died on April 4, 1919.

WILLIAM D. KELLEY

William D. Kelley, general superintendent of meters of the Consolidated Gas Company of New York, died on April 12, 1919, at his home in Mount Vernon, N. Y. Mr. Kelley was identified with the gas industry in New York City for forty-one years. He began this business career as a "fitter" but developed rare business qualifications and received frequent advancements so that at the time of his death Mr. Kelley was considered one of the most efficient men in the gas industry.

Mr. Kelley was educated in the public schools of New York City. At the time of his death he was a member of the American Gas Association, the Society of Gas Engineering of New York City and the National Association of Corporation Schools. He became a member of our Society in 1915. Mr. Kelley took a very active part in the proceedings of these organizations and was greatly interested in their work.

His funeral was held on April 15, at Mount Vernon, N. Y., and was attended by many officials and employes of the Consolidated Gas Company of New York.

HENRY DONALD KEMP

Henry Donald Kemp died at Montreal, Canada, on October 4, 1919. He was born in Boston, Mass., in October 1890, and was graduated from the Massachusetts Institute of Technology in 1912 with the degree of B.S. in electrical engineering. For several years he was in Rio de Janeiro, Brazil, first as assistant to the engineer in charge of the construction of a coal-handling plant and later in charge of the erection of a half-million dollar coal pier. His connection at this time was with the Mead-Morrison Manufacturing Co. and following his return from Brazil he was assistant engineer of the foreign department of this concern.

During the war he was engaged in the production of munitions. He was superintendent of the British Munitions Company, Ltd., near Montreal, Canada, and in charge of all work of the concern, the only American among 4,000 Canadians. Following an unsuccessful attempt to enter active service, he accepted the position of assistant works manager of the National Conduit & Cable Co., an ordnance concern, at Hastings-on-the-Hudson, N. Y. Later he returned to Montreal and reëngaged in production work in munitions supplies. At the time of his death he was about to enter the firm of Howard Smith Paper Co., Ltd., as assistant to the president.

Mr. Kemp was a member of the Technology Club of New York, of the American Institute of Electrical Engineers, and a junior member of the Society since 1916.

MATTHEW LEANDER KING *

Matthew Leander King, Major, U. S. A., died on October 23, 1919. Major King was born in Panora, Ill., on May 20, 1878. He was graduated from the mechanical engineering department of Iowa State College in 1906.

He spent five years as an experimentalist in agricultural engineering with the Experiment Station of Iowa State College, Ames, Iowa, during which time he invented the hollow clay tile silo. For two years he was superintendent and general manager of the David M. Bradley Implement Works at Bradley, Illinois. He organized the Iowa City manufacturers into the Permanent Buildings Society for the development of new designs of and uses for hollow-clay building tile.

Mr. King entered the army in September, 1917, with the rank of Captain and was assigned to the Aviation School of Aerial Observation at Post Field, Fort Sill, Okla., in charge of maintenance and repair of aeroplanes. He was advanced to the rank of Major in August, 1918, and in November of that year was assigned to Indianapolis as chief engineering officer for aviation in the Northern District. In February 1919, he was made acting director of aviation for the Northern District. In April he was transferred to Washington, D. C., and from there he was assigned on special missions until July when he became fight commander and chief engineering officer of the All-American Pathfinding and Recruiting Expedition. He was transferred from the Officers' Reserve Corps to the regular army with the rank of Major in October about a week before his death. While at Post Field he learned to fly and was given the classification of Reserve Military Aviator.

Major King was a charter member of the American Society of Agricultural Engineers, a member of the American Society for Testing Materials, and belonged to various aeronautical and officers' clubs. He became a member of our Society in 1912.

HENRY J. KLAER

Henry J. Klaer was born on October 28, 1888, in Milford, Pike County, Pa. He was prepared for college at Blairstown Academy and was graduated from the University of Pennsylvania in 1909 with the degree of B.S. in chemical engineering. Immediately upon graduation he entered the employ of the American Steel Foundries at its Chester plant and by progressive steps became chief engineer, resigning in 1914 to become chief engineer of the Liquid Carbonic Co., Chicago, which position he held until 1916 when he became treasurer of Robert Wetherill & Co., Inc., Chester, Pa. Later in the same year he became general manager and in 1917 was elected vice-president of the Penn-Seaboard Steel Corporation, holding this position at the time of his death, October 15, 1918.

Mr. Klaer was captain of the Chester company of the Pennsylvania Reserve Militia and contracted influenza while preparing the armory of that organization for use as an emergency hospital during the epidemic in the fall of 1918.

He was a member of the Sigma Psi Society and of several clubs. He became an associate-member of our Society in 1916.

WALTER W. KREISER

Walter W. Kreiser was born on July 23, 1892, in Jersey City, N. J. He was prepared for college at Stevens Preparatory School and then entered Stevens Institute of Technology, from which he was graduated in 1916 with the degree of M. E. He spent a year in chemical research work and then became connected with the mechanical engineering department of the Western Union Telegraph Company, New York. In September 1917 he became assistant engineer of tests, in the Brooklyn Navy Yard. Here, while performing his regular duties in the mechanical laboratory, he was injured by falling into a manhole carelessly left open, and, after an eleven-months' illness, died on October 11, 1919.

Mr. Kreiser was a member of the American Chemical Society. He became a junior member of our Society in 1916.

CLARENCE BOOTH LAMONT

Clarence B. Lamont was born in February, 1887, in Van Etten, N. Y. He was graduated in 1900 from Cornell University with the degree of M. E. He was first employed by the Union Iron Works, San Francisco, as engine draftsman, leaving there to become ship draftsman in the United States Navy. From 1902 to 1908 he was associated for various periods with the following firms: Moran Brothers Co., Seattle, as engine draftsman; the White Pass & Yukon Route, as inspecting engineer, second in charge of all construction and repairs and the operation of about eighteen steamers on the upper Yukon; the Pacific Coast Steamship Co., as superintending engineer in full charge of the ocean-going fleet of about twelve to fifteen steamers; the Pacific Engineering Co., as president. In 1908 he became connected with the Seattle Construction & Dry Dock Co., as assistant to the president, and was with this firm until 1914 when he opened a consulting office in Seattle. When the United States entered the war Mr. Lamont offered his services to the Government and was commissioned a Captain in the Engineers' Corps.

Mr. Lamont became a member of the Society in 1914. He died on March 21, 1918.

FREDERICK LESTER LANE

Frederick L. Lane was born in St. Johnsbury, Vt., on March 6, 1856. He received his early education in the public schools of Springfield, Mass., and as a young man was employed in the same city by the Smith & Wessen Co. and at the Springfield Arsenal. He was also connected for about five years with the E. S. Stacy Co., machinists and engineers at Springfield, on machine-tool design and construction.

He later entered the employ of the Chapman Valve Manufacturing Co., where he spent nearly thirty years, being appointed superintendent in 1905. In 1907 Mr. Lane accepted the position of mechanical superintendent with the Haines, Jones and Cadbury Co., Philadelphia, manufacturers of plumbing supplies. Here he had entire charge of the factory output. He was with this company for about twelve years when in February of this year he became works manager of the McCambridge Co., Philadelphia, which position he held at the time of his death, March 6, 1919.

Mr. Lane was a member of the American Academy of Political and Social Science. He became a member of the Society in 1915.

FRED A. LARKIN

Fred A. Larkin was born in Mason, N. H., on December 2, 1850. He was educated in Massachusetts schools and gained his technical knowledge through special instruction. He served an apprenticeship in shop practice, and for three years served as engineer in charge of engines and boilers in manufactories.

He then became connected with the Allis-Chalmers Manufacturing Company, Milwaukee, representing them in New York. For a number of years he was connected with the General Electric Company, and for two years was with the Foundation Company, New York.

Mr. Larkin was a member of the Engineers' and also of the Republican Club. He became a member of the Society in 1888.

CLIFFORD LEE

Clifford Lee was born in Forrest Grove, Oregon, on October 23, 1886. He was educated in the common schools of Washington and of Oregon and in 1911 was graduated from Purdue University as a mechanical engineer. Upon graduation he became connected with the Willamette Iron and Steel Co. Works, Portland, Ore., where he served a "technical apprenticeship" course in the boiler and structural shop, later being advanced to the engineering force where his work dealt with design and estimating. His next position was with the United States Rubber Corporation, first as development engineer on the installation and erection of vulcanizing apparatus and then as mechanical engineer in charge of construction and maintenance at the India Rubber Co., New Brunswick, N. J., the hard-rubber plant for the corporation. In August, 1918, Mr. Lee resigned from this position to become mechanical engineer at the Government explosive plant in Nitro, West Va. At the close of the War he returned again to take up his position at the India Rubber Co. but was compelled very shortly to give up his work on account of ill health. He died in Portland, Ore., on May 24, 1919.

Mr. Lee became a member of the Society in 1916.

AUGUST S. LINDEMANN

August S. Lindemann, secretary and treasurer of the Milwaukee-Waukesha Brewing Co., who died in the early part of 1919 was born in Milwaukee, Wis., in June 1866. He was graduated from the University of Wisconsin in 1885 with the degree of B.M.E. and in 1887 received his master's degree in mechanical engineering.

He served an apprenticeship as machinist with Filer & Stowell and with E. P. Allis & Co., both firms in Milwaukee. He was employed in the drafting room of Flanders & Bottum for one year when he became connected with the Pfister & Vogel Leather Co. as erecting engineer. From 1889 to 1898 he was associated with J. P. Lindemann, first as erecting engineer and later as designer and constructor of new stamping and metal-forming machinery. His next position was with the Milwaukee-Waukesha Brewing Co., as manager and general superintendent, designing and developing machinery used in bottling works. His advancement was rapid and at the time of his death he held the position of secretary-treasurer of the company.

Mr. Lindemann became a member of the Society in 1913.

FRANK J. LOGAN

Frank J. Logan, who, until his retirement in 1914, was vice-president and treasurer of the Logan Iron Works, Brooklyn, died on April 10, 1919. Mr. Logan was born on September 25, 1855, in Brooklyn, N. Y. He was educated in the public schools of the city and later attended the Flushing Institute and Cooper Union. He started work with his father in the Logan Iron Works. At the time of his father's death, in 1879, Mr. Logan entered the firm and continued with them until his retirement, May 1, 1914.

Mr. Logan belonged to a number of social clubs and business organizations. He became a member of the Society in 1894.

CHARLES EDWARD LORD

Charles Edward Lord, general patent attorney and manager of the patent department of the International Harvester Company, Chicago, died suddenly on September 25, as the result of injuries received in an accident at the Deering Works the evening before.

Mr. Lord was born in Somerville, Mass., on October 31, 1875, attended the public and high schools at Somerville, and was graduated from the Massachusetts Institute of Technology with the degree of B.S. in 1898. For a year he was employed in the inspection department of the American Telephone and Telegraph Company, with headquarters at Philadelphia. He then returned to the Massachusetts Institute of Technology as instructor for a short time, and later became assistant examiner in the United States Patent Office, Washington, D. C.

The following resolution was adopted by the executive committee of the Chicago Section:

WHEREAS: Our fellow-member and former chairman of the Chicago Section, A.S.M.E., Mr. Charles E. Lord, who has served devotedly and who has sacrificed much in the interests of the mechanical engineers of the country, the Society and Section, has unfortunately been taken from us in the midst of a useful and active career; be it

Resolved, By the Executive Committee of the Chicago Section that we record our deep regret at the loss of our friend and fellow-engineer. That we wish to express our appreciation of his high qualities, his interest in his profession, and the untiring zeal with which he has labored for its advancement and of its members, his great ability and high attainments. Be it further

Resolved, That we express to his family and his associates in business our appreciation of his worth and our sincere sympathy with them in their loss.

Executive Committee, Chicago Section,

ARTHUR L. RICE, *Chairman*,
G. R. BRANDON, *Secretary*.

A similar resolution was passed by the Committee on Aims and Organization upon the receipt of the news of Mr. Lord's death.

In 1902 he entered the employ of the General Electric Company as assistant attorney in the patent department, which position he resigned in 1904 to take charge of the patent departments of the Bullock Electric Manufacturing Company and the Allis-Chalmers Company, spending four years at Cincinnati and four years at Milwaukee in this work. During part of this period he was president of the Bullock Electric Company.

He studied law at the Georgetown University Law School, Washington, D. C., was admitted to the bar in Ohio, Wisconsin and Illinois and to practice in the federal courts and in the Supreme Court of the United States.

Mr. Lord was associate editor of the Encyclopedia of Engineering and wrote several textbooks for this publication when it was getting under way. He was a lecturer on patent law at Marquette University, and during the days of the War was a member of the War Committee of the Technical Societies of Chicago. For two years and until recently he was chairman of the committee on patents of the National Implement and Vehicle Association.

During 1918 he served as chairman of the Chicago Section of our Society and at the time of his death was a member of the Committee on Aims and Organization, in the work of which he took a keen and active interest. He was also interested in the pioneer movement inaugurated by the Western Society of Engineers, in Chicago, for a strong and all-embracing local engineering society in that city.

His ability as an inventor, though not widely known, is evidenced by a record of nearly forty United States patents.

Mr. Lord was a fellow of the American Institute of Electrical Engineers, a member of the Society of Automotive Engineers, the Engineers' Club of Chicago, the American Patent Law Association, the University Club of Washington, D. C., and several other technical and social organizations. He became a member of our Society in 1909. [F.F.F.]

HOWARD BAIRD LYON

Howard B. Lyon, instructor in the department of marine engineering and naval construction in the U. S. Naval Academy, Annapolis, Md., died on March 8, 1919. Mr. Lyon was born on September 24, 1890, in Baltimore, Md., and was educated in the schools of that city, being graduated from the Polytechnic Institute in 1910 with the degree of B.S.

He was first employed by the Consolidated Gas Co., in Baltimore, where his work dealt with the designing and selling of gas-burning appliances. In the latter part of 1911 he became connected with the Baltimore & Ohio Railroad Co. as draftsman on locomotive details. The following year he accepted a position with the U. S. Naval Engineering Experimental Station at Annapolis as assistant to the mechanical engineer, working up data from tests, etc. After four years he resigned to become an instructor in the Naval Academy, which position he held at the time of his death.

Mr. Lyon became a junior member of the Society in 1917.

CHARLES H. McGWIRE

Charles H. McGwire, assistant chief engineer for the Board of Public Utilities, Los Angeles, Cal., died on August 6, 1919. Mr. McGwire was born in 1868 in High Ridge, Conn. He studied for one year in Cornell University. He was formerly connected for varying periods with the following firms: Daniels & Fisher Store Co., Denver, Colo.; Barber Asphalt Paving Co., Denver; Frazer Mountain Copper Co., Twining, N. M.; Longmont Sugar Co., Longmont, Cal., Pacific Electric Railroad Co., Los Angeles, and the Busch-Sulzer Bros. Diesel-Engine Co., Los Angeles. Mr. McGwire became a member of the Society in 1909.

EDWARD BYRON McKINNEY

Edward B. McKinney was born on February 14, 1854, in New York City. He was educated in the public schools of the city and served an apprenticeship from 1870 to 1874 with the Delamater Iron Works, New York. He then entered the ocean merchant service as third assistant engineer and was gradually advanced to the position of chief engineer and from 1882 to 1889 served in this capacity on ocean-going steamers from the port of New York.

From 1889 to 1895 Mr. McKinney was chief engineer of the Louisiana Electric Light Co., New Orleans, and of other plants owned by them. In 1895 he was appointed chief engineer of the St. Charles St. Railroad Company's power house and also of the New Orleans Traction Co., holding both positions until 1902 when he was made superintendent of power, in charge of eight plants, of the New Orleans Railway & Light Co., which position he held until 1917 when he retired from active business.

Since then, however, Mr. McKinney did a remarkable piece of work in the renovating and installing of new machinery on the interned German ship *Teresa*, which was converted into a transport.

Mr. McKinney became a member of the Society in 1907. He died on March 30, 1919.

ANATOLE MALLET

Anatole Mallet, Honorary Member of The American Society of Mechanical Engineers, died in Nice, France, in October 1919. He was one of the few men whose name became the designation of a standard type of apparatus in a great industry. Born in 1837 at Carouge, and graduated in 1858 from the Central School of Arts and Manufactures in Paris, Mr. Mallet devoted all his attention for a number of years to problems in civil engineering. He was first connected with the Bureau of Direction of the General Company of Railroad Materials in France, and subsequently with the work on the Suez Canal. In 1864 he was engineer for the company which undertook the dredging of ports of Italy.

In 1867 he engaged for the first time in mechanical engineering, giving particular attention to steam engines with double expansion. The first application of this system to locomotives was made in 1876, when he introduced the first two-cylinder compound locomotive, which operated on the line from Bayonne to Biarritz. A fuel economy 20 per cent superior to that given by the standard types of the time was attained, which at once placed the inventor in the foremost ranks of locomotive designers.

The great success of the compound locomotive led to a material increase in the size of the tractive unit and to the development successively of three- and four-cylinder compound locomotives. In this way it became evident that the limit of size of a locomotive of rigid construction, especially on lines with sharp curves, would soon be reached, and the problem arose of building a locomotive of still larger size which would have the necessary flexibility. This problem was solved by Mallet in the design of the articulated locomotive, an invention which made his name familiar to every railroad engineer throughout the world. The most powerful locomotives now in existence are of the Mallet articulated type.

He was a member of the Society of Civil Engineers of France, The Society for the Encouragement of National Industry of France, and The Franklin Institute of Philadelphia. The French Society of Civil Engineers awarded him the Schneider prize in 1902, and the annual prizes in 1909 and 1911. He was made a Knight of the Legion of Honor in 1885 and promoted to Officer in 1905. The Institution of Mechanical Engineers of London awarded him a gold medal in 1915.

In addition to carrying on important engineering work, Mr. Mallet took an active part in the work of the French Society of Civil Engineers. From 1880 to within a few months of his death he was editor of the *Chronicle* of the Bulletin of the Society and furnished it with numerous technical notes and important memoirs, the last of which, treating of The Practical Evolution of the Steam Engine, earned for him the honors conferred by the Society.

CHARLES H. MANNING

Captain Charles H. Manning, honorary member of the Society and for many years prominent in engineering circles in New England, died at Manchester, N. H., on April 1. Captain Manning was born in Baltimore, Md., on June 9, 1844, and received his early education in that city and in Cambridge, Mass., later entering the Lawrence Scientific School of Harvard University, from which he was graduated in 1862 with the degree of B. S.

In February 1863 he was appointed third assistant engineer of the Navy. His knowledge and comprehensive grasp of scientific matters brought him to the attention of Chief Engineer Isherwood, who assigned him to the making of experiments on superheating steam on the U. S. S. *Adelaide*. He served on the *Adelaide* for two years, when he left her to join the sloop-of-war *Dacotah*. In 1870 Captain Manning was assigned to shore duty as an instructor in the Naval Academy where he remained five years. During this period he assisted in organizing a course of instruction for cadet engineers at the Academy, and this is considered one of his most valuable achievements.

Captain Manning also served as a member of the first Advisory Board that prescribed the specifications for the so-called "New Navy." Other members of the Board were Rear-Admiral John Rogers and Chief Engineers Benjamin F. Isherwood and Charles H. Loring.

In 1882 Captain Manning retired from active duty in the Navy and was not called into active service again until the Spanish-American War, when he was stationed at the Naval Station at Key West as chief engineer of the repair of the machinery of warships which gathered there. He was commissioned chief engineer on the retired list in February, 1911.

In 1883 Captain Manning became associated as chief engineer with the Amoskeag Manufacturing Company, Manchester, N. H., the largest cotton mills in the world. He had charge of all their power plants and saw the steam power plant grow from one with 870 sq. ft. of grate surface and burning about 400 tons per week, to one with 5800 sq. ft. of grate surface and capable of burning some 3000 tons of fuel per week.

The resourcefulness of Captain Manning is well typified by the following incident: In the fall of 1891 a 30-ft. flywheel burst, and being dissatisfied with the metal put into flywheel rims at the time, he designed a new wooden-rimmed wheel of the same diameter with a face of 108½ in. and a thickness of 12 in. This rim was made up of forty rings of ash, and was doubtless the largest wooden-rim wheel in the world.

In addition to his position with the Amoskeag Company, Captain Manning acted as consulting engineer for several other large mills. He was the designer of the well-known Manning boiler and was associated with many pioneer power-plant designs, including one of 200-hp. horizontal water turbine which at the time of its installation in 1885 was the largest of its kind.

In 1913 Captain Manning retired from the Amoskeag Company and opened offices as a consulting engineer.

As a resident of Manchester, Captain Manning was a very useful citizen. For 28 years he was a member of the Board of Water Commissioners — most of the time its president, and during his incumbency the system was greatly extended and improved. For 18 years he was a member of the school board, where his advice was much respected.

Captain Manning became a member of the Society in 1884 and was made an honorary member in 1913. From 1893 to 1895 he served as manager and from 1895 to 1897 as vice-president of the Society. He was also a member of the Army and Navy Club of New York, the American Society of Naval Engineers, the United States Naval Institute, the American Society of Naval Architects and Marine Engineers, the American Association for the Advancement of Science and the American Association of Cotton Manufacturers.

ERNEST G. MARBLE

Ernest G. Marble was born on September 19, 1876, in Methuen, Mass. He was educated in the schools of Methuen and later attended Tufts College, from which he was graduated in 1899 with the degree of B.S.

Upon graduation he worked for one year as mechanical engineer with C. W. Varney. He was then connected with the Duluth & Iron Range Railroad as a transit man and somewhat later as surveyor and draftsman with the Factory Mutual Fire Insurance Co., Boston, Mass. From 1904 to 1907 Mr. Marble was connected with the Otis Elevator Co., Philadelphia, Pa., as superintendent of construction in charge of the works in that city. He resigned from this position to become assistant general manager of the American Ship Windlass Co., Providence, R. I., and the following year was made treasurer with full charge of the financial and manufacturing interests of the business, specializing in shop costs. In 1912 he became associated with the American Engineering Co., Philadelphia, as general sales and works manager, which position he was forced to resign in 1918 because of ill health.

Mr. Marble became a member of the Society in 1909. He died on May 6, 1919.

LOUIS H. MARTELL

Louis H. Martell was born in Tusket, Nova Scotia, Canada, on February 26, 1864. When but a child his family moved to the United States and settled in Mt. Clemens, Mich., where he attended both grammar and high school.

In 1882 he entered the shops of the Michigan Central Railroad, at Jackson, Mich., where he served a three-years' apprenticeship, at the close of which he worked for a short time for the Walcott Shaper Co., Jackson. In 1886 Mr. Martell was employed in the shops of the Boston, Revere Beach & Lynn Railroad, a narrow-gage line running from Boston to Lynn. While with this concern, he studied marine engineering and took out his engineer's papers, for a short while running on the railroad company's ferry boats which ply between the Boston passenger station and the rail terminal in East Boston.

His next position was as engineer on the steamer *Longfellow*, sailing between Princetown and Boston, and later he was employed in the Merchants and Miners Line to Savannah and Southern ports. In 1892 he became connected with the Jos. Knox Shoe Machine Co., Lynn, Mass., remaining with that firm until the summer of 1895 when he entered the employ of W. B. Merrill & Co., makers of metallic packing. In 1897 he removed to Pittsburgh, Pa., as western representative of the firm and remained in charge of this branch until 1901 when he organized a packing manufacturing shop under the name of the Pitt Manufacturing Co. Later the plant was removed to Ellwood City, Pa., and finally merged with the Garlock Packing Co.

In 1908 he became associated with the Metallic Packing & Manufacturing Co., Elyria, Ohio, later incorporating under the name of the Martell Packings Co., and at the time of his death, January 31, 1919, Mr. Martell was president of the corporation.

Many devices for metallic packing were perfected and patented by Mr. Martell, who was considered an authority on the subject. Just previous to his death he was working on a perfected type of locomotive packing which bade fair to become the type universally used on American railroads. Mr. Martell became a member of the Society in 1906.

ALDEN R. MEEK

Alden R. Meek, manager of the New England district for the Westinghouse Electric & Manufacturing Co., died on November 11 in Harrisburg, Pa., of typhoid.

Mr. Meek was born on January 24, 1886, in Houtzdale, Pa. He was educated in the public schools of Harrisburg and later attended Pennsylvania State College, from which he was graduated in 1909, receiving his post-graduate degree in electrical engineering in 1914. Before entering college, Mr. Meek obtained his shop experience at the Harrisburg Foundry & Machine Works. Upon graduation he became connected with the Ridgway Dynamo & Engine Co., Ridgway, Pa., and was with this concern until 1917, when he resigned from his position as manager of the Boston office to become sales engineer in the turbine section of the power department of the Westinghouse Co. During the war he supervised the company's interests in Washington, D. C., returning to take the position of paper and rubber specialist in the industrial department, later becoming supply manager of the New England District.

Mr. Meek became an associate-member of the Society in 1918.

LOUIS MOHR

Louis Mohr was born on September 27, 1858, in Chicago, Ill. He attended both the grammar and high schools of that city and was for two years a special student at the Chicago Athenæum. He completed the four-year course in mechanical engineering at the University of Illinois in three years and in June of 1917 the University conferred upon him the degree of B.S. as of the class of 1882.

During the year of 1881 he conducted a series of experiments on slag and refuse of blast furnaces in the laboratory of the North Chicago Rolling Mills. He then became connected with the Old Excelsior Iron Works of which his father was president, as assistant draftsman and later as chief draftsman. In 1882 he associated himself with his father and brothers in the firm of John Mohr & Sons as consulting engineer. In 1893 in addition to his consulting work he assumed the duties of secretary of the firm and since 1917 until the time of his death, August 24, 1919, held the office of president.

Mr. Mohr was a member of a large number of clubs and societies among which are included the American Institute of Mining Engineers, the American Society of Naval Engineers and the Western Society of Engineers. He became a member of our Society in 1886.

FREDERICK LEO NEELY

Frederick L. Neely was born on January 1, 1883, in St. Petersburg, Pa. He was graduated from the Pennsylvania State College in 1904 and then became connected with the Allis-Chalmers Co., where he served a special apprenticeship course. For three years he was superintendent of erection of the company and the next two years he spent in connection with their sales work. During his connection with the company he installed, tested and superintended the erection of machinery in many cities throughout the United States.

His next position was with the Macon Railway & Lighting Co., Macon, Ga., as superintendent of the public utilities plants of the company.

In August, 1915, he entered the employ of the E. I. du Pont de Nemours Co., as assistant engineer, supervising the installation of power-plant equipment and power-distributing systems in connection with the building of the large smokeless-powder plant at Carney's Point, N. J. For two years he was engaged on mechanical, experimental and development work on smokeless-powder processes wherein he exhibited an unusual aptitude for scientific engineering investigation. When the U. S. Government commissioned the du Pont Co. to build the munitions plant near Nashville, Tenn., Mr. Neely was assigned to the design division where he filled most creditably a responsible position in connection with the design of the Old Hickory Smokeless Powder Factory.

At the time of his death, January 5, 1919, Mr. Neely was engaged upon important development work relative to the du Pont company's peace-time activities. He became a junior member of the Society in 1909 and was promoted to full membership in 1914.

ALLEN EUGENE NICHOLS

Allen E. Nichols was born on September 30, 1888, in Madison, Wis. At the age of six his parents moved to Philadelphia where he received his early education. Later he entered Purdue University from which he was graduated in 1910 with the degree of B.S. In 1913 he received the degree of C.E.

His first employment was with the Baltimore & Ohio Railroad as rodman and levelman in West Virginia on construction work. Early in 1911 he became associated with R. L. Sackett, consulting engineer for the State of Indiana, as assistant in the design of water works and sewage-disposal plants. His next position was with the Chicago & Western Indiana Railroad as assistant engineer on track elevation and maintenance work. From 1912 to 1914 he was connected with the firm of Alvord & Burdick, consulting engineers, as assistant engineer on design and construction for sewage-disposal and water-supply plants. The following year Mr. Nichols spent at the University of Pennsylvania, making a special study of the chemistry of water. From 1915 to January, 1918, he was engineer in direct charge of design, specifications and contracts for the Bureau of Waste Disposal for the City of Chicago. During this period he also made reports for the Department of Public Works on the mechanical and chemical character of garbage tankage produced by various cities of the United States and established a comparison with the standard requirements of the fertilizer trade. He also took charge of design and specifications for the water-purification plant of the City of Oshkosh, Wis.

When the United States entered the War Mr. Nichols endeavored to enlist in the Army but owing to a permanent injury he was not accepted by the military authorities. He thereupon entered the employ of the DuPont Engineering Co. as engineer of construction for a 70,000,000-gal. water-filtration plant for the Government powder factory at the Old Hickory Works at Nashville, Tenn., which position he retained until construction was completed. Early in 1919 Mr. Nichols, in conjunction with his father and brother, entered into a general engineering and contracting business with which work he was connected at the time of his death, May 8, 1919.

Mr. Nichols was an associate member of the American Society of Civil Engineers and a member of the Franklin Institute, the American Water Works Association and the Sons of the American Revolution. He became an associate-member of the Society in 1917.

DAVID BRAINERD OVIATT

David B. Oviatt was born in Salem, N. Y., on May 20, 1858. He served an apprenticeship in the shops of the Bennington Machine Works under Mr. Olin Scott from 1879 to 1883 when he entered Cornell University and was graduated in 1887 with the degree of M.E. The following year he received his master's degree in mechanical engineering. For one year he served as an instructor of drawing in the University of Tennessee, when he left to accept a similar position with the Georgia School of Technology, where he was located for about seven years.

Mr. Oviatt was at one time associated with the American Bridge Co., Trenton, N. J., and was also assistant engineer in the engineering department of the New York Rapid Transit Railroad Commission. For several years he was connected with the New York & Long Island Railroad Co., in their New York office as inspector of steel.

In April, 1909, Mr. Oviatt was appointed as assistant engineer to the Board of Water Supply, New York City, which position he was holding at the time of his death, February 12, 1919.

Mr. Oviatt was a member of various fraternal associations. He became a member of the Society in 1891.

CHARLES W. PRARAY

Charles W. Praray was born on May 29, 1873, in Pawtucket, R. I., and was educated in the public schools of that city. He served an apprenticeship in one of the local cotton mills and for ten years was connected with the B. B. & R. Knight Cotton Manufacturing Co. as master mechanic and engineer. For several years he was master mechanic for the Manomet Mills in New Bedford, Mass., when he became associated with the Holmes Manufacturing Co., also in New Bedford, as superintendent and architect.

About twelve years ago he opened a mill engineer's and architect's office in New Bedford and proved highly successful in his work. He was the designer of some six or seven mills in New Bedford and of a great number of others throughout New England. He was actively connected with this work at the time of his death, March 22, 1919.

Mr. Praray held a first-class Massachusetts engineer's license. He was a member of the National Association of Cotton Manufacturers and of a number of social clubs. He became a member of the Society in 1918.

ALFRED H. RAYNAL

Alfred H. Raynal, member of the Society since 1884 and manager from 1889 to 1902, died at his home in Washington, D. C., on March 1. Mr.

Raynal was born on August 5, 1848, in Hamburg, Germany, serving his apprenticeship as mechanic and draftsman in a marine-engineering establishment there. In May 1862 he came to the United States and worked as a machinist and draftsman for private concerns in New York. In 1863 he was engaged by the Navy Department as draftsman on the design of twenty light-draft monitors. In 1864 he made the drawings for and superintended the construction of the monitor *Squando* at the works of McKay Aldus at East Boston, Mass.

He next entered the employ of the Babcock & Wilcox Co., Providence, R. I., working out details of their engine and boiler which latter is now used so extensively in the U. S. Navy. In 1870 Mr. Raynal became superintendent of the works of Poole & Hunt, Baltimore, who were building engines and boilers under the Babcock & Wilcox patents. From 1880 to 1884 he was superintendent of the Wheelock Engine Works, Worcester, Mass. He was next connected with the DeLamater Iron Works, New York, as superintendent, where he did considerable work for Capt. John Ericsson, building his submarine gun, high-expansion engine, sun motor, etc. In recognition of his services to him, Captain Ericsson presented Mr. Raynal with a rare and valuable book on his inventions. This book was presented to the Society by Mr. Raynal for record.

From 1890 to 1897 he successively held the following positions: Superintendent of the Richmond Locomotive Works, building the machinery of the U. S. battleship *Texas;* superintendent of Samuel Moore & Son's Shipyard, Elizabeth, N. J., building the revenue steamer *Maple* and the practice cruiser *Bancroft;* superintendent of the Corliss Engine Co., Providence, R. I.; superintendent of the Walker Co., Cleveland, Ohio. In 1897 Mr. Raynal studied patent law, working in an attorney's office in Cleveland.

In 1898, during the Spanish War, at the personal request of Admiral Melville, he entered the Navy Department as draftsman to design engines for the first sixteen torpedo-boat destroyers, as his experience in building successfully the spider-frame high-speed engines of the *Bancroft* had been unusually valuable.

From that time Mr. Raynal's work was in the Navy Department where he was of signal service in the Bureau of Steam Engineering. Mr. Raynal acted as expert engineer in the courts in suits resulting from explosions of boilers, bursting of flywheels, etc., and in admiralty cases.

He was a member of the American Society of Naval Engineers and Marine Architects, the American Society of Naval Engineers, the Washington Society of Engineers, and the American Society of Marine Draftsmen.

CYRUS T. RAYNER

Cyrus T. Rayner was born in New York City on January 28, 1879. He was educated in Southern schools and was graduated from Tulane University in 1902 with the degree of M.E. He spent one year in graduate work when he became connected with the Mississippi River Commission as junior engineer in charge of local surveys. From 1911 to 1912 he was associated with C. T. Rayner & Sons, consulting engineers. In 1912 he became first assistant engineer on the New Orleans Levee Board and four years later was made chief engineer, which position he was holding at the time of his death, January 19, 1919.

Mr. Rayner became a member of the Society in 1916.

LLOYD CECIL REYNOLDS

Lloyd C. Reynolds was born in Atglen, Pa., on November 20, 1893. He received his early education in the local schools and later entered Drexel Institute, taking the mechanical-engineering course, from which he was graduated in 1913. He was associated from 1903 to 1916 with the Overland Motor Company, Philadelphia, the Chambersburg Engineering Company, Chambersburg, Pa., and the Remington Arms Company, Eddystone, Pa. From July 1916, to the time of his death, January 6, 1919, Mr. Reynolds was employed as assistant engineer in the steam-engineering department of the Midvale Steel & Ordnance Co., Coatesville, Pa.

Mr. Reynolds was a member of the Drexel Club of Engineers. He became a junior member of the Society in 1917.

CHARLES B. RICHARDS

Charles Brinckerhoff Richards, Higgin professor of mechanical engineering in the Sheffield Scientific School, Yale University, for a quarter of a century and since 1909 professor emeritus, died at his home in New Haven, Conn., on Sunday, April 20, 1919, in his eighty-sixth year. Professor Richards was born in Brooklyn on December 23, 1833, and was educated in private schools in that vicinity.

In an article for the *Yale Daily News* Professor Lockwood of Sheffield writes:

"In the death of Professor Charles B. Richards, the mechanical engineering profession loses one of its most prominent members. His early life was spent in Hartford, Conn., at a period when engineering colleges were unknown. He was obliged, therefore, to acquire his theoretical education by the power of personal application, of broad reading and diligent study — this all carried on while busily engaged in practical daily work along manufacturing lines.

"His school was the famous Colt's Armory where his mechanical genius was early recognized. At the age of twenty-seven he opened an office in New York as consulting engineer and at once leaped into wide prominence by his invention of the steam-engine indicator, which made the name of Richards a familiar one wherever steam engines were used. This instrument was produced at a time when the high-speed engine was in its infancy and aided greatly in its development. For this achievement he was made a Chevalier of the Legion of Honor of France, as well as receiving medals from the London Exposition of 1862, the American Institute in 1869, and the French Exposition in 1878.

"With the outbreak of the Civil War Mr. Richards was recalled to Colt's as their assistant superintendent and consulting engineer, where he spent nineteen years of active life. He developed the platform-scale testing machine for testing the strength of metals and was also an authority on the heating and ventilation of buildings, acting as consulting engineer for several large projects, among them the Capitol at Hartford and several of the Yale University buildings. His fertile mind was constantly at work on inventive problems. He was deeply interested in the microscope and made several improvements for which he was granted patents.

"In 1880 Mr. Richards became superintendent of the Southwark Foundry and Machine Company in Philadelphia, manufacturers of the Porter-Allen high-speed engine, marine and other large machinery. Four years later he was invited to become professor of mechanical engineering at the Sheffield Scientific School of Yale University and entered upon the new work of teaching. An inborn taste for study and research undoubtedly led to this choice. For twenty-five years Professor Richards served as the head of the mechanical engineering department, where successive classes were inspired by his friendly spirit and high ideals.

"He was engaged in occasional activities in addition to teaching. He was an American Commissioner at the Paris Exposition in 1889, and edited the report of mechanical appliances. As associate editor he was responsible for the technical terms in two editions of Webster's New International Dictionary.

"In 1909 at the age of seventy-five Professor Richards retired from active work. The esteem of his classes was expressed on this occasion by their presentation to the school of an oil portrait of Professor Richards. This portrait now hangs in the faculty room of the Sheffield Scientific School.

"Professor Richards was a stern disciplinarian and believed in the doctrine of work. Yet his friendly interest in the individual student and his enthusiasm for engineering aroused and maintained a helpful spirit of coöperation with his classes while his great ability and wide experience were recognized and admired."

Professor Richards was a member of the Société Industriellè de Mulhouse, Alsace, the Society of Naval Architects and Marine Engineers, the Connecticut Academy of Sciences, and a Fellow of the American Academy of Arts and Sciences. He was a charter member of our Society, being present at the organization meeting on April 7, 1880.

BARTON J. ROBINSON

Barton J. Robinson was born in Claiborne County, Miss., on February 14, 1868. He received his early education in the public schools of New Orleans and served his apprenticeship at the machinist trade with the Shakespeare Iron Works, New Orleans. While working there he attended night school to perfect himself in his chosen trade. He worked as a machinist for a number of years in shops in and around New Orleans, and for several years after leaving that city was employed in railroad work. The last railroad position he held was as machine-shop foreman for the Yazoo & Mississippi Valley Railroad at Vicksburg, which position he gave up in 1896 to start in business for himself under the firm name of the B. J. Robinson Machine Works at Vicksburg.

He specialized in rebuilding saw-mill and cottonseed-oil mill machinery. He started on a small scale and through his endeavors built up a business well-known in the South. At the time of his death, October 4, 1919, he was manager and proprietor.

Mr. Robinson was prominent in fraternal circles and belonged to several local business organizations and clubs. He became an associate member of the society in 1917.

HENRY FREDERICK ROSENOW

Henry F. Rosenow was born on September 7, 1871, in Baltimore, Md., and received his early education in the schools of Michigan. He served his apprenticeship on the Great Lake steamers, starting as fireman and being advanced to the position of engineer, thus acquiring his first engineering knowledge which he supplemented by evening and correspondence courses.

He was connected for varying periods with the Regina Flour Manufacturing Co., the Imperial Light & Power Co., the Nammer Bros. White Lead Co., and the Laclede Power Co. For fifteen years Mr. Rosenow was associated with the Brown Shoe Co., St. Louis, as chief engineer, and was responsible for the design and construction of eight of their plants. At the time of his death, October 18, 1918, he was manager of the Baetz Heating Department of Skinner Brothers Manufacturing Co.

In 1915 Mr. Rosenow took up the study of law in the Benton College of Law and had conferred upon him the degree of LL.D. Three years later he was admitted to the bar in the State of Missouri. He was a member of the Engineers' Club of St. Louis, the National Association of Stationary Engineers, The Universal Craftsman and several fraternal organizations. He became a member of the Society in 1917.

FREDERICK SARGENT

Frederick Sargent, senior member of the firm of Sargent & Lundy, of Chicago, and probably the most prominent consulting engineer in the United States specializing in the design of electrical generating sections, died at his home, in Glencoe, Ill., July 26, having been taken ill while abroad. An Englishman by birth, Mr. Sargent had made numerous trips to his native country, the last of which was made in April and May of this year in company with his close friend, Samuel Insull.

Frederick Sargent was born in Liskeard, Cornwall, England, on Nov. 10, 1859, which is also the exact date of the birth of Samuel Insull, another Englishman and Chicagoan, with whom Mr. Sargent was destined to be intimately associated during practically all of his engineering activities. His people were of the farming class, but young Sargent developed a decided mechanical bent, and eight years of his boyhood and youth were spent in acquiring practical mechanical knowledge and experience in the works of John Elder & Co., the great shipbuilders on the River Clyde, near Glasgow. During this time he gained an extensive and practical knowledge of mechanical engineering, paying particular attention to heavy machinery. He also improved his education by going to night school at Glasgow University. Coming to the United States about 1880, he found employment in Eastern shipbuilding yards as a designer of steam engines. He then went West as a designer for the Sioux City (Iowa) Engine Co. A year or so later he accepted a position with E. P. Allis & Co., of Milwaukee, predecessors of the present Allis-Chalmers Manufacturing Co. Here he attracted the attention of the officers of the Western Edison Light Co., organized in Chicago in 1882 to exploit the electric-lighting inventions of Thomas A. Edison in the West, and in the fall of 1884 he moved to Chicago and began his career as an electrical engineer in that city.

Succeeding the Western Edison Light Co., the Chicago Edison Co. was formed in 1887. This was the first distinctively Edison central-station company in Chicago. Mr. Sargent became its consulting engineer, and he has been consulting engineer of that company and its successor, the present Commonwealth Edison Co., ever since.

About 1889 he went to New York under contract with the Edison United Manufacturing Co. In this position he had general charge of all the work done by that company in the United States and Canada. Shortly after this the company in New York was reorganized as the Edison General Electric Co. Mr. Sargent was made assistant chief engineer of the new corporation, of which Samuel Insull was vice-president in charge of manufacturing. But Mr. Sargent had determined to open an office of his own, and, in August 1890, he returned to Chicago and established himself as an independent electrical and mechanical engineer. The firm of Sargent & Lundy was formed in 1891.

In 1891 and 1892 Mr. Sargent was consulting electrical engineer for the World's Columbian Exposition, and he designed the power plant and had much to do with the other mechanical and electrical features of the great World's Fair of 1893.

The original Edison central station in Chicago was built about 1889. Mr. Sargent made the plans for the machinery layout of that station. In 1892 Samuel Insull came to Chicago as president of the Chicago Edison Co., and that company at once took on a new lease of life. Under Mr. Insull's direction the old Harrison Street station, recently torn down to make way for the Union Station railroad improvements, was built. Mr. Sargent was the designer of that station, also the Fisk Street Station, the Quarry Street Station and the Northwest generating stations of the Commonwealth Edison Company.

He was one of the first mechanical engineers who recognized the great part that the steam turbine was destined to play in the development of electric generating stations. The Fisk Street Station was the pioneer of all the large turbine central stations of the world, and it became deservedly famous for its many original features of design and for its simplicity and economy of operation. After this station had been in operation for a short time, Mr. Sargent, at the request of Mr. Insull, President of the Commonwealth Edison Co., went to London to follow the inquiry of the Parliamentary Committee in charge of the London Power-Supply Bill. This hearing crystallized in Mr. Sargent's mind some ideas he had been developing about the importance of unified power supply for great industrial centers so as to reduce the cost of production, and on his return he submitted his ideas to Mr. Insull, and they were worked out in the power-station development of the Commonwealth Edison Company.

Mr. Sargent's engineering work was not confined to Chicago. He was consulting engineer for many of the important electric-light and power companies throughout the country, including the Edison Electric Illuminating Co., Boston; American Gas and Electric Co., New York; Electric Bond and Share Co., New York; the Union Gas and Electric Co., Cincinnati, and many other smaller organizations.

He designed the great combined central power station of the American Gas and Electric Co., and the West Penn Power Co., located on the Ohio River, north of Wheeling, W. Va., which was the first large electric station to be built in a favorable locality near a coal mine for the distribution of power to industrial centers at long distances.

He designed the great new station of the Union Gas and Electric Co., at Cincinnati, which was recently completed. He also designed the new station for the Kansas City Light and Power Co., which is soon to be put into operation. He went to Chile in 1916 as consulting engineer for the Guggenheim mining interests on the development of a power supply for their mine at Chuquicamata.

During the war Mr. Sargent was consulting engineer for the power station of the Edgewood Arsenal, at Edgewood, Md., and also consulting engineer for the United States Government in other wartime projects demanding the application of power on a large scale.

In his profession Mr. Sargent was noteworthy for the clear vision and strong common sense with which he grappled with the essentials of an engineering problem. He was simple, clear, direct and practical. He was a man of broad outlook, tolerant, modest, seeking to achieve results rather than to uphold theories. And he was eminently successful in obtaining results, for his electrical generating stations were milestones of achievement in the economical production of electrical energy.

An idea of the esteem with which Mr. Sargent was held by his business associates was shown in an interview with Wm. S. Monroe, his friend and partner in the firm of Sargent & Lundy for many years, who said:

"Mr. Sargent had an exceptionally keen and active intellect and a vigorous and forceful personality. He was a man of absolute integrity and fearless independence and high idealism in his work. He had an infallible intuition regarding engineering and scientific matters, and the responsible men in the companies for which he was doing his engineering learned to place the utmost confidence in his judgment. He had a remarkable combination of extreme daring and careful conservatism. With a broad and ambitious view of important and fundamental principles of his engineering work, Mr. Sargent combined an accurate knowledge of all the underlying details, and no detail was too small for his personal attention.

"He kept in close touch with everything that was new in the engineering profession. He was a great traveler and made repeated trips to Europe as well as through this country in order to post himself on the important developments not only in the direct line of his own work but in all departments of the engineering field.

"His idealism was at times almost prophetic and he was very ambitious for the highest achievements in his work, but his idealism was held in restraint by a practical common-sense judgment, which combined to give a distinct originality to every new power station which he designed, and made it systematic and harmonious, economical and a perfect working machine."

Mr. Sargent was awarded a medal by the World's Columbian Exposition in 1893. He was a member of the jury of awards in power engineering at the St. Louis Exposition of 1904. He was a member of several societies and clubs, including the Western Society of Engineers, University Club, Chicago Yacht Club, and the Engineers' Club, New York. He became a member of our Society in 1901.

He is survived by a widow, one daughter and two sons.

ALBERT SCHMID

Albert Schmid, who was so closely identified with the early development of electrical machinery in the United States and prominent in the electrical world of France, Switzerland, Italy and Great Britain, died on December 31, 1919, in New York.

Mr. Schmid was born in Zurich, Switzerland, in 1857, and received his education in that city. He began his real career by entering the employ of the

French Westinghouse Air Brake Co., where Mr. Westinghouse met him in the early eighties and being impressed by his ability invited him to come to this country.

Soon after his arrival Mr. Schmid turned his attention to designing work for the Westinghouse Air Brake Co., then located in Allegheny, Pa., where his keen mechanical perception and insight brought him rapid advancement. When Mr. Westinghouse became interested in the Union Switch & Signal Co. and started there his original electrical work, he engaged Mr. Schmid as his chief designer and engineer in that field. In 1886 he was transferred to the newly created Westinghouse Electric Co., becoming its first chief engineer and in 1896 its general superintendent.

In 1897 he went to Europe for the purpose of studying the continental development in the electrical art and the manufacturing possibilities there, and as a result of this trip, the formation of the French Westinghouse Co. was soon under way. He was made director general of that organization. He also held the positions of director of the Westinghouse Electric Co., Ltd., England, president of the Compagnie des Lampes a Filament Metallique of France and at the time of his death in addition to his position as consulting engineer for the American Westinghouse Co., represented the Westinghouse Lamp Co., and had general supervision of its interests abroad.

It can be said truly that Mr. Schmid ranked foremost in the field of mechanical design among the engineers of the last century, and the creations of his mind constitute an enduring monument to his genius.

Mr. Schmid became a member of the Society in 1890.

GEORGE SCHUHMANN

George Schuhmann, vice-president and general manager of the Reading Iron Co., Reading, Pa., died on November 12, 1919. Mr. Schuhmann was born on October 18, 1855, in Weinheim, Baden, Germany, and was educated in the industrial and high schools both of that city and of Mannheim.

Having served an apprenticeship as machinist, he came to this country in 1874 where he was employed as draftsman by Nathan & Dreyfus, New York City. He spent a short period with the Brown & Sharpe Manufacturing Co., Providence, R. I., as machinist when he became connected with P. H. & F. M. Root, Connersville, Ind., as draftsman. From 1877 to 1891 he was associated for various intervals with the Philadelphia Hydraulic Works, as draftsman, the Gregg Brick Machine Co., Philadelphia, as draftsman on brick-making machinery, and with the Reading Iron Works, Reading, Pa., as chief draftsman and mechanical engineer. From 1891 to 1919 he held successively the positions of assistant, department and general superintendent, general manager and vice-president of the Reading Iron Works, successor to the Reading Iron Works.

Mr. Schuhmann patented many rolling-mill and pipe-mill devices, and was a recognized authority on wrought iron and the manufacture of welded pipe, as well as the author of a number of technical treatises on similar subjects. He was a member of the American Society for Testing Materials. He became a member of our Society in 1884.

GEORGE ROBERT SHEPHEARD

George R. Shepheard was born on October 24, 1889, in Wilmington, Del. He obtained his early education in the grammar and high schools of Wilmington, while his later training consisted of home study and courses taken in the International Correspondence Schools.

He was first connected with the Taylor Iron & Steel Co., High Bridge, N. J., in estimating and designing. He left that concern to work on general drafting and repairs in the machine shop of the Edgar Allen American Manganese Steel Co., New Castle, Del. For a short period following this he was with the J. Morton Poole Co., Wilmington, Del., as a machine-tool designer. In December 1912 he accepted a position with the E. I. du Pont Co., Wilmington, in their experimental station for designing experimental apparatus for chemists. In 1913 he became associated with the Viscose Co., Marcus Hook, Pa., where his duties dealt with the designing of buildings and special machinery. In 1916 he was made chief engineer of the company and in this capacity superintended the building construction and installation of machinery.

He was holding this position when he contracted influenza which developed into pneumonia, resulting in his death on May 1, 1919. Mr. Shepheard became an associate-member of the Society in 1918.

ANGUS SINCLAIR

Angus Sinclair, D.E., founder and for the last 32 years editor-in-chief of *Railway and Locomotive Engineering*, died January 1, 1919, at his home in Milburn, N. J. He was born at Forfar, Scotland, and began his railroad career as a telegraph operator, gaining his engineering knowledge at the shops of the Scottish Northeastern Railway at Arbroath. After some service as a marine engineer he came to America and again took up railroading, first with the Erie and later as a locomotive engineer on the Burlington, Cedar Rapids and Northern. He attended the chemistry classes of the Iowa State University and was later appointed chemist on the railway, combined with the duties of roundhouse foreman. It was during this period that he first gave serious attention to the problem of fuel economy and smoke prevention. His methods, which met with considerable opposition at first, are now universally approved.

In 1883 he joined the editorial staff of the *American Machinist* and later became proprietor and editor of *Railway and Locomotive Engineering*. In a short time this paper became a leading authority in its field and has maintained its high character and standing ever since. Dr. Sinclair was the author of many popular books on engineering subjects. In 1908 he received the honorary degree of Doctor of Engineering from Purdue University. About this time he was also appointed special technical instructor in the mechanical department of the Erie Railroad. He traveled extensively in Europe as well as in America and was everywhere received as among the foremost authorities on all matters connected with the mechanical department of railways. He was closely identified with the work of many of the leading engineering societies in America and in Europe. He was elected a member of the American Railway Master Mechanics' Association in 1873. He served as secretary of the association from 1887 to 1896, was elected treasurer in 1900 and served continuously until the time of his death.

He became a member of the Master Car Builders' Association in 1873. He was the first president of the New Jersey Automobile and Motor Club. He was a delegate to the International Railway Congress, held at Washington, D. C., St. Louis, Mo., and Berne, Switzerland.

Among the societies which he aided in establishing was the Traveling Engineers' Association, founded in his office in 1892. He was a Knight Templar in the Masonic fraternity, a governor in the St. Andrew's Society, ex-president of the Burns Society, besides being a member of the American Railway Guild, Lawyers' Club, New York Railway Club and numerous railway, Scottish and other societies.

Dr. Sinclair became a member of our Society in 1883.

JOHANNES GEORGE STENGER

Johannes George Stenger was born in March 1884, in Rotterdam, Holland, and received his early education in the schools there. Later he took a course in marine engineering in the Technical School of Rotterdam.

He served an apprenticeship with the Royal Netherland Engine Factory, Helmond, Holland, for one year. From 1902 to 1904 he worked for the Royal Netherland Engine & Ship Building Co. in Rotterdam as a machinist. In 1905 he became connected with F. A. Smulders & Co., Schiedam, Holland, as engine draftsman.

The following year Mr. Stenger came to the United States where he was associated for short periods with the W. & A. Fletcher Co., Hoboken, as engine draftsman and with the John N. Robbins Drydock Co., Brooklyn, in a similar capacity. In 1907 he accepted the position of draftsman and checker in the marine department of the Bayonne Works of the Babcock & Wilcox Co., and four years later became chief draftsman of that department, which position he was holding at the time of his death, March 27, 1919.

Mr. Stenger became a member of the Society in 1918.

CARL A. STROM

Carl A. Strom was born on August 22, 1868, in Matala, Sweden. He was graduated from the Royal Polytechnic Institute, Stockholm, in 1889, and then served a five-year machine-shop apprenticeship in Motala and in Atlas.

His first connection in the United States was with the Chicago, Omaha and Southwestern Railroad as a draftsman. In 1898 he was appointed general foreman of the Paducah Shops of the Illinois Central Railroad and in 1900 was made mechanical engineer on the same road. Later he held the position of superintendent of motive power.

Mr. Strom was the chief mechanical engineer of the Isthmian Canal Commission and was engaged for several years in the construction of the Panama Canal. He was next associated with the American Locomotive Co., at Richmond, Va., and was in charge of the steam shovel and dredge department, which was later transferred to the old Rogers Locomotive Works, Paterson, N. J., at which plant Mr. Strom served as manager. In January, 1912, he was transferred to the Pittsburgh Works of the company and was manager there until September, 1917, when because of his health he again returned as manager of the Paterson

Works which position he was holding at the time of his death, March 26, 1919.

Mr. Strom became a member of the Society in 1905.

JOHN M. TOPPIN

John M. Toppin was born in Penrith, England, May, 1890, and received his early education in English schools. Upon his coming to the United States in 1903 he supplemented this early training by attending and being graduated from the Hartford high school and by courses taken in correspondence schools.

He was first employed by the Hartford Rubber Co. with which concern he worked for about a year when he became connected with the General Electric Co., Pittsfield, Mass., as architectural draftsman and later as assistant mechanical engineer of the small-motor department.

He was next associated with the Rhodes Manufacturing Co., Hartford, as general manager, resigning from this position to open an office as mechanical and consulting engineer.

Mr. Toppin became a junior member of the Society in 1915. He died in May 1918.

WALTER V. TURNER

Walter V. Turner, manager of engineering of the Westinghouse Air Brake Co., a leading mechanical engineer of this country and an inventor of international prominence, died at the Columbia Hospital, Wilkinsburg, Pa., on January 9 after an illness of eight weeks.

Mr. Turner was born in 1866 in Epping Forest, Essex, England, and was educated in the Textile Technical School of Yorkshire. He came to this country in 1888 and settled in the West where he became the manager of a sheep and cattle company, later going into the ranching business for himself. For several years he was general air-brake inspector for the Atchison, Topeka and Santa Fe Railroad and in 1903 became connected with the Westinghouse Air Brake Co. as engineer, remaining with that company and associated interests until his death.

More than four hundred devices in use on railroads and in industrial plants were patented by Mr. Turner, his air brake and other safety inventions having had much to do with the speeding up of railroad travel and also in reducing the number of accidents. As a reward for his achievements as an inventor in the development of the air brake he was awarded the Elliott Cresson and the Edward Longstreth medals by the Franklin Institute, of Philadelphia. He was made a Doctor of Engineering by the University of Pittsburgh. The Air Brake Association of America at its convention at Richmond, Va., 1912, passed a resolution expressing its recognition of him as the leading air-brake expert.

Mr. Turner was the author of several books on engineering subjects and a contributor to scientific periodicals in this country. He became a member of the Society in 1913.

RICHARD CHARLES VEIT

Richard C. Veit, head of the marine department of the Standard Oil Co., New York, died on August 29, 1919. Mr. Veit was born in New York City on November 17, 1855. He was educated in the public schools of the city and when comparatively young, secured employment with the firm of Rockefeller, Andrews

& Flagler, the predecessor of the Standard Oil Co., with which concern he remained for fifty-two years. He was advanced from one position to another, gradually acquiring a comprehensive grasp of the oil industry. In 1880 he was made chief of the lighterage department at the time when the Standard Oil Co. had its beginning. For many years Mr. Veit was a director of the company and in 1911 he became secretary.

Mr. Veit was affiliated with various philanthropic institutions, among these being the old J. Hood Wright Memorial Hospital, the American Museum of Natural History, the Metropolitan Museum of Art and the New York Zoölogical Society. He was a member of a number of clubs and social organizations. He became a member of the Society in 1891.

HENRY WEICKEL

Henry Weickel, who died on November 28, 1919, was born in Germany on October 31, 1852. He was graduated from the Technical Institute at Kaiserslantern, Bavaria, and came to the United States shortly afterward, in 1872.

He entered the employ of the Government as a draftsman at the Watertown Arsenal, Mass., and remained there five years when he became connected with the Hinckley Locomotive Works, Boston, in a similar capacity. In 1882 he accepted a position with the Yale & Towne Manufacturing Co., Stamford, Conn., as a designer of cranes and hoisting machinery, making this branch of the profession his life work from that time on. For thirteen years Mr. Weickel was connected with this concern when they disposed of their crane interests to the Brown Hoisting and Conveying Machinery Co., Cleveland, Ohio, He then became identified with the Cleveland company, in whose employ he remained until the fall of 1900, when he accepted a position with the Pawling & Harnischfeger Co., Milwaukee, manufacturers of electric cranes. He was actively identified with the designing of cranes, and for a number of years, up to the time of his death, was consulting engineer for the company.

Mr. Weickel became a member of the Society in 1895. He was a charter member and past president of the Milwaukee Section of the Society.

SAMUEL T. WELLMAN

Samuel T. Wellman, a past-president of the Society, and a pioneer in the world's steel industry, whom Charles Schwab once characterized as "the man who did more than any other living person in the development of steel," died suddenly from heart trouble on July 11, 1919, in Stratton, Me., while on his way to camp. His death came as a shock to his very large circle of friends, many of whom had only recently met him at the Spring Meeting of the A.S.M.E., at Detroit.

Mr. Wellman was born on February 5, 1847, in Wareham, Mass. At that time his father was superintendent of the Nashua Iron Co., of Nashua, N. H. He was educated in the schools of Nashua and in Norwich University, Norwich, Vt., where he spent one year in the engineering course. During the Civil War he served as a corporal in the First New Hampshire Heavy Artillery.

Perhaps the most interesting account of Mr. Wellman's entrance into and activities connected with the steel business is that given in a letter which he wrote to the Cleveland Engineering Society at the time of his election to honorary membership in that body. Mr. Wellman wrote:

"Perhaps the story of my leaving home at that time might be interesting. My father was superintendent of a small iron works among the hills of New Hampshire, which among other activities included the rolling of steel locomotive tires which they made from hammered steel blooms imported from England. To properly heat these blooms, they had arranged with C. W. Siemens to build a regenerative gas furnace. The drawings were sent over and turned over to me, who had never seen a drawing of that kind before. My father asked me if I thought I could build the furnace and have it exactly like the drawings. After studying them over a little while, I told him that I thought I was equal to the task. He told me to get the furnace built at the earliest possible moment. I had just finished the furnace and had a drying-out fire in it when a big black-whiskered Englishman walked into the office and announced that he was the Siemens engineer who had been sent from England to build this particular furnace. My father turned him over to me to go out into the works and show him where the furnace was to be located, which I did, and my English friend was very much amazed to find the furnace all finished. He was pleased as well, and said, 'Why, we have only to start the furnace now; you have made a proper job of building it.' So everything else being ready, we started up the gas producers and the furnace and everything worked to perfection. The engineer was so pleased with what I had done that when he left he asked me to go with him to Pittsburgh as his assistant in the starting of a crucible-steel furnace which was finished and waiting for him to start. It is needless to say that I did not have to have a second invitation, but went to Pittsburgh with him. This was in 1867, and I spent over a year there. The first few months of that time were spent starting up and operating the first crucible-steel furnace built in America, at the works of Anderson, Cook & Co. To show what a tremendous saving this furnace was over the old coke-fired furnaces, I will only say that it melted a ton of steel with an average of 1000 lb. of nut coal which cost less than \$1, while to melt a ton of steel in crucibles in the old-fashioned coke furnaces took three tons of the very best coke, costing anywhere from \$2 to \$10 per ton. This style of furnace was a great success and in a very few years had driven the coke furnace out of use.

"From Anderson, Cook & Co. I went to Singer, Nimick & Co's. works, where I built two crucible-steel melting furnaces of the same type as the one which I went to Pittsburgh to start. After that I spent some time in the office of the Siemens agents in Boston, and also its steel works in different parts of the country starting crucible-steel furnaces of the same type. I then built a crucible-steel melting furnace at the Chrome Steel Works in Brooklyn, N. Y., which was a success. From there I went to the Bay State Iron Works, in South Boston, Mass. (having separated myself from the agents of the Siemens furnace some time before this), where I built the first open-hearth furnace that was a commercial success in the United States. This was a pronounced success in a great many ways, making a quality of steel which up to that time had not been reached in this country. The principal use to which it was put at that time was in the manufacture of locomotive fireboxes. From there I went back to the old works in New Hampshire, where my father was still superintendent, and built for them an open-hearth furnace, a plate-rolling and bar-rolling mill.

"I then came to Cleveland in 1873 to design and build the Otis Steel Works, with which I was connected as engineer and superintendent for sixteen years. It is useless for me to say very much about the history of the Otis Steel Co., as it is too well known here. But there are two inventions which I worked out during the time I was connected with the Otis company that are today absolutely indispensable to the economic operation of any open-hearth steel works. I refer to the open-hearth charging machine and the use of the electromagnet for handling pig iron and scrap steel. Just a few figures will give you a little idea of their importance to the trade and what they are saving every day.

"There were made in this country in 1916 approximately 39,000,000 tons of pig iron and about the same quantity of steel ingots of all kinds. Very conservative figures show that at least half of this, or say 20,000,000 tons, was pig iron and scrap handled and used in open-hearth furnaces. This was all handled by the open-hearth charging machine and electromagnet at least once, the bulk of it twice, and a great deal of it three times. By the use of the electric open-hearth charging machine, the direct saving in labor is estimated by one of the large users at 25 cents per ton. This was about 10 years ago, and of course labor is much higher today. At the same time he estimated the indirect saving in handling of the material charged into the open-hearth furnace (calling it only 20,000,000 tons) of \$10,000,000. If we go back 17 years, this saving amounts to not less than \$85,000,000. This is a big sum of money, but the estimate is far below the maximum amount which has been actually saved. The saving in labor by the use of the electromagnet in the United States per year at the rate pig iron and scrap are being handled today is not less than \$1,500,000. We can very safely say that in the last ten years at least five times that amount, or the sum of \$7,500,000, has been saved; or a total for both of these inventions of nearly \$100,000,000 — a saving of which any inventor might well be proud. Every open-hearth plant of any size in the world today is equipped with these inventions, and they are considered as much a necessary part of the equipment as the furnace itself."

Mr. Wellman's machine for charging open-hearth furnaces with white-hot steel was his greatest contribution to steel making and has been adopted by the steel concerns of the world, earning the inventor international fame. Machinery built under his patents is being used in Great Britain, France, India, Japan, Russia, Spain, Egypt and Germany.

Mr. Wellman was a member of the American Society of Civil Engineers, the American Institute of Mining Engineers, the British Iron and Steel Institute, the British Institution of Mechanical Engineers and the Cleveland Engineering Society, in which latter he had held the offices of vice-president and president. He was a life member of our Society, joining in 1881, and served it respectively as manager from 1885 to 1888, vice-president from 1896 to 1898, and president in 1901.

At the age of twenty-one Mr. Wellman married Julia A. Ballard at Stoneham, Mass., who died in Cleveland, July 3, 1914. He is survived by two daughters, Mrs. A. D. Hatfield, Mrs. C. W. Comstock, and three sons, W. S. Wellman, president of the Wellman Products Co., and M. C. and F. S. Wellman, of the Wellman Bronze Co.

The funeral service was held Monday afternoon, July 14, at the Euclid Avenue Congregational Church, Cleveland, and was attended by many prominent engineers, manufacturers, and business men. Robert W. Hunt and Ambrose Swasey represented the Society. Members of the Cleveland Engineering Society and delegations from the Wellman-Seaver-Morgan Co., and Wellman Bronze Co. were present.

WILLIAM TRIMBLE WHEELER

William T. Wheeler was born on August 22, 1862, in Calais, Me. He was educated in the public schools of that city and Pratt Institute, Brooklyn.

From 1880 to 1889 he was connected with the engineering department of the Equitable Life Insurance Co. Building in New York City, where he gained practical experience in the erection of engines, boilers, dynamos, elevators, steam heating and electric equipment. The year of 1889 he was assistant to the superintendent of the Equitable Building and his work consisted of the alteration of the steam plant. The following year Mr. Wheeler visited practically all the large European cities on a tour of inspection of the various Equitable buildings located there. Upon his return he installed the steam, electrical and elevator plants in the company's buildings at St. Louis, Mo., and Des Moines, Iowa.

After some twenty years of service with this concern he accepted the position of chief engineer of the New York Life Insurance Co. Building, New York City, installing a new system of steam heating and connecting the plant at their printing establishment to the main building plant.

He was next associated with the H. O'Neill Co., New York, where he was engaged in installing a new system of elevators, electric light and steam heating. He also installed a lead- and zinc-mining plant at Joplin, Mo., for the Devonshire Mining Co.

At the time of his death, April 27, 1919, Mr. Wheeler was actively connected with two concerns, the Trinity Engineering Co. and the Diamond Carbonating Co., both in New York City.

Mr. Wheeler belonged to several fraternal organizations. He became a member of the Society in 1905.

HUGH M. WILSON

Hugh M. Wilson, who, until his resignation in January, 1917, was for six years first vice-president of the McGraw Publishing Co., died suddenly on September 19 at his home in Stockbridge, Mass. Mr. Wilson was born in June 1866 in Jacksonville, Ill. He was a graduate of the Illinois College.

His first journalistic work was as a reporter for the daily press of the middle west. In 1899, however, he joined the staff of the *Mississippi Valley Lumberman* and since that time he devoted his great energy and ability to the development of magazines in the trade and technical field. It was through the medium of the *Northwestern Railroader* that he entered the railroad field in which he was to become a great leader. That journal was soon consolidated with *Railway Age*. Mr. Wilson becoming secretary-treasurer of the new organization. His dual training in editorial and business affairs proved so valuable that he subsequently undertook, in addition to his former duties, those of manager of *Railway Age*. He was elected president of the company in 1899. In 1906 the Wilson Co. was organized and took over the publication of *Railway Age* and the *Electric*

Railway Review. Two years later Mr. Wilson sold both papers. The *Railway Age* was consolidated with the *Railway Gazette* while the *Electric Railway Review* was purchased by the McGraw Publishing Co. and combined with the *Electric Railway Journal.* After a year as vice-president of the Barney & Smith Car Co., Dayton, Ohio, Mr. Wilson joined the McGraw Publishing Co.

Mr. Wilson became an associate of the Society in 1903.

EMIL F. BREYER

Emil F. Breyer was born in 1885 in Alsace Lorraine. He was educated abroad, being graduated from the Friedrichs Polytechnikum, Germany, in 1908 with the degree of M. E. and served his shop apprenticeship in Montigny and Metz, Germany. Upon his coming to the United States he became connected with the Doe Run Lead Co., Rivermines, Mo., where he worked for sixteen months as draftsman and designer on mill and power-plant layouts. He then became associated with the St. Joseph Lead Co., also in Rivermines, as mechanical engineer in charge of the shops, power plants, mining and milling machinery; he was also assistant to the mechanical superintendent of the company. At the time of his death, November 18, 1919, Mr. Breyer was connected with the Meloy Motor Co., Globe, Ariz.

Mr. Breyer became an associate-member of the Society in 1915.

FREDERICK LINCOLN EMORY

Frederick L. Emory, professor of mechanics and applied mathematics in the University of West Virginia, died on December 31, 1919. Professor Emory was born in Lunenburg, Mass., on April 9, 1867. He received his common school education there and attended college at Worcester Polytechnic Institute, from which he received his bachelor's degree in mechanical engineering in 1887. For the next two years he was the director of the mechanical and manual training department of the Washington, D. C., high school. During 1890 and 1891 he established and superintended the Trades School in the Massachusetts Reformatory, Concord Junction, Mass.

In the fall of 1891 he was selected by the late Dr. Eli Marsh Turner, then president of the University of West Virginia, to establish and take charge of the new department of mechanical engineering. This he did most successfully, constructing the first mechanical hall on the site now occupied by the University heating plant. From 1892 to 1895 he was engaged in establishing and directing the Industrial Training and Technical High School for the city of Indianapolis, Ind. In the fall of 1895 he entered Cornell University to take postgraduate work and in 1896 received the degree of master of mechanical engineering. He then returned to the University of West Virginia to become professor of mechanics and applied mathematics. In 1899, Worcester Polytechnic Institute, his alma mater, conferred upon him the degree of master of engineering.

Professor Emory became a member of our Society in 1894. He was also a member of the Society for the Promotion of Engineering Education.

BARTHOLD D. GERDAU

Barthold D. Gerdau, who became a member of the Society in 1911, died in Dusseldorf, Germany, in 1918. Mr. Gerdau was born in Altona, Germany, in October, 1852. He was educated in the technical high school at Brunswick, the Latin gymnasium at Altona and the industrial schools at Hamburg and Altona. He served his apprenticeship with an engineering firm in Altona and with the Reiherstieg Shipyard and Engineering Works at Hamburg.

He then became connected with the Bolckow & Vaughan Co., Middlesborough, England, in the construction and erection of new steel works at Eaton. Three years later he resigned from his position as engineer and designer in the hydraulic department to engage in work for Sulzer Brothers, Winterthur, Switzerland, as superintendent of construction of hydraulic machinery for cutting the Arlburg tunnel and also tunnels on the Gotthard Railroad.

In 1895 Mr. Gerdau became associated with the firm of Haniel & Lueg, Dusseldorf, Germany, executing many important installations for the firm, including a complete hydraulic plant and hydraulic center power station for the Hamburg State Government, the large hydraulic swing bridges over the North Baltic Canal and the hydraulic ship lift at the Dortmund-Ems Canal near Henrichenburg. At the time of his death, Mr. Gerdau was chief engineer and managing director of the firm.

RALPH B. HAMILTON

Ralph B. Hamilton was born in Toledo, Ohio, on April 11, 1875. He received his early education in the public schools of Saginaw, Mich., and later attended the Royal Institute of Technology, Dresden, Saxony, and also Cornell University from which he received his M. E. degree in 1896.

Upon graduation Mr. Hamilton began his career as a draftsman with the Buffalo Engineering Co. From 1897 to 1898 he was assistant engineer of the Howard Iron Works, Buffalo. The following two years he spent as assistant manager of the Iroquois Iron Works also in Buffalo.

In 1901 Mr. Hamilton moved to St. Catherines, Ont., as acting manager of the Packard Electric Co. Six years later he was made secretary-treasurer of the company and in 1912 was appointed president and general manager, serving in that capacity until his departure in 1919 to accept the position of vice-president and production manager of the National Safe Co., Cleveland.

Mr. Hamilton was exceedingly active in business life, being chairman of the Board of Directors of the Packard Electric Co., president of the Precision Manufacturing Co., St. Catherines, president of the Cary Safe Co., Buffalo, president of the Packard Fuse Co., Ltd., and president of the Canadian Standard Products Co., Ltd., all in St. Catherines.

Besides belonging to many clubs of a purely social nature, Mr. Hamilton was also an associate-member of the American Institute of Electrical Engineers. He became a member of the Society in 1907. He died on December 24, 1919.

GEORGE EDWARD RICHARDSON *

George E. Richardson was born in Lowell, Mass., September, 1875. He received his early education in the Lowell schools and then attended Tufts College from which he received his B. S. degree in 1897. Upon graduation he became connected with the Lancaster Mills as electrician, laying out and planning the equipment for the machine shop at the Cavite Navy Yard; later he was advanced to the position of electrician in charge, supervising the repairs and installations in the Asiatic Station and installing complete equipment on the *Don Juan de Austria*, *Isla de Cuba*, *Isla de Luzon*, *Princeton*, etc. In 1908 Mr. Richardson became associated with the General Electric Co. as mill power engineer, in charge of lighting installations.

Mr. Richardson died in the Service in 1918. He became an associate of the Society in 1911.

CHARLES W. WEICK

Charles W. Weick, assistant professor of mechanical drawing and design, Teachers' College, Columbia University, died on November 5, 1919. Professor Weick was born on October 26, 1864, in Newark, N. J. He received his early education in Buffalo, N. Y., and later attended Ohio State University, where he taught manual training for six years and studied at the same time, receiving his B.S. degree in 1898.

Prior to this date Mr. Weick had been connected with the J. P. Ross Co., Buffalo; the Johnson Steel Rail Co., Johnstown, Pa.; the Buffalo Forge Co., Buffalo, and the Horner Machine Co., Holyoke, Mass., obtaining his shop and drawing-room experience and becoming acquainted with the design and construction of complicated automatic machinery. In 1899 he left Ohio State University to accept the position of instructor of manual training at Teachers' College, Columbia University. He was later appointed assistant professor of mechanical drawing and design, which position he held at the time of his death.

Professor Weick was the author of two books on drawing: Elementary Mechanical Drawing, published in 1915, and Mechanical Drawing Problems, published in 1917. He became a member of the Society in 1901.

No. 1736

INDEX TO VOLUME 41

NOTE

Names of authors and discussers, also those of deceased members preceding an obituary notice are in caps and small caps. A discusser is distinguished from the author of a paper by (*D*), placed after the name of the paper.

Titles of papers, where placed after the name of the author, and appearing in their exact form, are in italics. Papers are indexed not under their titles but under their subject-matter.

[1] See MECHANICAL ENGINEERING, September 1919.
[2] See MECHANICAL ENGINEERING, May 1919.

[1] See MECHANICAL ENGINEERING, October 1919.

[1] See MECHANICAL ENGINEERING, October 1919.

[1] See MECHANICAL ENGINEERING, October 1919.
[2] See MECHANICAL ENGINEERING, September 1919.
[3] See MECHANICAL ENGINEERING, April 1919.

[1] See MECHANICAL ENGINEERING, September 1919.

[1] See also MECHANICAL ENGINEERING, September 1919.
[2] See MECHANICAL ENGINEERING, December 1919.

[1] See MECHANICAL ENGINEERING, April 1919.
[2] See MECHANICAL ENGINEERING, October 1919.

[1] See MECHANICAL ENGINEERING, October 1919.
[2] See MECHANICAL ENGINEERING, May 1919.
[3] See MECHANICAL ENGINEERING, September 1919.
[4] See MECHANICAL ENGINEERING, December 1919.

ONE DAY